Springer

Journal of Mechanical Science and Technology

Journal of Mechanical Science and Technology 22 (2008) 1977~1983

www.springerlink.com/content/1738-494x
DOI 10.1007/s12206-008-0742-9

Performance analysis and improvement for CC-OTEC system †

WANG Tong*, DING Liang, GU Chuangang and YANG Bo

School of Mechanical Engineering, Shanghai Jiao Tong University 800 Dongchuan Road, Shanghai, 200240, China

(Manuscript Received April 16, 2008; Revised June 11, 2008; Accepted July 23, 2008)

Abstract

Ocean Thermal Energy Conversion (OTEC) is a way to generate electricity by the temperature difference of seawater from the upper surface to different depths. Closed Cycle of OTEC (CC-OTEC) pumps the working fluid vapor to high pressure to propel the turbine-generator and produce electricity. However, low temperature difference of seawater is the largest constraint to the utilization compared to the conventional power plants. Solar energy reheated power cycle is proposed to improve the practicability. Instead of traditional efficiency, net cycle efficiency (NCE) is applied to assess the performance of CC-OTEC system. The factors that influence the performance, such as working fluid and corresponding evaporating pressure, superheating temperature and turbine outlet pressure, are discussed. Considered the engineering application, the appropriate net output power should be at least 50kw.

Keywords: CC-OTEC; Thermal efficiency; Performance; Solar energy

1. Introduction

The cycle of Ocean Thermal Energy Conversion (OTEC) is a technique that generates electricity by the temperature difference between surface seawater and deep seawater. In the ocean, the temperature of the water decreases with an increase in depth. It is generally considered that as long as the temperature difference between the warm ocean surface and the ocean deep exceeds 20℃, the OTEC system can be propelled to generate electricity. In the Closed Cycle of Ocean Thermal Energy Conversion (CC-OTEC) system, warm surface seawater and cold seawater are used to vaporize and condense a working fluid separately, such as anhydrous ammonia, which drives a turbine-generator in a closed loop producing electricity.

As early as in 1881, French physicist D'Arsonval proposed the idea of OTEC by ammonia as the working fluid, which was firstly demonstrated by Laude. A small OTEC plant (Mini-OTEC) mounted on a barge off Hawaii was built in 1979 by the state of Hawaii, which produced more than 50 kW of gross power, with a net output of up to 18 kW. Subsequently, U.S. and Japan have performed extensive studies on OTEC and built several research OTEC plants, with a growing power one by one.

In 1980 Taiwan power company of China planed to put the power of OTEC plant together with the residual heat from 3rd an 4th nuclear plant. Later in 1985 Guangzhou Institute of Energy Conversion of Chinese Academy of Science studied a Droplets Upgrade Cycle method used in OTEC system to improve seawater potential energy and built two Mini-OTEC experimental benches of 10W and 60W in 1991. Tanner (1995) came to a conclusion in his project that Taiwan represents the closet fit between technology and market with regards to OTEC as an energy source in the near future. Ignoring other factors, Wu C. (1998) chose the specific power as the objective function in the design of an OTEC Rankine power plant. Odum (2000) made an energy analysis of a land-based OTEC system by energy evaluation methods.

† This paper was presented at the 9th Asian International Conference on Fluid Machinery (AICFM9), Jeju, Korea, October 16-19, 2007.
*Corresponding author. Tel.: +86 21 3420 6576, Fax.: +86 21 6447 0256
E-mail address: twang@sjtu.edu.cn

Considering the parameters of pipe length, pipe diameter, seawater depth, and flow rate of seawater, Yeh R. H. (2005) studied the effects of the temperature and flow rate of cold seawater on the net output of an OTEC plant. Due to the low temperature difference, the power generated by the turbine is not sufficient to support persistent operation of OTEC system. Gérard (2007) proposed 1D model for temperature distribution to get that worldwide power sources which could be extracted from the operation of OTEC plants might be limited. In order to increase the feasibility of OTEC system, the other kinds energy should be introduced to raise the temperature difference, such as solar energy, geothermal heat and some industrial exhaust heat, etc. In the districts with rich ocean thermal energy, there is also abundant solar energy. It can be as the additional heat source to enlarge the temperature difference between heat sources. As for CC-OTEC system, the power cycle is improved as solar energy reheated power cycle.

2. Solar energy reheated power cycle

A sketch of CC-OTEC system with solar collector is shown in Fig. 1. Reheated by solar energy, the working fluid at state point 1 is at higher pressure and superheating status compared with the original system. Superheated vapor expands in the turbine and propels the turbine to rotate, and then the turbine can drive the motor to generate electricity. Flowing out of the turbine, the working fluid gets to state point 2′ with inevitable losses in expanding process. Cooled by the cold seawater in condenser, the working fluid gets to state point 3′ with flowing loss in condensation process and under supercooled liquid status. Then it is pumped to higher pressure state point 4′. Preheated by the warm seawater and reheated by solar energy, it passes state point 5′ and returns to state point 1. The status of working fluid varies process in p-h diagram is shown in Fig. 2 by the solid line 12′3′4′5′1 correspondingly. The ideal process without any losses is represented by the dashed line 123451. During the ideal process, the working fluid at state point 1 expands adiabatically in turbine to propel it and gets to state point 2, and the heat exchanging processes are at constant pressure.

In the designed circulation, the temperature difference by heat exchange and pressure losses must be considered. At state point 4′ the working fluid is normally preheated to 20~25℃ by the surface warm water, at state point 3′ it is condensed to 8~12℃ by the cold seawater. Generally, the ordinary solar collectors can at least raise the temperature of thermal source to 70℃. Compared with the temperature of ocean surface, the pressure and temperature of working fluid can be greatly increased by the solar collector, i.e. the temperature difference between the cold and thermal source is enlarged.

Vega (2002) suggested that the parameters at different status in CC-OTEC system could be expressed as followings. With the internal and mechanical loss in turbine specific enthalpy at state point 2′ is computed as

$$h_2' = h_1 - (h_1 - h_2)\eta_t \qquad (1)$$

Considering the head of cold seawater pump, actual pressure at state point 3′ can be expressed as

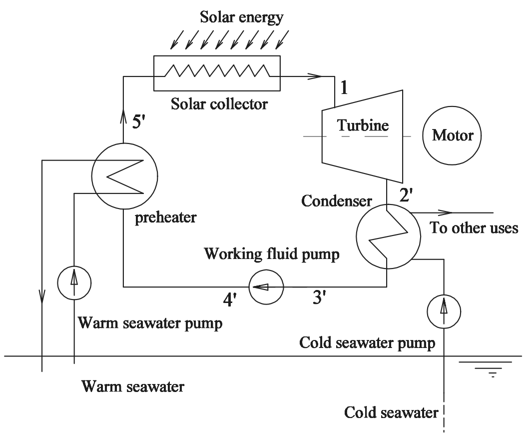

Fig. 1. Sketch of CC-OTEC system with solar collector.

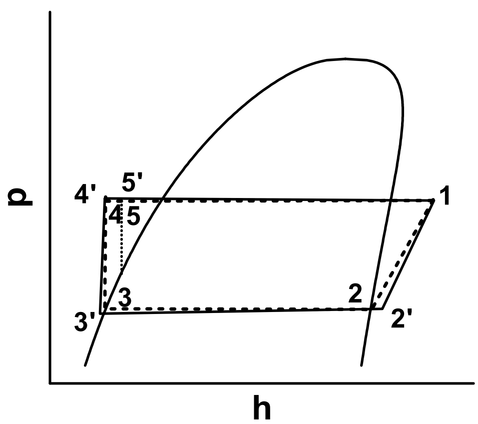

Fig. 2. p-h diagram of CC-OTEC system.

$$p_3' = p_2 - \Delta p_{cond} \tag{2}$$

Correspondingly, pressure at state point 4′ is expressed as

$$p_4' = p_1 + \Delta p_{preh} + \Delta p_{evap} \tag{3}$$

Where, Δp_{preh} is the pressure loss of working fluid in preheater by the ocean surface water, Δp_{evap} is the pressure loss of working fluid in solar collector. The consumed work of working fluid pump is:

$$W_{fp} = \frac{(p_4' - p_3')v_3' M_f}{\eta_{fp}} \tag{4}$$

Actual enthalpy at state point 4′:

$$h_4' = h_3' + \frac{W_{fp}}{M_f} \tag{5}$$

Mass of cold seawater can be get by heat exchange balance,

$$M_c = \frac{(h_2' - h_3')M_f}{Cp_c \Delta t_c} \tag{6}$$

In CC-OTEC system, long adiabatic ducts must be fixed to transfer the cold seawater from the ocean deep. There exists local frictional resistance and on way frictional resistance. Yeh (2005) computed the work of cold seawater pump by the method. For simplicity only head of cold seawater pump L is provided here to get the work:

$$W_{cp} = \frac{M_c g L}{\eta_{cp}} \tag{7}$$

Compared with that of warm seawater to preheater, W_{cp} is much greater. So only W_{cp} is considered here. In the solar collectors the absorption heat is

$$Q = (h_1 - h_5')M_f \tag{8}$$

Turbine output work is always expressed as

$$W_t = (h_1 - h_2')M_f \tag{9}$$

Then the net power of the cycle is

$$W_n = W_t - W_{fp} - W_{cp} \tag{10}$$

Vega (2005) expressed the traditional thermal efficiency as

$$\eta_{tr} = \frac{W_n}{(h_1 - h_4')M_f} \tag{11}$$

3. Working fluid selection

Once the solar collector is introduced, the temperature of heat source can be at least increased to 70℃, the temperature of working fluid in CC-OTEC varies between 10 ~ 65℃, which is near the normal atmospheric temperature. The low boiling point working fluids can be applied. The most commonly used working fluids in CC-OTEC are ammonia, Freon, water and a few kinds of hydrocarbon. Several aspects should be considered under the selection process of working fluid, such as the characteristic, flowing losses, heat exchange, stability, corrosivity, safety and feasibility of lab and engineering simulations. Taking into account these factors, we propose the following principle for the system.

① Considered the environmental protection problem, non-cfc materials whose ODP=0 is demanded. It should have low GWP value, narrow two-phase zone and proper pressure and temperature intervals, low viscosity, big thermal conductivity, stable chemical property. Also, it should be non-poisonous and non-combustible.

② Considering from the engineering feasibility, we demand the low seawater amount to guarantee proper seawater pipe diameter and length.

③ Considering from the equipment design, we demand low evaporation pressure to reduce the intensity requirement of turbine, evaporator and venting pipe.

Of course, selecting a working fluid to satisfy all the requirements above, it's actually a little bit difficult. According to the above principles and former experience, the performances of NH_3, R22, R11, R502, R12 are compared. Provided that the system conditions are as the followings: condensation temperature is 10℃, evaporation temperature is 40℃. Pressure loss of condenser, preheater, solar collector is 0.05bar, 0.05bar, 0.03bar, respectively. Head of cold seawater pump is 30m. Efficiency of turbine, seawater pump, and working fluid pump is 0.63, 0.72 and 0.72, respectively.

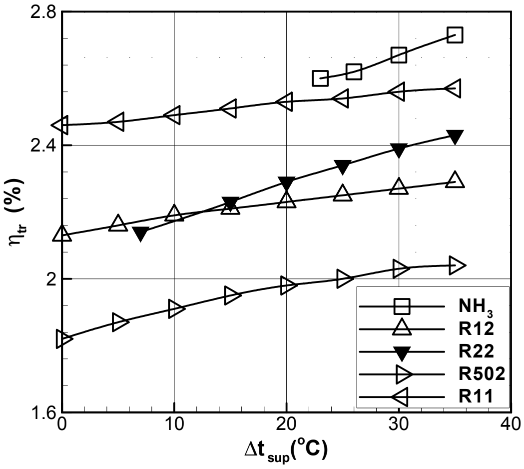

Fig. 3. η_{tr} of different working fluid.

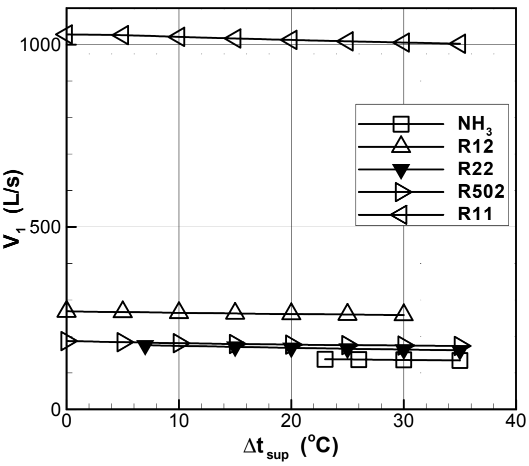

Fig. 4. Volume flow rate at state point 1 of different working fluid.

Different working fluid cycle performances are plotted in Fig. 3 to Fig. 5. Abscissa denotes Δt_{sup}, i.e. superheating temperature. Ordinate denotes traditional thermal efficiency η_{tr}, and volume flow rate of working fluid at state point 1 V_1. η_{tr} is a direct criterion when choosing the working fluid; V_1 determines the diameter of seawater pipe. All of parameters are important factors during the process of selecting the working fluid.

As shown in the figures, ammonia has a pretty good quality. Under the same condition, compared with the others, ammonia has higher cycle efficiency and lower absorbed energy from solar collectors. No doubt that ammonia is the best choice just according to cycle performance.

However, ammonia must guarantee some degrees of superheating temperature, otherwise at state point 2 it can easily enter into two-phase zone. It can be found from the figures that ammonia only has the data over 20 ℃ of superheating temperature. It results in high demand for solar collector and increases the investment cost. Besides, ammonia is easily dissolved in water and has certain toxicity, not satisfied with principal ①. Once it is leaked out, it will have a bad effect on halobios. Consequently ammonia is not applied here.

R11 relatively has a fine performance, but with a nonzero number of ODP, it is not satisfied with principal ①, either. As specific volume of R11 is much more, from Fig. 4 the inlet volume flow rate of turbine is greater than the others, which results in a very large component requirement.

As a result, R22 is selected as the working fluid, which has a satisfying quality according to parameters and nearly satisfies all the three principals presented above.

4. Cycle efficiency discussion

Compared with the traditional steam power cycle, there is no additional mineral resources consumption in the solar energy reheated power cycle. Whatever the ocean thermal energy or the solar energy in the CC-OTEC system, both of them are primary energy source, which are abundant, clean and free. Accordingly it's not so proper to judge the performance of CC-OTEC system by the traditional thermal efficiency η_{tr}.

Based on the definition of cycle efficiency, net cycle efficiency (NCE) should be defined as the following expression.

$$\text{NCE} = \frac{W_n}{W_t} \tag{12}$$

Supposed all of the system conditions are the same as above, NCE of different working fluid cycle is plotted in Fig. 5. It nearly has the same tendency as η_{tr} in Fig. 3, but the absolute data is different. The system can't be applied in the industry because of the too low efficiency. Once NCE is used as the evaluating indicator, it has the same magnitude as that of the normal steam power cycle. So NCE is more suitable than η_{tr} as the evaluating indicator for performance of CC-OTEC system.

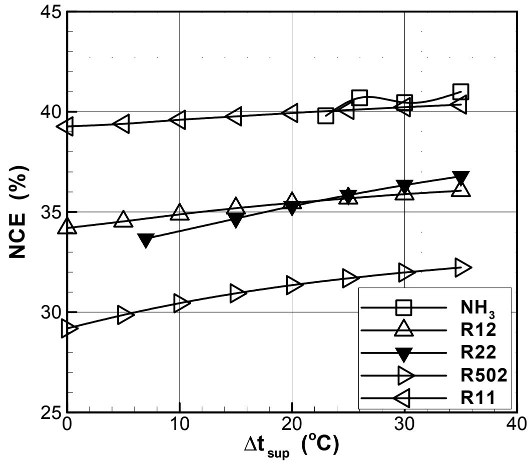

Fig. 5. NCE of different working fluid.

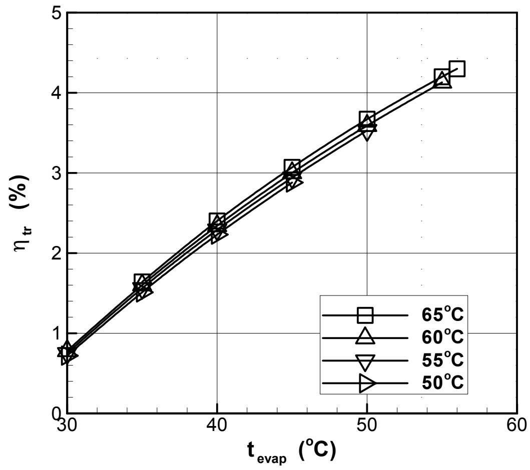

Fig. 6. η_{tr} of different turbine inlet temperature.

There are three methods to improve thermal efficiency for the traditional steam power cycle: increase of inlet temperature or pressure of turbine and reduction of the outlet pressure of turbine. In terms of the results of calculation, these three methods can also raise NCE.

The effect of increasing the inlet pressure of turbine is equivalent to that of the addition of evaporation temperature. Assumed that system conditions are the same as the former discussed and R22 is taken as the working fluid, the efficiencies under different turbine inlet temperature are shown in Fig 5-6, where 65℃, 60℃, 55℃, 50℃ represents the varying inlet temperature of turbine. The abscissa denotes evaporation temperature; the ordinate denotes η_{tr} and NCE, respectively.

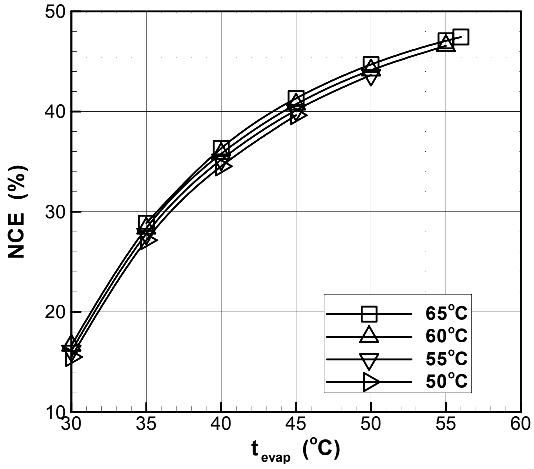

Fig. 7. NCE of different turbine inlet temperature.

Compared with them in Fig. 6 and Fig. 7, the influence of increment rate of evaporation temperature on the efficiency is much greater than that of superheated temperature. This is to say, increment of evaporation temperature is more effective than that of superheated temperature in increasing the CC-OTEC efficiency. For the subject investigated in our problem, the outlet pressure is restricted by the temperature of cold seawater, which can be considered as constant. Consequently, under the compromise of no additional load of exchangers and assurance of appropriate working fluid's quality, increasing the turbine inlet pressure, i.e. evaporation temperature, is the most effective way to improve OTEC performance. Besides, there is a very important message in Fig. 6 and Fig. 7. η_{tr} increases with t_{evap} linearly, while NCE does not. It signifies that it is unnecessary to add the evaporation temperature to improve the performance of CC-OTEC system.

Therefore, NCE is more suitable for assessing the performance of the system.

5. Engineering example

According to all the relations obtained above, the CC-OTEC design conditions are provided in Table 1 and the status at each state point is displayed in Table 2. The parameters of flow rate, pumps' work and efficiencies are calculated and demonstrated in Table 3.

Instead of η_{tr}, NCE is comparative to the efficiency of traditional power plant. Consequently the CC-OTEC system with solar energy reheated has the great market in practical application.

Table 1. Design conditions for CC-OTEC system.

working fluid	R22
t_{evap}(℃)	55
t_{cond}(℃)	10
$\triangle t_{sup}$(℃)	10
$\triangle t_{oc}$(℃)	2
$\triangle t_{c}$(℃)	3
$\triangle t_{w}$(℃)	2
$\triangle t_{sol}$(℃)	5
$\triangle p_{cond}$(bar)	0.05
$\triangle p_{preh}$(bar)	0.05
$\triangle p_{evap}$(bar)	0.03
η_{fp}	0.72
η_{cp}	0.72
η_{t}	0.63
L (m)	30
W_n(kw)	50

Table 2. CC-OTEC system status at each state point.

	temp. (℃)	pressure (bar)	specific volume (m^3/kg)	specific enthalpy (kJ/kg)
1	65	21.7	0.0112	628.2
2	11.24	6.8	0.035	609.1
3	8	6.75	0.0008	409.4
4	8	21.8	0.0008	410.9
5	25	21.77	0.0008	423.7

Table 3. CC-OTEC system parameters.

η_{tr} (%)	4.12
NCE (%)	46.7
V_1 (m^3/s)	0.0624
V_2 (m^3/s)	0.1952
Q_{sol} (kw)	1141.9
W_{fp} (kw)	9.3
W_{cp} (kw)	37
W_t (kw)	96.4

Table 4. Size of primary components.

parameters		turbine	medium pump	seawater pump
inlet	volume flow rate (m^3/s)	0.0624	0.0045	0.0888
	diameter of duct (mm)	63	61.8	275
outlet	volume flow rate (m^3/s)	0.1952	0.0045	0.0888
	diameter of duct (mm)	111.5	61.8	275

According to flow rate of working fluid, the parameters of cold seawater flow rate and the pipe diameter of turbine are demonstrated in Table 4. As the volume flow rate is small, the turbine employed here is not common industry equipment for generating electricity and should be specially designed. As the results of 50kw net power, the size of the equipments and width of pipes just fit the conditions above. If the net power reduced, the data of efficiencies, such as η_{fp}, η_{cp}, η_t , would reduce greatly, and the equipments would not be found as products in the market.

6. Conclusion

With the development of industry and economy, the energy became the global problem. Just as other marine energies, OTEC is a clean way to generate electricity. It does not consume primary energy sources, and is not subject to the depletion of energy.

- According to the principals presented above, R22 is the more suitable choice as the working fluid.
- Solar collector can be introduced into the OTEC system, which will greatly increase the temperature difference and therefore improve the cycle performance and practical application.
- Net cycle efficiency (NCE) is more proper to assess the cycle performance of CC-OTEC system.
- Under the compromise of no additional load of exchangers, increase of the turbine inlet pressure is the most effective way to improve the system performance.
- For the industrial application, the appropriate net output power of the system should be at least 50kw.

Acknowledgments

It was supported by the Project of Department of Chinese Science and Technology.

Nomenclature

Cp	:	Specific heat, kJ/(kg·K)
g	:	Acceleration of gravity, m/s^2
h	:	Ideal specific enthalpy, kJ/kg
h'	:	Actual specific enthalpy, kJ/kg
L	:	Head of cold seawater pump, m
M	:	Mass of flow rate, kg/s
NCE	:	Net cycle efficiency
p	:	Ideal pressure, bar
p'	:	Actual pressure, bar
Q	:	Exchanging heat, kw
t	:	Temperature, ℃

v'	:	Actual specific volume, m^3/kg
V	:	Volume flow rate, m^3/s
W	:	Power , kw
W'	:	Net power, kw

Greek Letters

$\triangle p$	:	Pressure loss, bar
$\triangle t$	:	Temperature difference, ℃
η	:	Efficiency

Subscripts

1,2,3,4,5	:	Status's point designation
c	:	Cold seawater
cond	:	Condenser or condensation
evap	:	Evaporator or evaporation
f	:	Working fluid
n	:	Net amount
p	:	Pump
preh	:	Preheater
sup	:	Superheating
t	:	Turbine
tr	:	Traditional
w	:	Warm seawater

References

[1] G. C. Nihous, A Preliminary assessment of ocean thermal energy conversion resources, *Journal of Energy Resources Technology*, 129(1) (2007) 10-17.

[2] H. T. Odum, Emergy evaluation of an OTEC electrical power system, *Energy*, 25 (2000) 389-393.

[3] D. Tanner, Ocean thermal energy conversion: current overview and future outlook, *Renewable Energy*, 6(3) (1995) 367-373.

[4] L. A. Vega, Ocean thermal energy conversion primer, *Marine Technology Society Journal*, 36(4) (2002) 25-35.

[5] C. Wu and T. J. Burke, Intelligent computer aided optimization on specific power of an OTEC Rankine power plant, *Applied Thermal Engineering*, 18(5) (1998) 295-300.

[6] R. H. Yeh, T. Z. Su and M. S. Yang, Maximum output of an OTEC power plant, *Ocean Engineering*, 32(5-6) (2005) 685-700.