Sensitivity Analysis of A Closed Cycle Ocean Thermal Energy Conversion Power Plant

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***Abstract*—In order to have a thorough parameter identification of a closed cycle ocean thermal energy conversion (OTEC), sensitivity analysis method is suggested. It can illustrate the effects of each parameter variation on the output of the system and also can reveal their order of significance. In this paper, effect of different input parameters such as input warm and cold water temperatures, warm and cold water pipe diameters, warm and cold water velocity in pipes, and evaporator and condenser working temperatures on the net output power and efficiency of the system is investigated. A 100 MW power plant case study has been chosen for the sensitivity analysis process. The main advantage of this study is providing an acceptable range for system parameters which gives designers an insight to select system configuration properly.**

***Index Terms*—OTEC, Closed cycle, Sensitivity analysis, Efficiency, Net output power**

Nomenclature

A	Heat transfer surface area (m^2)
C_p	Specific heat at constant pressure $(\mathrm{kJ\,kg^{-1}K^{-1}})$
d	Pipe diameter (m)
D	Diameter (m)
f	Coefficient of friction
g	Acceleration of gravity $(\mathrm{m\,s^{-2}})$
h	Enthalpy $(\mathrm{kJ\,kg^{-1}})$
ΔH	Pressure loss (kPa)
k	Heat conductivity $(\mathrm{W\,m^{-1}K^{-1}})$
l	Length (m)
L	Latent heat $(\mathrm{kJ\,kg^{-1}})$
$\dot{m}$	Mass flow rate $(\mathrm{kg\,s^{-1}})$
p	Pressure (Pa)
P	Power (W)
Q	Heat flow rate (kJ)
Re	Reynolds number
T	Temperature $(^\circ\mathrm{C})$
ΔT_m	Logarithmic mean temperature difference $(^\circ\mathrm{C})$
U	Overall heat-transfer coefficient $(\mathrm{W\,m^{-2}K^{-1}})$
V	Velocity $(\mathrm{m\,s^{-1}})$
w	Width of plate (m)
Δy	Clearance (m)

Greek symbols

δ	Thickness (m)
η	Efficiency
ρ	Density $(\mathrm{kg\,m^{-3}})$
λ	Friction factor
μ	Dynamic viscosity $(\mathrm{Pa.s})$
ν	Kinematic viscosity $(\mathrm{m^2\,s^{-1}})$

Subscripts

C	Condenser
CS	Cold sea water
d	Density
eq	Equivalent
E	Evaporator
G	Generator
I	Input
max	Maximum
N	Net
O	Output
P	Piping, Pump
T	Total, Turbine
W	Wall
WF	Working fluid
WS	Warm sea water

I. Introduction

Ocean thermal energy conversion (OTEC) is a power generation method utilizes the temperature difference between warm surface water and cold deep water of the oceans to produce electricity. Therefore, OTEC is a reliable source of energy in islands near equator and also oil platforms owing to the capability of generating rather stable amount of electricity during the year and in all hours. Depending on the working fluid utilization, OTEC cycles are divided into open and closed cycles. Open cycles vaporize warm sea water as working fluid to drive the turbine. On the other hand, a closed cycle OTEC, was first proposed by D'Arsonoval in 1881 [1], uses high vapor density liquids such as freon, propane or ammonia as its working fluid [2]. Due to a small temperature difference (approximately 15 - 25 °C) between the surface water and deep water of the ocean, the Rankine-cycle efficiency is limited to be about 3% - 5%. Therefore, there has been considerable research effort aimed at improving the

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performance of an OTEC power plant. The effect of various working fluids on the performance of a cycle has been investigated by Kim et al. [3]. They also examined the influence of different cycles such as closed system, a regeneration system, and a Kalina system on improving the efficiency. Furthermore, hybrid cycles were developed to improve the efficiency of closed cycle OTEC power plants. For instance, Yamada et al. [4] proposed a solar-boosted OTEC system and stated that the net thermal efficiency of the operation with 20 °C solar boost is 2.7 times higher than that of OTEC operation under the daytime conditions at Kumejima Island. A combination of OTEC cycle with solar pond was introduced by Straatman et al. [5] to increase the temperature difference in the Rankine cycle, which leads to an improved efficiency of 12%.

Another approach leads to system efficiency increment is optimizing plant design parameters. Wu [6], [7] utilized a finite-time thermodynamic approach to a Rankine closed cycle OTEC analysis which gave a much more realistic heat engine power, specific power and efficiency prediction than did the classical Carnot ideal cycle. A detailed optimization was carried out by Uehara and Ikegami [8] using Powell method. In addition, Najafi et al. conducted multi-objective genetic algorithm in order to find optimum set of plant design parameters [9]. However, it must be noted that parameter selection plays an important role in optimization procedure. Thus, it needs a thorough understanding of the effects of parameters on system performance. To this goal, the sensitivity analysis methods are developed and widely used in engineering applications. Moreover, sensitivity analysis provides suitable background to compare the effects of each parameters on the desired system output. Yeh et al. [10] theoretically investigated the effects of the temperature and flow rate of cold seawater on the net output of a pilot OTEC plant. They only considered the net output power while the energy quality which is expressed in system efficiency is neglected.

In the present work, a sensitivity analysis is performed to investigate the effect of various independent design variables on efficiency and net output power of a closed cycle OTEC power plant. The main advantage of this study is providing an acceptable range for system parameters which gives designers an insight to select system configuration properly.

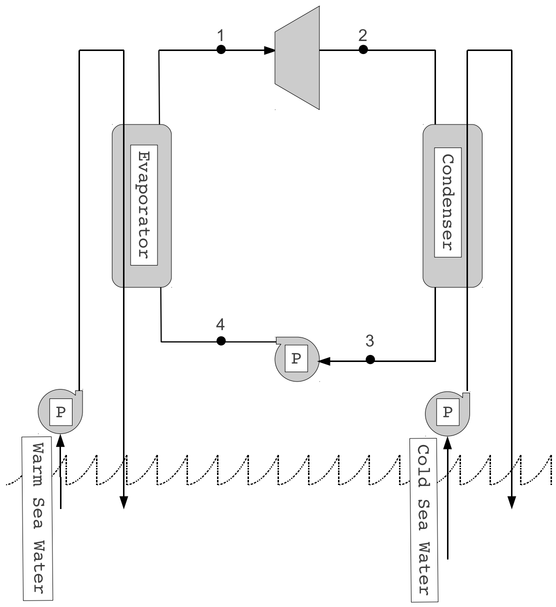

Fig. 1. Schematics of conventional closed Rankine cycle OTEC operation

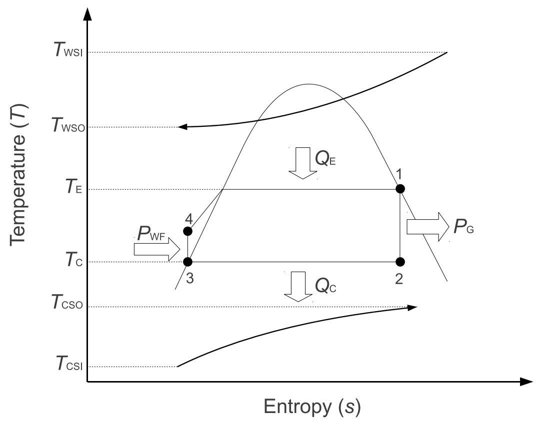

Fig. 2. T-s diagram of the closed Rankine cycle

II. System Model

Aside from the media used to accomplish the heat transfer, a closed cycle OTEC is the same as a Rankine cycle. The medium gains the heat through an evaporator from the ocean surface warm water, and produces power by passing through a turbine. Cold water from deep parts of the ocean cools the outlet flow from the turbine in a condenser and completes the cycle as shown in Fig. 1. The temperature-entropy (T-s) diagram of a closed Rankine cycle used in OTEC power plants is illustrated in Fig. 2. Heat transfer between pipes and sea water is assumed to be negligible, due to the massive amount of water flow rates and the large pipe diameters. Therefore, the temperature change of inlet cold sea water is not taken into account [8], [10].

The net power P_{N} of the OTEC operation is written as [8]

$$P_{\mathrm{N}} = P_{\mathrm{G}} - P_{\mathrm{TP}} \tag{1}$$

where P_{G} is the turbine generator power and P_{TP} is the total pumping power in a power plant defined as:

$$P_{\mathrm{TP}} = P_{\mathrm{WS}} + P_{\mathrm{CS}} + P_{\mathrm{WF}} \tag{2}$$

P_{WS}, P_{CS}, and P_{WF} are the pumping powers required for warm sea water, cold sea water, and the working fluid, respectively. The value of turbine generator power, P_{G} is calculated from the product of the mass flow rate of the working fluid and the adiabatic enthalpy difference between the evaporator and the condenser as:

$$P_{\mathrm{G}} = \dot{m}_{\mathrm{WF}} \eta_{\mathrm{T}} \eta_{\mathrm{G}} \left(h_1 - h_2\right) \tag{3}$$

where η_{T} and η_{G} are the turbine and generator efficiencies, respectively. Other power parameters shown in Eq. (2) are given as:

$$P_{\mathrm{WS}} = \dot{m}_{\mathrm{WS}} \Delta H_{\mathrm{WS}} \mathrm{g} / \eta_{\mathrm{WSP}} \tag{4}$$

$$P_{\mathrm{CS}} = \dot{m}_{\mathrm{CS}} \Delta H_{\mathrm{CS}} \mathrm{g} / \eta_{\mathrm{CSP}} \tag{5}$$

$$P_{\mathrm{WF}} = \dot{m}_{\mathrm{WF}} \Delta H_{\mathrm{WF}} \mathrm{g} / \eta_{\mathrm{WFP}} \tag{6}$$

where g is the acceleration due to gravity and $\dot{m}_{\mathrm{WS}}$, $\dot{m}_{\mathrm{CS}}$, $\dot{m}_{\mathrm{WF}}$, ΔH_{WS}, ΔH_{CS}, ΔH_{WF}, η_{WSP}, η_{CSP}, and η_{WFP} are mass flow rates, total pressure differences (i.e. head), and pump efficiencies of the piping used for warm sea water, cold sea water, and working fluid, respectively.

ΔH_{CS} is the total pressure difference of the cold sea water pipe, defined as:

$$\Delta H_{\mathrm{CS}} = \left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{P}} + \left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{C}} + \left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{d}} \tag{7}$$

where $\left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{P}}$ is the friction factor of the cold sea water straight pipe, defined as [8]:

$$\left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{P}} = 6.82 \frac{l_{\mathrm{CS}}}{{d_{\mathrm{CS}}}^{1.17}} \left(\frac{V_{\mathrm{CS}}}{100}\right)^{1.85} \tag{8}$$

where l_{CS} is the length of the cold sea water pipe, d_{CS} is the diameter of the cold sea water pipe, and V_{CS} is the velocity of the cold sea water. $\left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{C}}$ is the cold sea water pressure difference in the condenser which is defined by experimental data from related works for a specific type of condenser as [11]:

$$\left(\Delta H_{\mathrm{CS}}\right)_{\mathrm{C}} = 3.3 {V_{\mathrm{CS}}}^{1.95} \tag{9}$$

$\left(\Delta H_{\mathrm{CS}}\right)_{d}$ is the pressure difference caused by the density difference between the warm and the cold sea water, defined as:

$$\left(\Delta H_{\mathrm{CS}}\right)_{d} = l_{\mathrm{CS}} - \frac{1}{\rho_{\mathrm{CS}}} \left(\frac{1}{2} \left(\rho_{\mathrm{WS}} + \rho_{\mathrm{CS}}\right) l_{\mathrm{CS}}\right) \tag{10}$$

ΔH_{WS} is also the total pressure difference of the warm sea water pipe, defined as:

$$\Delta H_{\mathrm{WS}} = \left(\Delta H_{\mathrm{WS}}\right)_{\mathrm{P}} + \left(\Delta H_{\mathrm{WS}}\right)_{\mathrm{E}} \tag{11}$$

where the the value of $\left(\Delta H_{\mathrm{WS}}\right)_{\mathrm{P}}$ is evaluated like the cold sea water pipe calculation with warm water pipe properties. $\left(\Delta H_{\mathrm{WS}}\right)_{\mathrm{E}}$ is also the warm sea water pressure difference in evaporator and is calculated by the experimental data for a given evaporator in the same way as condenser [11]:

$$\left(\Delta H_{\mathrm{WS}}\right)_{\mathrm{E}} = 4.72 {V_{\mathrm{WS}}}^{2.12} \tag{12}$$

ΔH_{WF} is the total pressure difference of the working fluid piping, defined as:

$$\Delta H_{\mathrm{WF}} = \left(\Delta H_{\mathrm{WF}}\right)_{\Delta p} + \left(\Delta H_{\mathrm{WF}}\right)_{\mathrm{P}} + \left(\Delta H_{\mathrm{WF}}\right)_{\mathrm{C}} \tag{13}$$

$\left(\Delta H_{\mathrm{WF}}\right)_{\Delta p}$ is the saturation pressure difference between evaporator and condenser, defined as:

$$\left(\Delta H_{\mathrm{WF}}\right)_{\Delta p} = \frac{p_{\mathrm{E}} - p_{\mathrm{C}}}{\rho \mathrm{g}} \tag{14}$$

where ρ is the density of the working fluid.

The total heat transfer surface area, A_{T}, is given as:

$$A_{\mathrm{T}} = A_{\mathrm{E}} + A_{\mathrm{C}} \tag{15}$$

A_{E} and A_{C} are the heat transfer areas of the evaporator and the condenser which are shell and plate type heat exchangers and defined as:

$$A_{\mathrm{E}} = \frac{Q_{\mathrm{E}}}{U_{\mathrm{E}} (\Delta T_{\mathrm{m}})_{\mathrm{E}}} = \frac{\dot{m}_{\mathrm{WS}} C_{P_{\mathrm{WS}}} \left(T_{\mathrm{WSI}} - T_{\mathrm{WSO}}\right)}{U_{\mathrm{E}} (\Delta T_{\mathrm{m}})_{\mathrm{E}}} \tag{16}$$

$$A_{\mathrm{C}} = \frac{Q_{\mathrm{C}}}{U_{\mathrm{C}} (\Delta T_{\mathrm{m}})_{\mathrm{C}}} = \frac{\dot{m}_{\mathrm{CS}} C_{P_{\mathrm{CS}}} \left(T_{\mathrm{CSI}} - T_{\mathrm{CSO}}\right)}{U_{\mathrm{C}} (\Delta T_{\mathrm{m}})_{\mathrm{C}}} \tag{17}$$

where $\left(\Delta T_{\mathrm{m}}\right)_{\mathrm{E}}$ and $\left(\Delta T_{\mathrm{m}}\right)_{\mathrm{C}}$ are also logarithmic mean temperature differences of the evaporator and the condenser, respectively, defined as:

$$\left(\Delta T_{\mathrm{m}}\right)_{\mathrm{E}} = \frac{(T_{\mathrm{WSI}} - T_{\mathrm{E}}) - (T_{\mathrm{WSO}} - T_{\mathrm{E}})}{\ln \left(\dfrac{(T_{\mathrm{WSI}} - T_{\mathrm{E}})}{(T_{\mathrm{WSO}} - T_{\mathrm{E}})}\right)} \tag{18}$$

$$\left(\Delta T_{\mathrm{m}}\right)_{\mathrm{C}} = \frac{(T_{\mathrm{C}} - T_{\mathrm{CSI}}) - (T_{\mathrm{C}} - T_{\mathrm{CSO}})}{\ln \left(\dfrac{(T_{\mathrm{C}} - T_{\mathrm{CSI}})}{(T_{\mathrm{C}} - T_{\mathrm{CSO}})}\right)} \tag{19}$$

Q_{E} and Q_{C} are also the heat transfer rate of the evaporator and the condenser, respectively, defined as:

$$Q_{\mathrm{E}} = \dot{m}_{\mathrm{WF}} \left(h_1 - h_4\right) \tag{20}$$

$$Q_{\mathrm{C}} = \dot{m}_{\mathrm{WF}} \left(h_2 - h_3\right) \tag{21}$$

where $\dot{m}_{\mathrm{WF}}$ is the working fluid flow rate, given as:

$$\dot{m}_{\mathrm{WF}} = \frac{P_G}{\eta_{\mathrm{T}} \eta_G \left(h_1 - h_2\right)} \tag{22}$$

Calculating the system properties, power plant performance might be evaluated by first law efficiency. According to Carnot's theorem, maximum efficiency of a system working between two cold and warm reservoirs could be calculated by:

$$\eta_{\mathrm{I,\,max}} = 1 - \frac{T_{\mathrm{C}}}{T_{\mathrm{H}}} \tag{23}$$

Actual efficiency of the system can also be calculated utilizing the amounts of output net power and heat transfer rate of the evaporator as:

$$\eta_{\mathrm{I}} = \frac{P_{\mathrm{N}}}{Q_{\mathrm{E}}} \tag{24}$$

III. Sensitivity Analysis

Sensitivity analysis (SA) is the study of how the variation in the output of a statistical model can be attributed to different variations in the inputs of the model [12]. Put another way, it is a technique for systematically changing variables in a model to determine the effects of such changes.

In the present paper, the variation of net power and efficiency of a 100 MW OTEC power plant as system objectives is studied by the system parameters variation. The most important system parameters in an OTEC power plant design are input warm and cold water temperatures, warm and cold water pipe diameters, warm and cold water velocity in pipes, and evaporator and condenser working temperatures.

In order to achieve feasible results, appropriate constraints for parameters is needed. According to the study done by Najafi et al [9], reasonable constraints could be established as,

$$22 \leq T_{\mathrm{WSI}} \leq 28 \tag{25}$$

$$4 \leq T_{\mathrm{CSI}} \leq 8 \tag{26}$$

$$5 \leq D_{\mathrm{CS}}, D_{\mathrm{WS}} \leq 25 \tag{27}$$

$$0.7 \leq V_{\mathrm{CS}}, V_{\mathrm{WS}} \leq 2 \tag{28}$$

$$T_{\mathrm{WSI}} - 7 \leq T_{\mathrm{E}} \leq T_{\mathrm{WSI}} - 3 \tag{29}$$

$$T_{\mathrm{CSI}} + 3 \leq T_{\mathrm{C}} \leq T_{\mathrm{CSI}} + 7 \tag{30}$$

IV. Results and Discussion

To validate the model, the results from the simulations are compared with a case study from Uehara and Ikegami [8] in TABLE I. It can be observed that the results are in an acceptable agreement with different inlet warm sea water temperatures.

TABLE I
Validation

	T_{WSI} (°C)	P_{N} (MW)	P_{WS} (MW)	P_{CS} (MW)	P_{WF} (MW)
Ref. [8]	28	71.4	11.1	15.6	1.97
Present work	28	71.8	12.1	14.4	1.42

Results of sensitivity analysis are illustrated in figures 3 to 10. In each figure the variance of the net power and efficiency is depicted as the input parameters are changed. In Fig. 3 inlet warm water temperature is assumed as input parameter. It can be seen that by increasing the water temperature and fixing other parameters, both net power and efficiency will improve. However, net power will decrease as the input warm water temperature reaches to evaporator working temperature.

As expected, similar to previous results, net power will also decrease by increasing the inlet cold water temperature. This effect aggravates by reaching to condenser working temperature (Fig. 4).

Next parameters of sensitivity analysis are warm and cold water pipes diameters. Neglecting the economical and construction considerations, widening the pipes results in a larger inlet water mass flow rate which will raise the generated power as well as working pumps consuming power (Fig. 5 and Fig. 6). By enlarging pipe diameters, pumps powers will become massive enough to terminate increasing rate of net power. This is more vivid in cold water pipes due to longer pipe length. Moreover, it is notable that designing based on only higher output net power will not always bring about better performance, since pipe diameter selection plays an important role in plant efficiency. Efficiency diagram behaves as a parabola and has a maximum point at diameters about 15 meters. This result correspond data from Uehara and Ikegami [8].

By increasing the water velocity in pipes, efficiency rate has a descent behavior, while net output power increases at first then decreases in higher velocities. The net output power peak is located at about velocity of 1 meter per second which is near to the optimized results [9].

In figure 9 the effect of evaporator working temperature on net power and efficiency is illustrated. As it discussed earlier, net output power will decrease by increasing the evaporator working temperature toward the input warm water temperature. Hence, the best design is one that has an appropriate temperature difference between evaporator working temperature and input warm water temperature. From efficiency diagram, it can be observed that the suitable temperature is about $23.5°$ which has about $5.5°$ difference from input warm water temperature, that is similar to results of [6], [7].

V. Conclusion

A sensitivity analysis was carried out in order to find the variation in the output of a closed cycle OTEC plant by variations in the inputs of the model. Effect of different input parameters on the net output power and efficiency of a 100 MW power plant case study had been investigated.

It was observed that in locations with warmer surface water temperature, higher amounts of net output power and efficiency are achieved. It was also shown that widening the pipes in order to increase the mass flow rate would not improve the objectives necessarily and in some cases pumps consuming power would decrease the objectives. Each diagrams of pipe diameters and velocities contain a maximum efficiency point that corroborates the results in similar works in the literature. Furthermore, It was deduced that by approximating the input water temperatures to heat exchangers working temperatures, both objectives decrease simultaneously. Thus, an appropriate difference between these two temperatures is needed.

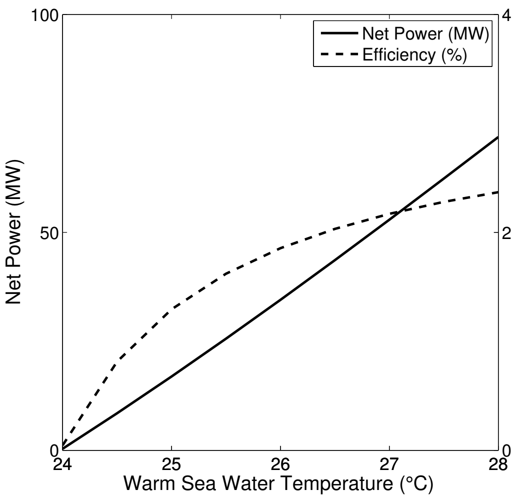

Fig. 3. Effect of inlet warm sea water temperature variation on objectives

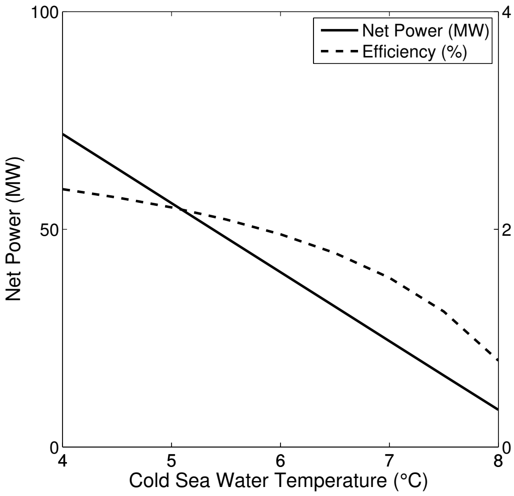

Fig. 4. Effect of inlet cold sea water temperature variation on objectives

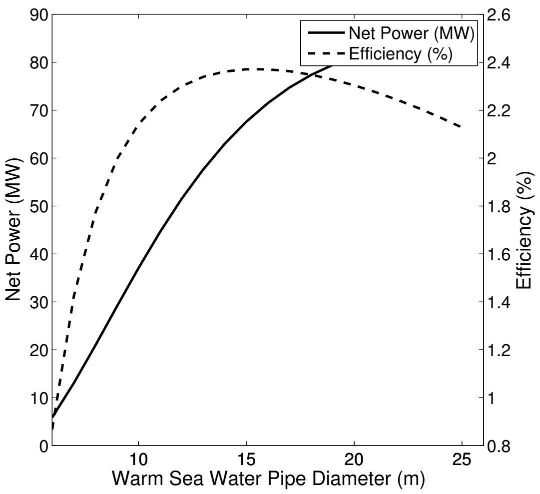

Fig. 5. Effect of warm sea water pipe diameter variation on objectives

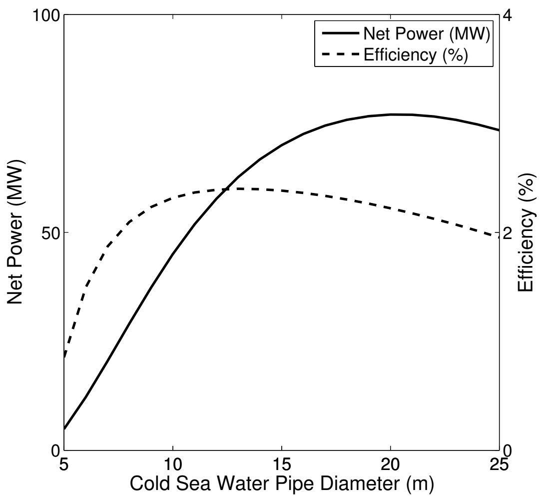

Fig. 6. Effect of cold sea water pipe diameter variation on objectives

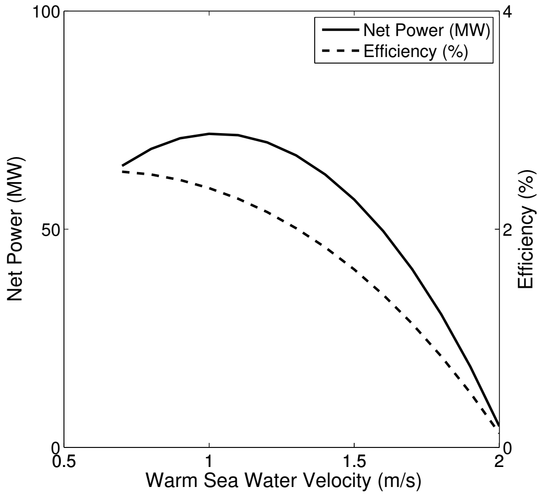

Fig. 7. Effect of warm sea water velocity variation on objectives

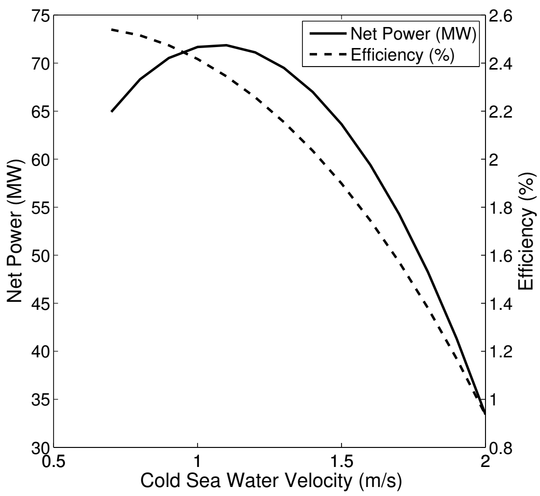

Fig. 8. Effect of cold sea water velocity variation on objectives

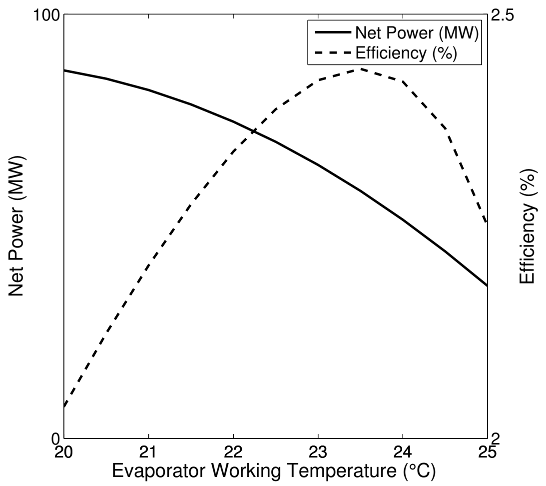

Fig. 9. Effect of evaporator working temperature on variation objectives

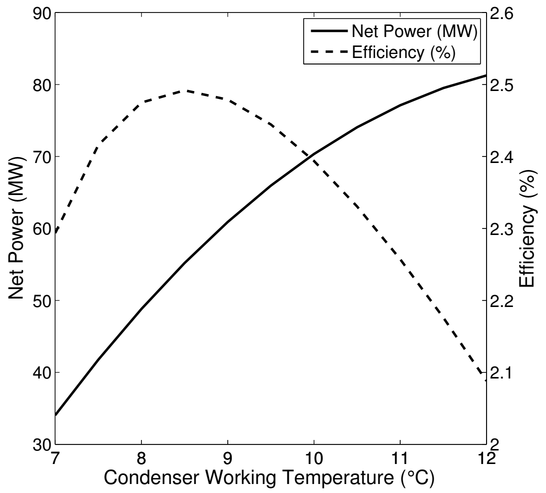

Fig. 10. Effect of condenser working temperature variation on objectives

REFERENCES

[1] W. H. Avery and C. Wu, *Renewable Energy from the Ocean: A Guide to OTEC*. Oxford University Press, 1994.

[2] C. Xie, S. Wang, L. Zhang, and S. J. Hu, "Improvement of proton exchange membrane fuel cell overall efficiency by integrating heat to electricity conversion," *Power Sources*, vol. 191, pp. 433–441, 2009.

[3] N. J. Kim, K. C. Ng, and W. Chun, "Using the condenser effluent from a nuclear power plant for ocean thermal energy conversion (otec)," *International Communications in Heat and Mass Transfer*, vol. 36, pp. 1008–1013, 2009.

[4] N. Yamada, A. Hoshi, and Y. Ikegami, "Performance simulation of solar boosted ocean thermal energy conversion plant," *Renewable Energy*, vol. 34, pp. 1752–1758, 2009.

[5] P. J. Straatman and W. G. van Sark, "A new hybrid ocean thermal energy conversion offshore solar pond (otec osp) design a cost optimization approach," *Solar Energy*, vol. 82, pp. 520–527, 2008.

[6] C. Wu, "Specific power optimization of closed cycle otec plants," *Ocean Engineering*, vol. 17, pp. 307–314, 1990.

[7] ——, "Specific power analysis of thermoelectric otec plants," *Ocean Engineering*, vol. 20, pp. 433–442, 1993.

[8] H. Uehara and Y. Ikegami, "Optimization of a closed-cycle otec system," *Solar Energy Engineering*, vol. 112, pp. 247–256, 1990.

[9] A. Najafi, S. Rezaee, and F. Torabi, "Multi-Objective optimization of ocean thermal energy conversion power plant via genetic algorithm," in *2011 IEEE Electrical Power and Energy Conference (IEEE EPEC 2011)"*, Winnipeg, Canada, oct 2011, pp. 41–46.

[10] R. H. Yeh, T. Z. Su, and M. S. Yang, "Maximum output of an otec power plant," *Ocean Engineering*, vol. 32, pp. 685–700, 2005.

[11] H. Uehara, H. Kusuda, M. Monde, T. Nakaoka, and H. Sumitomo, "Shell-and-plate-type heat exchangers for otec plants," *Solar Energy Enigeering*, vol. 106, pp. 286–290, 2011.

[12] A. Saltelli, M. Ratto, T. Andres, F. Campolongo, J. Cariboni, D. Gatelli, M. Saisana, and S. Tarantola, *Global Sensitivity Analysis: The Primer*. John Wiley & Sons, 2008.