Dynamic Model for Ocean Thermal Energy Conversion Plant with Working Fluid of Binary Mixtures

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Abstract: Ocean thermal energy conversion (OTEC) is an effective method of power generation, which has a small impact on the environment and can be utilized semi-permanently. This paper describes a dynamic model for a pilot OTEC plant built by the Institute of Ocean Energy, Saga University, Japan. This plant is based on Uehara cycle, in which binary mixtures of ammonia and water is used as the working fluid. Some simulation results attained by this model and the analysis of the results are presented. The developed computer simulation can be used to actual practice effectively, such as stable control in a steady operation, optimal determination of the plant specifications for a higher thermal efficiency and evaluation of the economic prospects and off-line training for the operators of OTEC plant.

Keywords: Electric power generation, Ocean Thermal Energy Conversion (OTEC), dynamic model, binary mixtures

1. INTRODUCTION

The problem of worldwide energy resource and environmental crisis is attracting more and more attentions. It is very urgent for our human being to explore new energy resources, especially renewable and environment-minded ones. The ocean thermal energy, which is stored in the temperature difference between warm surface seawater (about 25 to 30 degrees centigrade) and cold deep seawater (around 5 degrees centigrade at the depth of 500-1000 meters), is enormous in its volume and at the same time is quit stable, so it is a kind of renewable resource. Ocean Thermal Energy Conversion (OTEC) is the technology with which we can convert this thermal energy into electric power [1]. OTEC is an effective method of power generation, which has small impact on the environment and can be utilized semi-permanently. However, because of the small temperature difference between surface seawater and deep seawater, the thermal efficiency of the OTEC system (about 3% to 4%) is much lower than that of the thermal power generation plant using fossils or nuclear energy (about 50% to 60%). In order to improve the thermal efficiency of OTEC plant with a low plant construction cost, a new power cycle using ammonia and water mixture as working fluid has been developed in our recent study. And the thermal efficiency of the new power cycle was demonstrated to be higher than the conventional ones experimentally. In order to get satisfied efficiency and make the OTEC plant stable, a comprehensive understanding of the characteristics of the plant is required to achieve steady operation and control. Therefore, the development of dynamic model for the transient performance of the new power cycle using binary mixtures becomes necessary because of the complexity of the cycle structure and the different physical properties of the binary mixtures. This paper concerns with the dynamic model construction and simulation for OTEC plant with binary mixtures of ammonia and water as the working fluid, in which the temperature, pressure, mass and composition of the working fluid in each component are all time dependent. And then a simulation work was implemented using the developed model.

2. POWER CYCLE OF OTEC PLANT

There are three different types of OTEC technology, open cycle, closed cycle and hybrid cycle, where the third type is the combined technology of the first two types [1]. And it has been known that the closed cycle is more economical and also better fit for generation of larger output than the open cycle.

In the closed-cycle plant, warm surface seawater is used to evaporate an enclosed auxiliary working fluid such as ammonia or Freon. The vaporized working fluid drives a turbine which drives an electrical generator. Cold deep seawater is used to condense the vapor from the outlet of the turbine. This cycle, which is the same as the cycle in the steam power plant, is named as Rankine cycle. Since the temperature of the warm water is very low, the boiling point of the working fluid should also be very low. In order to increase the thermal efficiency, Dr. A. A. Kalina developed a new power cycle using working fluid of binary mixtures instead of a pure substance [2]. However, because of the use of binary mixtures as working fluid, the heat-transfer performance of the heat exchanger decreases in Kalina cycle [3]. Then, the construction cost of the plant will increase if a larger heat-transfer area is required. Dr. H. Uehara improved Kalina cycle by using the extraction process to lessen the load of condenser [4]. The improved cycle is named as Uehara cycle, which is utilized in our pilot OTEC plant.

The diagram of Uehara cycle is illustrated in Fig. 1 [4-5]. The binary mixtures of ammonia and water are used as the working fluid. The working fluid circulation pump 2 sends the working fluid to the regenerator, where the cool working fluid is warmed by the warm working fluid through heat exchanger. Then the working fluid enters the evaporator, where a part of it is vaporized by the warm water and enters the separator, where it is separated into saturation vapor and saturation liquid. The vapor enters the turbine 1. A part of the vapor is extracted and enters the heater. Another part of vapor enters the turbine 2. The saturation liquid is cooled by the cool working fluid through the regenerator and passes the diffuser. The outlet vapor from turbine 2 and the liquid from the diffuser are mixed in the absorber, and at the same time the liquid absorbs some of the vapor. The mixed vapor and liquid enter the condenser, where they are cooled by cool water. The working fluid circulation pump 1 sends the condensed working fluid to the heater, where the working fluid is warmed by the extracted vapor from turbine 1. The working fluid is returned to the regenerator and the cycle continuous.

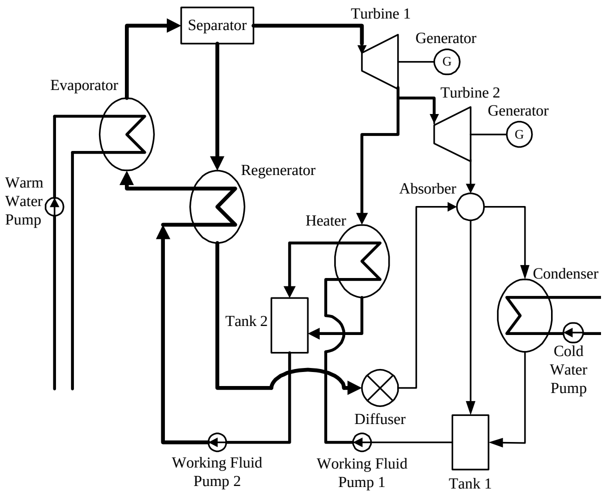

Fig. 1 Diagram of Uehara cycle

3. MODEL CONSTRUCTION FOR OTEC PLANT

3.1 Basic assumption

In order to simplify the mathematical representation of the plant, the models are developed under the following basic assumptions.

a) The models are constructed in lumped parameter forms, and the lumped parameters are represented by the corresponding parameters at the outlet.

b) The whole system is divided into six major components: (1) regenerator, (2) evaporator and separator, (3) absorber, condenser and tank 1, (4) heater and tank 2, (5) turbine, and (6) diffuser.

c) The whole power cycle is divided into high-pressure, middle-pressure and low-pressure subsystems, where the pressure difference in each subsystem is neglected. The high-pressure subsystem is from the outlet of pump 2 to the inlets of turbine 1 and diffuser, which is shown by the thick line in Fig. 1. The middle-pressure subsystem is from the outlets of turbine 1 and pump 1 to the inlet of pump 2. The low-pressure subsystem is from the outlets of turbine 2 and diffuser to the inlet of pump 1.

3.2 Description of models for major components

Model of evaporator and separator. The model for the evaporator was constructed by merging the evaporator and the separator into a unit as one component. The model structure is shown in Fig. 2. [5]

The working fluid is assumed in the liquid phase from the inlet of the evaporator to the liquid surface of the separator. Therefore, the heat transfer coefficient from the warm water to the working fluid keeps the same along the area of the heat exchanger, where the effect of overheat is neglected. Further, it is assumed that the separator separates the liquid and vapor completely.

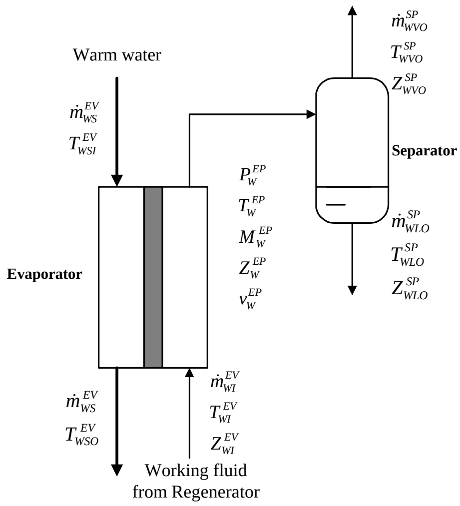

Fig. 2 Model structure of evaporator and separator

The equation of energy conservation law (the first law of thermodynamics) is written as

$$\frac{d}{dt}\left(M_W^{EP} h_W^{EP}\right) - V^{EP}\frac{d}{dt}P_W^{EP}$$
$$= \dot{m}_{WI}^{EV} h_{WI}^{EV} + Q^{EV} - \dot{m}_{WVO}^{SP} h_{WVO}^{SP} - \dot{m}_{WLO}^{SP} h_{WLO}^{SP} \qquad (1)$$

The heat transfer rate Q^{EV} between the warm water and the working fluid is calculated by [6]

$$Q^{EV} = U^{EV} A^{EV} \Delta T_m^{EV} \qquad (2)$$

where the logarithmic mean temperature difference ΔT_m^{EV} is

$$\Delta T_m^{EV} = \frac{(T_{WSI}^{EV} - T_W^{EP}) - (T_{WSO}^{EV} - T_{WI}^{EV})}{\ln \dfrac{T_{WSI}^{EV} - T_W^{EP}}{T_{WSO}^{EV} - T_{WI}^{EV}}} \qquad (3)$$

And the heat transfer coefficient U^{EV} is approximated by,

$$U^{EV} = U_s^{EV} (\dot{m}_{WS}^{EV} / \dot{m}_s^{EV})^{0.5} \qquad (4)$$

where U_s^{EV} and $\dot{m}_s^{EV}$ are the standard heat transfer coefficient and the standard mass flow rate respectively, which are determined from the experimental study.

The mass conservation law is written as

$$\frac{d}{dt}(M_W^{EP}) = \dot{m}_{WI}^{EV} - \dot{m}_{WVO}^{SP} - \dot{m}_{WLO}^{SP} \qquad (5)$$

Since binary mixtures of ammonia and water are used as the working fluid, the composition of the working fluid is time dependent, which is calculated by the equation of the ammonia mass conservation,

$$\frac{d}{dt}(M_W^{EP} Z_W^{EP}) = \dot{m}_{WI}^{EV} Z_{WI}^{EV} - \dot{m}_{WVO}^{SP} Z_{WVO}^{SP} - \dot{m}_{WLO}^{SP} Z_{WLO}^{SP} \quad (6)$$

The total volume of the evaporator and separator is a constant, so we have

$$\frac{d}{dt}(M_W^{EP} v_W^{EP}) = 0 \quad (7)$$

The property relationships of the binary mixtures are more complex than pure substance. There is one more independent intensive property in the system of binary mixture [7]. The enthalpy h and specific volume v are the functions of temperature T, pressure P and composition Z as

$$\begin{cases} h = f_1(T, P, Z) \\ v = f_2(T, P, Z) \end{cases} \quad (8)$$

In the saturation phase, the vapor and liquid compositions (Z_V, Z_L), enthalpies (h_V, h_L) and specific volumes (v_V, v_L) of the working fluid are related to the temperature and pressure as

$$\begin{cases} (Z_V, Z_L) = f_3(T, P) \\ (h_V, h_L) = f_4(T, P) \\ (v_V, v_L) = f_5(T, P) \end{cases} \quad (9)$$

In current model, the properties of the working fluid were calculated using a developed computational program of PROPATH (A PROgram PAckage for THermo physical properties of fluids), which supplies the property values of the binary mixtures of ammonia and water [8].

The model for absorber, condenser and tank1 and the model for heater and tank2, which are omitted in this paper, are similar to the model for evaporator and separator.

Model of regenerator. As the mass of the working fluid in the regenerator is comparatively less than that in the evaporator & separator, absorber, condenser & tank1, and heater & tank2, a static model for the heat exchanger was used to construct the regenerator model. The model structure of the regenerator is illustrated in Fig. 3. The residues on the heat exchanger wall are neglected, and the working fluids on the both sides of the heat exchanger are considered in the liquid state.

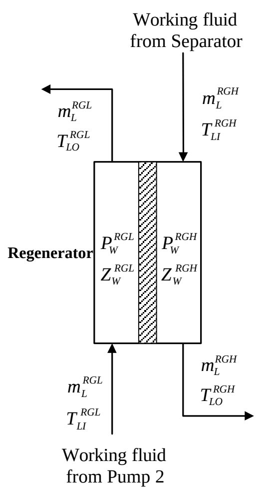

Fig. 3 Model structure of regenerator

The outlet temperatures of the regenerator T_{LO}^{RGL}, T_{LO}^{RGH} are calculated by [9],

$$T_{LO}^{RGL} = \frac{mT_{LI}^{RGL} W_L^{RGL} + (1 - e^{-U^{RG} m A^{RG}}) T_{LI}^{RGH}}{mW_L^{RGL} + (1 - e^{-U^{RG} m A^{RG}})} \quad (10)$$

$$T_{LO}^{RGH} = \frac{mW_L^{RGH} e^{-U^{RG} m A^{RG}} T_{LI}^{RGH} + (1 - e^{-U^{RG} m A^{RG}}) T_{LI}^{RGL}}{mW_L^{RGH} + (1 - e^{-U^{RG} m A^{RG}})(W_L^{RGH} / W_L^{RGL})} \quad (11)$$

where A^{RG} is the heat transfer area, and $m = 1/W_L^{RGH} - 1/W_L^{RGL}$, and the water equivalent W_L^{RGL}, W_L^{RGH} are given by

$$\begin{cases} W_L^{RGH} = \dot{m}_L^{RGH} c_W^{RGH} \\ W_L^{RGL} = \dot{m}_L^{RGL} c_W^{RGL} \end{cases}$$

where the specific heats (c_W^{RGH} and c_W^{RGL}) are considered as constants.

Model of turbine. The model construction of the turbine is shown in Fig. 4 [5]. Under the same consideration for regenerator, a thermostatic model for the turbine is used. Further, it is assumed that the degree of reaction equals zero, i.e., there is no enthalpy drop in the rotor blade of the turbine.

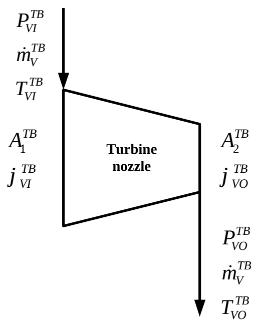

Fig. 4 Model structure of turbine

The isentropic enthalpy difference between the inlet and outlet of the turbine is given by [10]

$$h_{VI}^{TB} - h_{VOS}^{TB} = P_{VI}^{TB} v_{VI}^{TB} \frac{K_0}{K_0 - 1} \left[1 - \left(\frac{P_{VO}^{TB}}{P_{VI}^{TB}} \right)^{\frac{K_0 - 1}{K_0}} \right] \quad (12)$$

And the actual enthalpy difference is

$$h_{VI}^{TB} - h_{VO}^{TB} = h_{oi}^{TB} \left(h_{VI}^{TB} - h_{VOS}^{TB} \right) \quad (13)$$

where h_{oi}^{TB} is the relative internal efficiency of the turbine.

The velocity of the working fluid at the outlet of the nozzle j_{VO}^{TB} is calculated by,

$$h_{nz}^{TB} = \frac{(j_{VO}^{TB})^2}{2(h_{VI}^{TB} - h_{VOS}^{TB})} \quad (14)$$

where h_{nz}^{TB} is the efficiency of the turbine nozzle. Then mass flow rate through the turbine $\dot{m}^{TB}$ is given by,

$$\dot{m}^{TB} = A_2^{TB} j_{VO}^{TB} / v_{VO}^{TB} \quad (15)$$

where A_2^{TB} is the cross-area of the outlet of nozzle.

The power rate generated from turbine is written as

$$G^{TB} = \dot{m}^{TB} (h_{VI}^{TB} - h_{VO}^{TB}) \quad (16)$$

4. SIMULATION STUDY AND DISCUSSION

The mass flow rate of the warm water at inlet of evaporator is among the key parameters. The change of it affects the other parameters remarkably. Step test on this parameter was performed. The result of the step test on warm water flow rate is illustrated in Fig. 5.

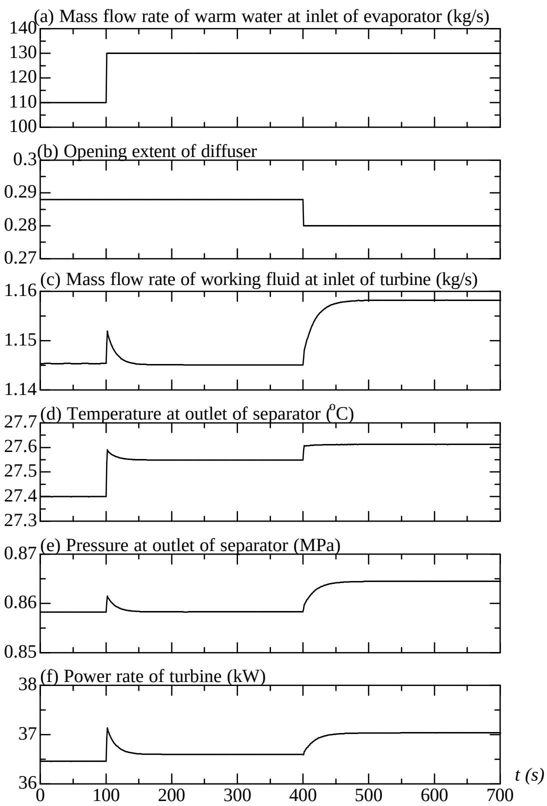

Fig. 5 Simulation result

When the mass flow rate of the warm water at inlet of evaporator ((a) in Fig. 5) increased suddenly at the time of 100 second, the mass flow rate of the working fluid at inlet of turbine ((c) in Fig. 5), the temperature ((d) in Fig. 5) and pressure ((e) in Fig. 5) of the working fluid at the outlet of separator increased at the same time. So the power rate of turbine ((f) in Fig. 5) increased also. However, these parameters decreased subsequently after reaching the peak. The steady state values of (c) and (e) returned to the initial values of (c) and (e) respectively.

The following is the analysis of the reason of this phenomenon. If the working fluid gets more heat from the warm water in the evaporator, more vapors will be generated, so the flow rate through turbine should increase to generate more power. However, the increasing of the pressure in separator causes the increasing of the mass flow rate through the diffuser, which can be calculated by

$$\dot{m}^{DI} = ka^{DI}\sqrt{\frac{P_W^{SP} - P_W^{AB}}{v_{WLO}^{SP}}} \tag{17}$$

where a^{DI} is the opening extent of the diffuser, and k is a coefficient[5]. So the mass flow rate through turbine decreases on the contrary. And major part of the increased energy flowed through the diffuser to the absorber. So only a small part of the increased energy was translated into power. This is not what we expect. In order to get more power, we should decrease the opening extent of the diffuser to an appropriate value.

The opening extent of the diffuser ((b) in Fig. 5) was decreased at the time of 400 second. Then (c), (d), (e) and (f) rose again, and finally the power rate of turbine reached the desire value.

Through the abovementioned result, we can conclude that if we want to increase the output power rate, adjusting the mass flow rate of warm water only is not enough, we should change the opening extent of the diffuser simultaneously. And from the simulation study, we can know how much we should change and how to change the manipulative parameters. That is to say, we can get the information for the controller design from the simulation study.

The developed computer simulation can be used to many other actual practices effectively, such as stable control in a steady operation, optimal determination of the plant specifications for a higher thermal efficiency, evaluation of the economic prospects and off-line training for the operators of OTEC plant.

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NOMENCLATURE

Normal		
A	[m^2]	Area
c	[J/kg•^{o}C]	Specific heat
G	[w]	Power rate
h	[J/kg]	Specific enthalpy
k	[m^2]	Coefficient
K_0		Heat insulation index
$\dot{m}$	[Kg/s]	Mass flow rate
M	[kg]	Mass
P	[MPa]	Pressure
Q	[w]	Heat transfer rate
T	[^{o}C]	Temperature
U	[w/m^2•^{o}C]	Heat-transfer coefficient
v	[m^3/kg]	Specific volume
V	[m^3]	Volume
W	[J/kg]	Water equivalent
Z	[kg/kg]	Composition
Greece		
a		Opening extent
h	[%]	Efficiency of turbine
j	[m/s]	Velocity
Superscript		
AB	Absorber	
DI	Diffuser	
EP	Evaporator & separator	
EV	Evaporator	
H	High temperature side	
L	Low temperature side	
RG	Regenerator	
SP	Separator	
TB	Turbine	
Subscript		
I	Inlet	
L	Liquid	
Nz	Nozzle	
O	Outlet	
Oi	Relative internal	
OS	Isentropic	
S	Standard(Experimental value)	
V	Vapor	
W	Working fluid	
WS	Warm water	