COMPREHENSIVE HEAT EXCHANGER PERFORMANCE EVALUATION METHOD ON OCEAN THERMAL ENERGY CONVERSION FOR MAXIMUM NET POWER

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ABSTRACT

Ocean thermal energy conversion (OTEC) generates electricity through a heat engine by exploiting the temperature difference between the ocean surface and deep seawater. Since the effective temperature difference for the heat engine depends on the heat exchangers' performance, the irreversible loss in the heat exchangers should be reduced. However, heat exchangers with high heat transfer performance generally have a large pressure drop, and the finite heat transfer performance and the fluid flow resistance cause irreversible loss during the heat transfer process. Performance evaluation methods for heat exchangers generally evaluate the heat transfer performance and pressure drop separately. The objective of this study was to develop a performance evaluation method that could be used for maximizing the net power of OTEC based on finite-time thermodynamics (FTT). On the basis of the FTT model, the relationship between a heat exchanger's performance for an irreversible heat engine and the net power output per heat transfer area is presented. Additionally, a comprehensive performance index for the heat exchanger is proposed. The performance evaluation procedure and the comparison of the plate heat exchanger performance are conducted to show the effectiveness of the performance evaluation method. The results show that the proposed evaluation method presents the beak down of the performance ratio of heat exchangers, which indicates the quantitative performance for every heat exchanger in terms of the maximum net power generation of OTEC.

1 INTRODUCTION

Global energy consumption continues to increase with global economic growth. While petroleum and coal have been the main energy sources so far, the emission of greenhouse gases associated with their consumption exacerbates global warming. An ocean thermal energy conversion (OTEC) system is an ocean-based renewable power generation system, and this technology has abundant potential, especially in tropical oceans. OTEC exploits the vertical temperature gradient in the ocean to drive a heat engine (Avery and Wu, 1994), thereby converting thermal energy into electricity. OTEC can be used to generate electricity stably 24/7, and it is not affected by sudden changes in the weather. Furthermore, the thermal energy discharged by the OTEC system can be used for seawater desalination, and the deep ocean water effluent can be used in industries such as aquaculture, agriculture, and cosmetics (Martine et al., 2016). Recently, 100-kW-class demonstration plants have been operated in Japan and the US, and a one MW demonstration plant was tested in Korea in 2019. The Korean plant produced 338 kW using a heat source temperature difference of 18.7 °C (IEA-OES, 2019).

Since the available temperature difference in OTEC is very small, the design philosophy used to achieve maximum power is completely different from that of the conventional power plant that maximizes thermal efficiency. In particular, the achievement of the optimum heat and mass balance, which maximize the work from heat engine in OTEC design, requires the consideration of the balance of the heat transfer rate and heat engine operating temperature range (Yasunaga and Ikegami, 2020a). The available power output in OTEC can be maximized using the concept of finite-time thermodynamics (FTT). This involves considering (1) energy conversion performance evaluations such as the normalization of the thermal and exergy efficiencies (Yasunaga and Ikegami, 2017, 2019 and 2020a; Yasunaga et al., 2021) and (2) the degradation of power generation due to the performance components including back work ratio associated with seawater feed pumps that consume part of the generated electricity. An FTT model is applicable to the performance evaluation and design optimization as well as the performance evaluation of its components.

In OTEC, the optimization and evaluation of the heat exchanger are important because of the very low available temperature of the heat source, and this component is one of the most significant generator of the pressure drop that increases the workload of seawater feed pumps. In other words, the pressure drop in the heat exchangers on the heat source side is dominant for these pumps. Since there is a trade-off between this pressure drop and the heat transfer performance, the selection of the heat exchangers is important in OTEC design (Yasunaga et al., 2018; Fontaine et al., 2019). While heat exchangers' performance is generally evaluated as the overall heat transfer coefficient and the pressure drops independently, or the heat transfer coefficient as Nusselt number and the friction factor as function of Reynolds number, in OTEC design, the performance evaluation of heat exchangers requires the optimization of the system performance, including the total system design (Owens, 1980; Uehara and Ikegami, 1990). And some OTEC system design parameters depend on site conditions such as the intake pipe configuration and available seawater temperature. Therefore, the development of an evaluation method for heat exchangers is imperative for the application to practical OTEC design. Wu et al. (2019) and Wu et al. (2020) proposed the optimization of the evaporator and condenser on the basis of the constructal theory, respectively. However, the objective function has a weighting function and the optimization depends on the weight and the function has no clear thermodynamic implication. Yasunaga et al. (2018) proposed the application of an FTT model to the performance evaluation of plate heat exchangers to maximize the net power; their performance evaluation method involved simplifying the overall heat transfer performance and the pressure drop as a mean velocity of the heat source. The performance degradation of the heat engine resulting from the working fluid pressure drop in the heat exchanger was not considered in their model. Yasunaga and Ikegami (2020b) extended the FTT model to include the irreversibility of the heat engine to evaluate the working fluid performance.

This study developed an FTT-model-based comprehensive performance evaluation method and index to apply for evaluation of single heat exchanger applicable to either of an evaporator or a condenser in terms of the relationship between the heat transfer performance and the pressure drops on the heat source and working fluid sides and between the heat transfer performance and the net power output of OTEC. This paper summarizes the basic equations of the FTT model, describes a comprehensive evaluation method for a single heat exchanger, and compares the performance of plate heat exchangers on the basis of available reference data on them.

2 BASIC EQUATIONS OF FTT MODEL

2.1 Maximum Power of Heat Engine

Figure 1 shows a model of an OTEC power generation system with a heat engine. Heat is transferred at the rate Q_{WS} from warm seawater at the temperature T_{WS} to a heat engine. This engine converts the thermal energy into work W, and heat is transferred at the rate Q_{CS} from the heat engine to deep seawater at the temperature T_{CS}. The energy conservation principle and heat transfer rate can be expressed as

$$W = Q_{WS} - Q_{CS} \tag{1}$$

$$Q_{WS} = C_{WS}(T_{WS} - T_{WSO}) = (UA\Delta T_m)_E \tag{2}$$

$$Q_{CS} = C_{CS}(T_{CSO} - T_{CS}) = (UA\Delta T_m)_C \tag{3}$$

where C is the heat capacity flow rate (product of the mass flow rate and specific heat) T is the temperature, U is the overall heat transfer coefficient, A is the heat transfer area, ΔT_m is the logarithmic mean temperature difference, and the subscripts WS, CS, and O indicate warm source, cold source, and outlet conditions, respectively. T_H and T_L are the highest and lowest temperatures of the heat engine, and for a pure working fluid, they are the boiling and condensing temperatures, respectively. The

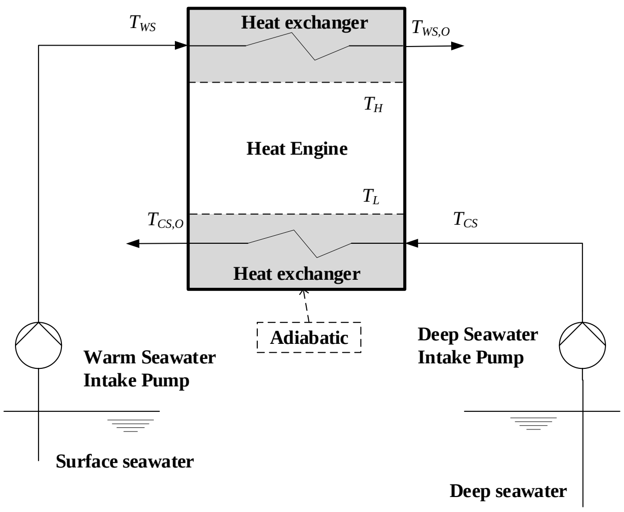

Figure 1: Model of an OTEC power generation system involving a heat engine.

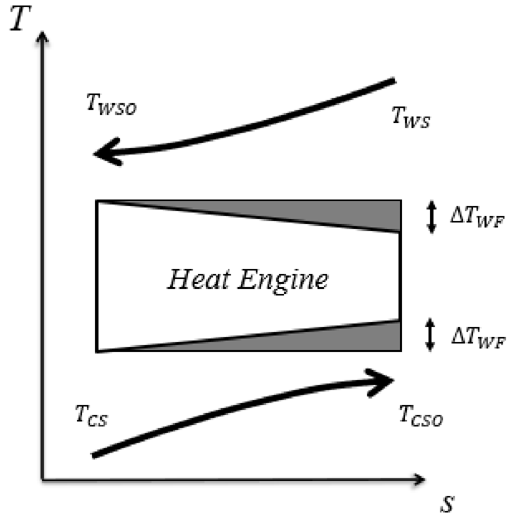

Figure 2: Conceptual diagram of irreversibility in an OTEC heat engine associated with working fluid pressure drop.

logarithmic mean temperatures in the case of the counter flow of an evaporator $(\Delta T_m)_E$ and in a condenser $(\Delta T_m)_C$ are given by

$$(\Delta T_m)_E = \frac{(T_{WS} - T_H) - (T_{WSO} - T_H)}{\ln\left(\frac{T_{WS} - T_H}{T_{WSO} - T_H}\right)}, (\Delta T_m)_C = \frac{(T_{CS} - T_L) - (T_{CSO} - T_L)}{\ln\left(\frac{T_{CSO} - T_L}{T_{CS} - T_L}\right)} \tag{4}$$

The entropy balance in the heat engine is

$$\oint ds = s_{in} - \emptyset s_{out} = \frac{Q_{WS}}{T_H} - \emptyset \frac{Q_{CS}}{T_L} = 0 \tag{5}$$

where Φ indicates the irreversible loss in the heat engine, including the irreversibility of the expansion (turbine) and compression (working fluid pump) processes, and flow resistance of the working fluid in piping and heat exchangers. From Equations (1)–(5), the work output of the heat engine as a function of T_{WSO} or T_{CSO} is one degree of freedom, and the maximum work output can then be derived from $dW / dT_{WSO} = 0$ or $dW / dT_{CSO} = 0$ with assumptions that Φ and ε are independent of seawater temperatures. The maximum work obtained by considering the heat exchanger performance $W_{m,NTU,\Phi}$ and the output temperature of heat source defined as optimum outlet temperatures $T_{WSO,opt}$ and $T_{CSO,opt}$, respectively, can be expressed as (Ibrahim et al., 1992, Yasunaga and Ikegami, 2020b)

$$W_{m,NTU,\phi} = \frac{\emptyset \varepsilon_E \varepsilon_C C_{WS} C_{CS}}{\varepsilon_E C_{WS} + \phi \varepsilon_C C_{CS}} \left(\sqrt{T_{WS}} - \sqrt{\frac{T_{CS}}{\phi}}\right)^2 \tag{6}$$

$$T_{WSO,opt} = \sqrt{T_{WS}} \frac{\{\varepsilon_E C_{WS} + \emptyset \varepsilon_C (1 - \varepsilon_E) C_{CS}\}\sqrt{T_{WS}} + \varepsilon_E \varepsilon_C C_{CS} \sqrt{\emptyset T_{CS}}}{\varepsilon_E C_{WS} + \emptyset \varepsilon_C C_{CS}} \tag{7}$$

$$T_{CSO,opt} = \sqrt{T_{CS}} \frac{\{\varepsilon_E (1 - \varepsilon_C) C_{WS} + \emptyset \varepsilon_C C_{CS}\}\sqrt{T_{CS}} + \varepsilon_E \varepsilon_C C_{WS} \sqrt{\emptyset T_{WS}}}{\varepsilon_E C_{WS} + \emptyset \varepsilon_C C_{CS}} \tag{8}$$

where ε is the heat transfer effectiveness defined as follows by using the net transfer unit (*NTU*):

$$\varepsilon_E = 1 - e^{-NTU_E}, \varepsilon_C = 1 - e^{-NTU_C} \tag{9}$$

$$NTU_E = \frac{(UA)_E}{C_{WS}}, NTU_C = \frac{(UA)_C}{C_{CS}} \tag{10}$$

As shown in Equation (6), the maximum work output depends on the seawater temperatures and heat capacities, heat exchanger performances, and the irreversibility of heat engine based on FTT, although, in general, the irreversibility of heat engine is mainly discussed in the thermodynamics such as the performance of turbine and working fluid pump.

2.2 Relationship between Net Power and Heat Exchanger Performance

Owing to the very low temperature difference of seawater in OTEC, the heat source pumping consumes a large amount of the generated power. Since the pumping power is not negligible for the work of the heat engine, it is necessary to examine the internal power consumption for the performance evaluation of the power generation system. In this study, the net power W_{net} was defined as

$$W_{net} = (W_T - P_{WF}) - (P_{WS} + P_{CS}) \tag{11}$$

where W_T is the turbine output, P is the pump power, and the subscripts WF, WS, and CS denote working fluid and warm and cold seawater, respectively. Since $W_T – P_{WF}$ is the heat engine output, when the power output is maximized, Equation (11) becomes

$$W_{net} = W_{m,NTU,\emptyset} - (P_{WS} + P_{CS}) \tag{12}$$

The pumping power is defined as

$$P = \Delta p \frac{\dot{m}}{\rho \eta_P} \tag{13}$$

where ρ is the density, η_P is the efficiency of pump, and Δp is the pressure difference between the inlet and the outlet of the pumps. It can be confirmed that the maximum work of the heat engine $W_{m,NTU,\Phi}$ and the seawater feed pump power $P_{WS} + P_{CS}$ increase with the heat source flow rate, and the net power at the maximum work $W_{m,net,NTU,\Phi}$ is theoretically presence at an optimum flow rate. Ikegami and Bejan (1998) obtained the maximum net power at the maximum work by considering infinite heat transfer performance. The existence of a pressure drop in the heat exchangers generally leads to a large ΔP. In the design of a heat exchanger to maximum the net power generation, the balance of the heat transfer and pressure drop in the heat exchanger is crucial.

3 APPLICATION TO PEFORMANCE EVALUATION OF HEAT EXCHANGERS

3.1 Proposal of Single Heat Exchanger Performance Evaluation Index

The basic equations in Section 2 can be used for the performance evaluation of a single OTEC heat exchanger, either of an evaporator or a condenser. For the examination of the index indicating the overall performance of the single heat exchanger, the two heat exchangers in the system were assumed to show identical performance and defines as a single heat exchanger and a heat source, respectively (i.e., $NTU_E = NTU_C \equiv NTU_{HE}$, $C_{WS} = C_{CS} \equiv C_{HS}$, $P_{WS} = P_{CS} \equiv P_{HS}$). Furthermore, the mechanical efficiency of the pumps was assumed to be 100%. From these assumptions, the net power at the maximum work in equation (6) will be

$$W_{m,netNTU,\phi} = \frac{\emptyset \varepsilon_{HE} C_{HS}}{1+\phi}\left(\sqrt{T_{WS}} - \sqrt{\frac{T_{CS}}{\phi}}\right)^2 - 2P_{HS} \tag{14}$$

In conventional studies, the working fluid saturation temperature in a heat exchanger is constant since the pressure drop on the working fluid side is ignored (Yasunaga, et al., 2018). However, to consider the working fluid pumping power, we considered a cycle in which the saturation temperature of the working fluid in the heat exchanger changed as shown in Figure 2. From Equations (1) – (5), the pressure difference between inlet and outlet associated with the pressure drop in the working fluid operating temperature, $\Delta T_{WF,\phi}$, resulting from the irreversibility in the heat engine due to the pressure drop can be expressed as

$$\Delta T_{WF,\emptyset} = \frac{(1-\emptyset)(T_{WS}+T_{CS})}{2(1+\emptyset)} \tag{15}$$

Under the assumption that the pressure drop of the working fluid Δp_{WF} affects only ϕ in Eq.(16), $\Delta T_{WF,\phi}$ corresponds to the temperature difference of the saturated vapor $\Delta T_{WF,\Delta PWF}$ at the pressures in Equation (17):

$$\Delta p_{WF} = p_{(T_{sat}),in} - p_{(T_{sat}),out} \tag{16}$$

$$\Delta T_{WF,\Delta P_{WF}} = T_{sat,in} - T_{sat,out} \tag{17}$$

where T_{sat} denotes the saturation temperature of the working fluid and the subscripts *in* and *out* indicate inlet and outlet. Under the assumption that the working fluid states at the inlet and outlet of the heat exchanger are saturated liquid and saturated vapor, from Equations (15) and (17), the irreversible coefficient Φ can be calculated from the equation $\Delta T_{WF,\Phi} = \Delta T_{WF,\Delta PWF}$.

In a comparison of heat exchangers, the net power per heat transfer area is an important factor for determining their performance and the required amount of heat transfer area. The heat exchanger material in OTEC is usually titanium, which is expensive but resistant to seawater corrosion. The researches of the optimization design in OTEC system (Uehara and Ikegami, 1990, Vera, et al., 2020), the total heat transfer area of the heat exchanger over the net power generation represents an economic benchmarks. In the actual capital expenditure depends on the site, and social economic condition. Therefore, this research also adopts the net power per unit heat transfer area as the economic benchmark. The net power per total heat exchanger $W_{m,net,\Phi}$ / A, was used as an evaluation index

$$\frac{W_{m,NTU,\phi}}{A} = \frac{1}{A}\left[\frac{\emptyset\varepsilon_{HE}C_{HS}}{1+\phi}\left(\sqrt{T_{WS}} - \sqrt{\frac{T_{CS}}{\phi}}\right)^2 - 2P_{HS}\right] \tag{18}$$

From Equation (6), the maximum power when NTU is infinite and Φ equals one is

$$W_m = \frac{C_{HS}}{2}\left(\sqrt{T_{WS}} - \sqrt{T_{CS}}\right)^2 \tag{19}$$

Hence, the maximum work of a heat engine is determined only by the temperature and flow rate of the heat source. From Equations (14) and (19), the exergetic efficiency, which is defined as the ratio of the net available power from the system per the theoretical maximum power, can be obtained as

$$\frac{W_{m,netNTU,\phi}}{W_m} = \frac{\emptyset\varepsilon_{HE}}{1+\phi}\left(\frac{\sqrt{T_{WS}} - \sqrt{\frac{T_{CS}}{\phi}}}{\sqrt{T_{WS}} - \sqrt{T_{CS}}}\right)^2 - \frac{4P_{HS}}{C_{HS}\left(\sqrt{T_{WS}} - \sqrt{T_{CS}}\right)^2} \tag{20}$$

Note that since this research objective is to evaluate single heat exchanger either of an evaporator or a condenser, the concept of exergetic efficiency in Eq. (20) is only applicable for that purpose.
Considering the characteristics of Equations (18) and (20), we propose an evaluation index ω given by

$$\omega = \frac{W_{m,netNTU,\phi}}{W_m A} = \frac{1}{A}\left[\frac{\emptyset\varepsilon_{HE}}{1+\phi}\left(\frac{\sqrt{T_{WS}} - \sqrt{\frac{T_{CS}}{\phi}}}{\sqrt{T_{WS}} - \sqrt{T_{CS}}}\right)^2 - \frac{4P_{HS}}{C_{HS}\left(\sqrt{T_{WS}} - \sqrt{T_{CS}}\right)^2}\right] \tag{21}$$

For a basic evaluation of plate heat exchangers, A can be considered as a single-path heat transfer area. Since the net power per heat transfer area in Equation (18) represents the unit heat transfer area performance in a single path of the heat exchanger, we defined ω as the effectiveness of the available thermal energy from the heat source (i.e., a larger ω requires a lower seawater flow rate for a given net power per heat transfer area).

3.2 Heat Exchanger Performance Evaluation Procedure

The overall heat transfer coefficient U can be expressed as

$$U = \frac{1}{\frac{1}{\alpha_{HS}} + \frac{t}{\lambda_P} + \frac{1}{\alpha_{WF}}} \tag{22}$$

where α and λ_P are the heat transfer coefficient and thermal conductivity, respectively, and t (in meters) is the plate thickness. In general, the heat transfer coefficient of a working fluid that has undergone a phase change is considerably greater than that of seawater for single-phase forced convection; hence, if $\alpha_{WF} >> \alpha_{HS}$, U is regarded as a function of α_{HS}. For single-phase forced convention, the Nusselt number is a function of the Reynolds number and Prandtl number, and the Reynolds number is a function of the mean velocity in the heat exchanger when the temperature change in the heat exchanger is small. Therefore, U and ΔP will be functions of the mean velocity V.

The evaluation of the heat exchanger performance involved the following steps.

1. Approximate formulas for the overall heat transfer coefficients and pressure drops were formulated as a function of the mean heat source velocity V_{HS} by fitting experimental data:

$$U = \zeta V_{HS}^{\beta}, \Delta P_{HS} = \xi V_{HS}^{\theta}, \Delta P_{WF} = \varsigma V_{WF}^{\gamma} \tag{23}$$

   where the working fluid was assumed to have a mean vapor quality x of 0.5. Note that the constants in Equation (23) in this research are determined by the approximation of the experimental data in the reference.

2. The maximum net power was determined by varying V_{HS} (C_{HS}) in Equation (18) and obtaining the maximum $W_{m,net,NTU,\Phi}$ / A at a mean velocity corresponding to the optimum mean velocity of seawater. The working fluid's mass flow rate was determined from heat balances:

$$m_{WF} = \frac{C_{WS}(T_{WS} - T_{WSO})}{L_E}, m_{WF} = \frac{C_{CS}(T_{CSO} - T_{CS})}{L_C} \tag{24}$$

   where L is the latent heat and the subscripts E and C indicate evaporator and condenser sides, respectively.

3. The parameter ω was calculated using Equation (21) and it served as a performance index of the OTEC heat exchanger.

4 RESULTS AND DISCUSSION

The proposed performance evaluation method was applied to existing plate heat exchangers. Table 1 provides the specifications of the herringbone type heat exchangers, and the coefficients in Equation (23) obtained by using experimental data from other researches are presented in Table 2. The plate No.1–3 are evaporators and No.4 is a condenser.

Figure 3 shows the relationship between the mean heat source velocity in the plate and the net power per heat transfer area. The hollow circles show the maximum points. Clearly, for all the plates, the maximum $W_{m,net,NTU,\Phi}$ / A occurred in the V_{HS} range of 0.0 – 1.0 m/s. A comparison of the maximum points in Figure 3 shows that the curve of Plate 2–4 are prominently higher than that of Plate 1.
According to Fig. 3, Plate 1 has maximum point at the lowest V_{HS} among the plates. Theoretically, an ideal plate provides a higher $W_{m,net,NTU,\Phi}$ at a lower V_{HS}.
Table 3 shows the heat exchange performance at the optimum flow velocity $V_{HS,opt}$ shown in Fig. 3.
Although Plate 2 showed the highest value of $W_{m,net,NTU,\Phi}$ / A, but the performance evaluation index ω of Plate 1, which reached the maximum value at the lowest heat source mean velocity, was the highest among the four plates. Plates 2–4 showed almost identical values of the maximum net power per heat transfer area, but ω of Plate 3 was the highest among those three plates. This reason is the difference of the width of plates against flow path of heat source in each heat exchanger, because W_m in Eq. (19) is the function of heat capacity of heat source.

Table 1: Heat exchanger specifications

PHE	Plate 1	Plate 2	Plate 3	Plate 4
Type of heat exchanger	Plate	Plate	Plate	Plate
Length (m)	1.83	1.83	1.83	1.47
Width (m)	0.82	0.82	0.82	0.55
Plate thickness (mm)	0.61	0.61	0.61	1.00
Clearance (mm)	3.91	3.91	3.91	3.40
Chevron angle (deg)	High	Low	Mix	58
Working fluid	Ammonia	Ammonia	Ammonia	R-22
Number of plates	100	100	100	30
Reference	Panchal et al., 1984	Panchal et al., 1984	Panchal et al., 1984	Uehara et al., 1990

Table 2: Approximation coefficients

PHE	U		ΔP_{HS}		ΔP_{WF}	
	ζ	β	ξ	θ	ς	γ
Plate 1	2.362	0.131	395.6	1.907	48.89	0.449
Plate 2	1.757	0.236	62.8	2.259	15.41	0.269
Plate 3	2.234	0.170	178.9	2.078	24.85	0.327
Plate 4	1.961	0.268	89.8	1.799	11.76	0.204

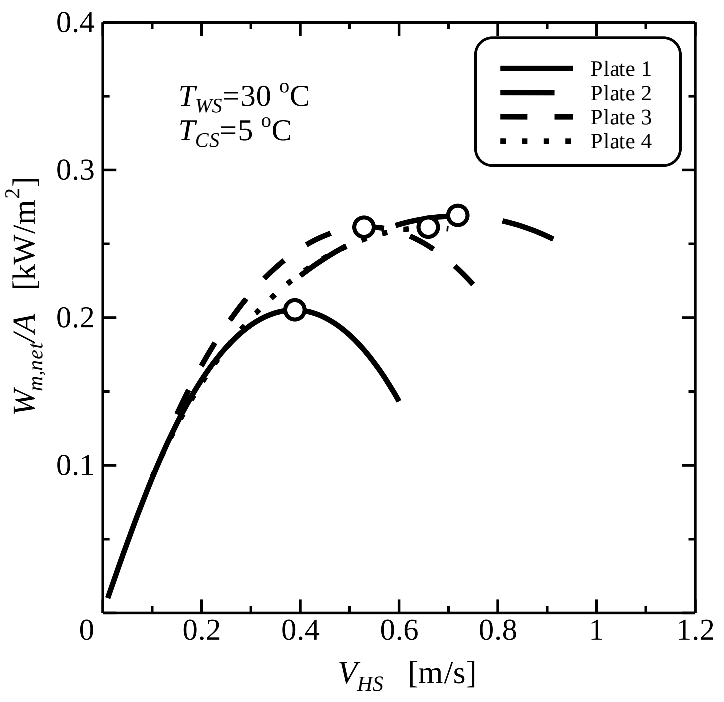

Figure 3: Comparison of the net power output per heat transfer area $W_{m,net}/A$ as a function of heat source velocity V_{HS}. The hollow circles show the maximum point at optimum heat source velocity.

Table 3: Heat exchanger conditions at the optimized mean heat source velocity

PHE	Plate 1	Plate 2	Plate 3	Plate 4
$V_{HS,opt}$ (m/s)	0.39	0.72	0.53	0.66
U_{HS} (kW/m^{2}K)	2.088	1.626	2.005	1.754
ε_{HS} (-)	0.748	0.441	0.622	0.491
ΔP_{HS} (kPa)	65.7	29.9	47.8	42.5
ΔP_{WF} (kPa)	32.8	12.6	19.6	11.0
Φ (-)	0.992	0.997	0.995	0.996
$W_{m,net}$ / A (kW/m^2)	0.205	0.269	0.261	0.261
$W_{m,net}$ / W_m (-)	0.503	0.357	0.472	0.373
ω (1/m^2)	0.213	0.203	0.211	0.190

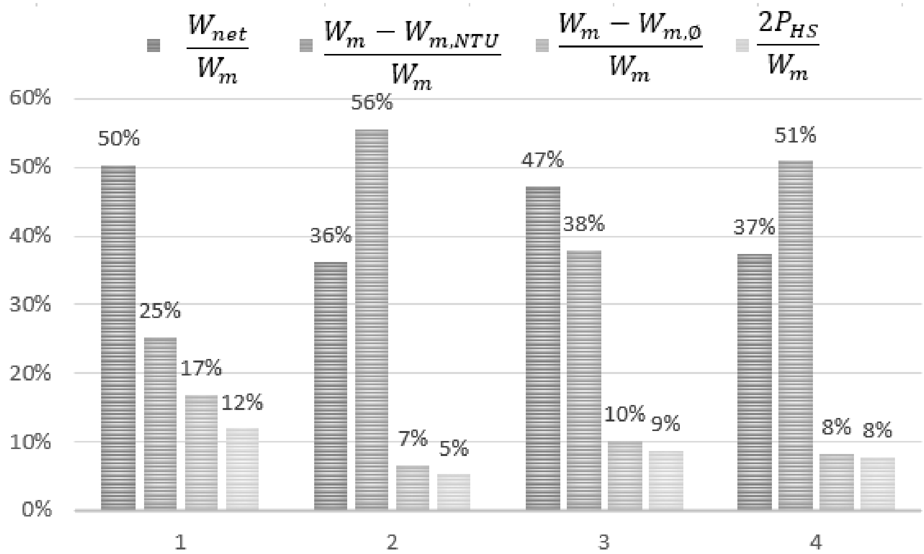

Figure 4: Comparison of heat exchanger performance in terms of the net power and net power degradation by het transfer performance, working fluid and heat source working fluid.

Figure 4 illustrates the breakdown of the ratio of net power W_{net}/W_m and degradation of available maximum power in heat transfer performance $(W_m - W_{m,NTU})/W_m$, in working fluid pressure drop $(W_m - W_{m,\phi})/W_m$, and in heat source pressure drop ($2P_{HS}/W_m$), respectively. According to Fig. 4, the net power ratio of Plate 1 is the highest of 50%. Significantly, although the degradation of net power due to pressure drops in both of working fluid and heat source are the highest, the net power degradation ratio in heat transfer performance in Plate 1 is about the half of the others. Therefore, in terms of the net available power, the balance of heat transfer performance and the pressure drop is totally important, and quantitatively, this analysis method visualized the effectiveness of the each element on the net available power in each heat exchanger.

5 CONCLUSIONS

This research applies FTT model to the heat exchanger performance evaluation on OTEC. The model newly proposed the irreversibility in a heat engine equivalent to the available net power degradation due to the working fluid pressure drop. Here, a thermodynamics-based performance index is proposed, and a performance evaluation procedure is explained. Firstly, from the economical point of view, the optimum operational condition of the heat source, which performs the maximum net power per heat transfer area, is clarified. Secondly, the proposed heat exchanger performance evaluation index at the optimum condition is easily calculate, and this method is also able to visualize the effectiveness of the heat transfer performance, and available power degradation due to working fluid pressure drop and heat source pressure drop. In this research, using reference experimental data, the performance of four plate heat exchangers was compared and heat exchangers with the best performance were identified.

Current research develops the conventional research on the heat exchanger performance evaluation based on finite-time thermodynamics incorporating with the working fluid pressure drop as the irreversibility of the heat engine. This research uses one typical temperature condition on OTEC, although the performance evaluation method depends on the heat source temperatures. Therefore, the sensitivity analysis on the heat source temperature difference to be confirmed for the practical application to designs and developments of the heat exchangers on OTEC.

NOMENCLATURE

A	Heat transfer area (m^2)
C	Heat capacity flow rate (kW/K)
L	Latent heat (kJ/kg)
m	Mass flow rate (kg/s)
NTU	Net transfer unit
Q	Heat transfer rate (kW)
T	Temperature (K)
t	Plate thickness (m)
P	Pumping power (kW)
p	Pressure (kPa)
U	Overall heat transfer coefficient ($kW/m^2 \cdot K$)
V	Mean velocity (m/s)
W	Work (kW)
x	Vapor quality
α	Heat transfer coefficient ($kW/m^2 \cdot K$)
Δp	Pressure difference (kPa)
ΔT	Temperature difference (K)
ΔT_m	Logarithmic mean temperature difference (K)
ε	Heat transfer effectiveness
λ_P	Coefficient of thermal conductivity of plate ($kW/m^2 \cdot K$)
ρ	Density (kg/m^3)
Φ	Irreversible coefficient
ω	Performance evaluation index ($1/m^2$)

Subscripts

C	Condenser
CS	Cold source
CSO	Outlet condition for cold source
E	Evaporator
HS	Heat source
HE	Heat exchanger
in	Inlet condition
m	Maximum
net	Net
out	Outlet condition
sat	Saturated condition
T	Turbine
WF	Working fluid
WS	Warm source
WSO	Outlet condition for warm source

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