Construction of a State Space Model for an OTEC Plant Using Rankine Cycle with Heat Flow Rate Dynamics*

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Abstract: In this research, a state space model for an ocean thermal energy conversion (OTEC) plant using Rankine cycle is proposed by considering the heat transfer dynamics. The model is constructed by using an existing simple dynamic model. The temperatures and heat flow rates of warm and cold seawater are selected as the state variables. The difficulty of the static calculation in the simple dynamic model on the construction of state space model is also clarified. To cope with this issue, in this research, the relationships between the state variables at steady state and the manipulated variable of warm seawater flow rate are derived. The usefulness and limitation of the proposed model is verified by simulation results.

Keywords: Ocean thermal energy conversion, state space model, Rankine cycle, heat flow rate dynamics, numerical simulation

1. INTRODUCTION

In recent years, renewable energies such as solar energy, wind energy, tidal energy, ocean wave energy and so on have been paid much attention from the environmental point of view, and the significance of their modeling and control is also increased. As one of the renewable energies, ocean thermal energy conversion (OTEC) technology has been developed over a few decades. (e.g., See Khaligh and Onar (2010).) Although the OTEC system has a drawback of low thermal efficiency, some kinds of OTEC systems with higher thermal efficiency have been constructed such as Uehara cycle Uehara et al. (1998), double-stage Rankine cycle Ikegami et al. (2018) and so on.

For the control of OTEC plant using Uehara cycle, in Goto et al. (2011), a plant model for numerical simulation was constructed based on physical laws about mass and energy. By using the model, in Matsuda et al. (2017-1), a power generation control method for OTEC plant using Uehara cycle was proposed based on seawater flow rate regulation, where warm seawater temperature variation was taken into account as a disturbance. The above model and control system for OTEC plant using Uehara cycle were evaluated by simulation results. Furthermore, liquid level control of separator in OTEC plant using Uehara cycle was considered in Matsuda et al. (2016), where the model was derived by using experimental data. The control system was designed based on LQG control theory. In Matsuda et al. (2017-2), the liquid level model was evaluated by using another experimental data. In Matsuda et al. (2017-3), the control system was improved by resetting the integral action of the controller.

On the other hand, OTEC plant using double-stage Rankine cycle was also modeled in Goto et al. (2017). In the model construction, dynamics on not only temperatures of warm and cold seawater but also heat flow rates (i.e., heat exchange in evaporator and condenser) were considered. By using the model, in Matsuda et al. (2017-4), a power generation control method of OTEC plant using double-stage Rankine cycle with warm seawater temperature variation was proposed. In the control system, either warm seawater flow rate or cold one was adopted as the manipulated variable. In Matsuda et al. (2018), simultaneous regulation of multiple flow rate for the control system in Matsuda et al. (2017-4) was considered, where not only seawater flow rate but also working fluid flow rate was utilized. Furthermore, in Matsuda et al. (2019), control system considered in Matsuda et al. (2018) with target power output changes was investigated. Here, the modeling of OTEC plant using double-stage Rankine cycle with time delay about the movement of fluids was also considered in Aosaki et al. (2019). In the above control system for OTEC plant using double-stage Rankine cycle, control parameters in PI controllers to determine the flow rates were selected by trial and error through many simulations. This problem comes from the complexity of static calculation in Goto et al. (2017). Here, a related work of state space model for OTEC plant using Rankine cycle in Jitsuhara et al. (1994) is explained. In Jitsuhara et al. (1994) is explained.
et al. (1994), a state space model was constructed to represent the vapor... (13)

\[ p = a_{p0} + a_{p1}T \]  
\[ s_v = a_{sv0} + a_{sv1}T \] (for saturated vapor)  
\[ s_l = a_{sl0} + a_{sl1}T \] (for saturated liquid)

To solve these issues, in this research, a state space model for overall OTEC plant using Rankine cycle was newly derived, where the model for overall OTEC plant using Rankine cycle was not proposed as long as the authors know except for Jitsuhara et al. (1994). Therefore, it is important to investigate the construction of state space model for overall OTEC plant using Rankine cycle.

2. SIMPLE DYNAMIC MODEL OF OTEC PLANT USING RANKINE CYCLE

2.1 Principle of Power Generation

The structure of OTEC plant using Rankine cycle is shown in Fig. 1. In evaporator and condenser, heat between seawater and working fluid with low boiling point is exchanged. The state of working fluid passing through evaporator is changed from liquid to vapor. The vapor working fluid rotates turbine connected to generator. Then, generator generates electricity. Pumps send seawater and liquid working fluid. Point 1

2.2 Simple Dynamic Model

Simple dynamic model has dynamics about temperatures \( T_{wso}(t) \) and \( T_{cs0}(t) \) of outlet warm and cold seawater and heat flow rates \( Q_{ws}(t) \) and \( Q_{cs}(t) \) of working fluid in evaporator and condenser:

\[ \tau_{T_{wso}} \frac{dT_{wso}(t)}{dt} + T_{wso}(t) = T_{wso}^s(t) \]  
\[ \tau_{T_{cs0}} \frac{dT_{cs0}(t)}{dt} + T_{cs0}(t) = T_{cs0}^s(t) \]
where \( T = T_e \) or \( T_c \), and the coefficients of polynomials were obtained from the approximation of steam table for Ammonia (Haar and Gallagher (1974)). The temperatures \( T_e, T_c \) are calculated from logarithmic mean temperature and overall heat transfer coefficient:

\[
T_e = -b_{c0} + b_{c1}T_{wso}^{ss}
\]

\[
T_c = -b_{c0} + b_{c1}T_{cso}^{ss}
\]

where

\[
b_{c0} = \frac{T_{wso}^{ss}}{\exp(UA_c/(m_{ws}c_p)) - 1}
\]

\[
b_{c1} = \frac{\exp(UA_c/(m_{ws}c_p))}{\exp(UA_c/(m_{ws}c_p)) - 1}
\]

\[
b_{c0} = \frac{T_{cso}^{ss}}{1 - \exp(UA_c/(m_{cs}c_p))}
\]

\[
b_{c1} = \frac{\exp(UA_c/(m_{cs}c_p))}{1 - \exp(UA_c/(m_{cs}c_p))}
\]

\[
U
\]

\( U \) is the overall heat transfer coefficient, and \( A_c \) is the heat transfer area of evaporator (condenser). The other relations about static calculation are similar with those in Goto et al. (2017).

For (5) and (6), from (7)-(10), (12)-(23) and the other relations we have

\[
k_{c1}T_{wso}^{ss} + k_{c2}T_{cso}^{ss} + k_{c3}(T_{cso}^{ss})^2 + k_{c4}T_{cso}^{ss} + k_{c5} = 0
\]

\[
k_{c1}(T_{cso}^{ss})^2 + k_{c2}T_{cso}^{ss} - k_{c3}T_{wso}^{ss} - k_{c4}T_{cso}^{ss} + k_{c5} = 0,
\]

where

\[
k_{c1} = a_{h0}b_{c1} - b_{c1}a_{p1}a_{v10} - b_{c1}b_{0}a_{p1}a_{v11} + \frac{m_{ws}c_p}{m_{wf}}
\]

\[
k_{c2} = b_{c2}b_{c1}a_{p1}a_{v11}
\]

\[
k_{c3} = a_{p1}a_{v11} + b_{c1}a_{h1}^2
\]

\[
k_{c4} = b_{c1}a_{h1} - b_{c1}a_{p1}a_{v10} - 2b_{c1}b_{c1}a_{p1}a_{v11}
\]

\[
k_{c5} = a_{h1} - a_{v1}b_{0}a_{v10} - b_{c1}a_{h1} + b_{c1}a_{p1}a_{v10} + b_{c1}b_{c1}a_{p1}a_{v11} + b_{c1}^2a_{p1}a_{v11} - \frac{m_{ws}c_pT_{wso}^{ss}}{m_{wf}}
\]

\[
k_{c1} = b_{c1}a_{h1}b_{11} - b_{c1}a_{h1}a_{v1}
\]

\[
- m_{cs}c_p(b_{c1}a_{sv1} + b_{c1}a_{sl1})
\]

\[
k_{c2} = b_{c1}a_{h11}(a_{hv0} + a_{v1}b_{0}a_{v10} - b_{c1}a_{h1} + b_{c1}a_{p1}a_{v10} + b_{c1}b_{c1}a_{p1}a_{v11} + b_{c1}^2a_{p1}a_{v11})
\]

\[
+ (a_{sv0} + b_{c1}a_{sv1}a_{sv0} - a_{sl0} + b_{c1}a_{sl1})
\]

\[
- m_{cs}c_p(a_{sv0} - b_{c1}a_{sv1}a_{sv0} - a_{sl0} - b_{c1}a_{sl1})
\]

\[
k_{c3} = b_{c1}a_{sv1}(b_{c1}a_{h1} - b_{c1}a_{h1}^2)
\]

\[
k_{c4} = b_{c1}a_{sv1}(a_{hv0} + a_{h1}b_{0}a_{h10} - b_{c1}a_{h1} + b_{c1}a_{p1}a_{h10} + b_{c1}b_{c1}a_{p1}a_{h11} + b_{c1}^2a_{p1}a_{h11})
\]

\[
k_{c5} = (a_{sv0} + b_{c1}a_{sv1}a_{sv0} - a_{sl0} - b_{c1}a_{sl1})
\]

\[
+ m_{cs}c_pT_{cso}(b_{c1}a_{sv1} + b_{c1}a_{sl1})
\]

\[
k_{c3} = b_{c1}a_{sv1}(b_{c1}a_{h1} + b_{c1}a_{h1})
\]

Since the equations (24) and (25) are severely nonlinear in \( m_{ws}, m_{cs} \) and \( m_{wf} \), controllers for the determination of them are difficult to design directly.

Here, in this research, the power output \( W \) is defined by

\[
W = \eta m_{wf} (h_1 - h_2),
\]

where \( \eta \) is the turbine efficiency. By the similar manner for (5) and (6) explained above, we also have a severely nonlinear expression of \( W(t) \):

\[
W = g(T_{wso}, T_{cso}, Q_{ws}, Q_{cs}, m_{ws}, m_{cs}, m_{wf}).
\]

The explicit expression of \( g \) is omitted due to lack of space.

3. CONSTRUCTION OF STATE SPACE MODEL

In this section, a state space model for OTEC plant using double-stage Rankine cycle is constructed, where as the input \( u(t) \) and output \( y(t) \), warm seawater flow rate \( m_{ws}(t) \) and power output \( W(t) \) are considered, respectively:

\[
u(t) = m_{ws}(t)
\]

\[
y(t) = W(t).
\]

Defining a vector

\[
x(t) = [T_{wso}(t) \ T_{cso}(t) \ Q_{ws}(t) \ Q_{cs}(t)]^T,
\]

we have

\[
\dot{x}(t) = Ax(t) - Ax^s(t)
\]

from (1)-(4), where

\[
x^s(t) = [T_{wso}^s(t) \ T_{cso}^s(t) \ Q_{ws}^s(t) \ Q_{cs}^s(t)]^T \]

\[
A = \text{diag} \left\{ -\frac{1}{\tau_{T_{wso}}}, -\frac{1}{\tau_{T_{cso}}}, -\frac{1}{\tau_{Q_{ws}}}, -\frac{1}{\tau_{Q_{cs}}} \right\}.
\]

As mentioned above, the vector \( x^s(t) \) is the nonlinear function of \( m_{ws}(t) \):

\[
x^s = f(m_{ws}).
\]

In this research, linear approximation of \( f(m_{ws}) \) is adopted as one of the simplest expression:

\[
f(m_{ws}) \approx \alpha_1 m_{ws} + \alpha_0,
\]

where the coefficient vectors \( \alpha_0 \) and \( \alpha_1 \) are determined by simulations using the conventional simple dynamic model. On the other hand, linear approximation of \( g(T_{wso}, T_{cso}, Q_{ws}, Q_{cs}, m_{ws}, m_{cs}, m_{wf}) \) with respect to \( T_{wso} \) and \( T_{cso} \) is considered:
Table 1. Conditions for numerical simulations

| Parameter                                | Value |
|------------------------------------------|-------|
| Warm seawater inlet temperature $T_{w_{si}}$ ($^\circ$C) | 29.0  |
| Cold seawater inlet temperature $T_{c_{so}}$ ($^\circ$C) | 9.0   |
| Mass flow rate of cold seawater $m_{cs}$ (kg/s) | 36.94 |
| Mass flow rate of working fluid $m_{wf}$ (kg/s) | 0.35  |
| Overall heat transfer coefficient (kW/(m$^2$·K)) | 2.0   |
| Heat transfer area of evaporator (m$^2$) | 87.4  |
| Heat transfer area of condenser (m$^2$) | 87.4  |
| Specific heat of seawater $c$ (J/(kg·K)) | 4.179 |
| Time constant $\tau_{w_{so}}$ (s) | 3.0   |
| Time constant $\tau_{c_{so}}$ (s) | 3.1   |
| Time constant $\tau_{w_{so}}$ (s) | 4.0   |
| Time constant $\tau_{c_{so}}$ (s) | 4.0   |
| Turbine efficiency $\eta_t$ | 0.85  |
| Error bounds $\varepsilon_c$ and $\varepsilon_c$ | 0.0001 |

The approximated results are depicted in Figs. 2 and 3.

4. SIMULATION RESULTS

In order to evaluate the state space model proposed in this research, numerical simulations were conducted. The simulation conditions are listed in Table 1.

4.1 Model Construction

In order to obtain linear approximations (33) and (34), simulation results by static calculation using conventional model for $m_{ws} = 38$-84 kg/s were used. Then, we obtained

$$
\begin{align*}
g(T_{w_{so}}, T_{c_{so}}) \approx & \beta_{w1} T_{w_{so}} + \beta_{c1} T_{c_{so}} + \beta_0 \\
& \text{(34)}
\end{align*}
$$

under the assumption that the behavior of $W(t)$ can be sufficiently captured by $T_{w_{so}}$ and $T_{c_{so}}$, where the coefficients $\beta_{w1}$, $\beta_{c1}$ and $\beta_0$ are determined by simulations using the conventional simple dynamic model.

Thus, by substituting (33) and (34) into (31) and (27), we obtain

$$
\dot{x}(t) = Ax(t) + Bu(t) + \zeta_x \\
y(t) = Cx(t) + \zeta_y,
$$

(35)

(36)

where

$B = -A\alpha_1, \quad \zeta_x = -A\alpha_0$

$C = [\beta_{w1} \beta_{c1} 0 0], \quad \zeta_y = \beta_0.$

4.3 Control Simulations

Furthermore, control simulations were carried out. To determine the control input $u(t) = m_{ws}(t)$, PI controller was applied:

$$
m_{ws}(t) = m_{ws0} + K_p \left( e(t) + \frac{1}{T_i} \int_0^t e(\tau) \, d\tau \right),
$$

(37)

where $e(t) = W_{ref} - W(t)$ is the control error for the target power output $W_{ref}$, and $m_{ws0} = 39.72$ kg/s is the standard flow rate. In Fig. 6, a simulation result for $W_{ref} = 15.2$ kW is shown, where the parameters $K_p$ and $T_i$ were set as $K_p = 0.2$ (kg/s)/kW and $T_i = 0.05$ s, respectively. In Fig. 7, another simulation result for $W_{ref} = 12$ kW is shown, where the parameters $K_p$ and $T_i$ were set as $K_p = 0.03$ (kg/s)/kW and $T_i = 0.05$ s, respectively. The parameters $K_p$ and $T_i$ were determined through simulations using the conventional model.

5. DISCUSSION

5.1 Model Construction

In this research, the static calculation in the simple dynamic model is explicitly described. Although the approx-
Fig. 4. Simulation result of step response for $m_{ws} = 40-50$ kg/s

Fig. 5. Simulation result of step response for $m_{ws} = 20-30$ kg/s

Estimated equations described by 1st order polynomials were used, the resultant relations were severely nonlinear and so complicated. This result clarifies the difficulty in the construction of control mechanisms using flow rates $m_{ws}(t)$, $m_{cs}(t)$, and $m_{wf}(t)$ based on the model. Therefore, to cope with this issue, in this research, linear approximation is considered. From the linear approximation we can derive a linear time-invariant system described by (35) and (36) easily. This system is useful in designing the control system by applying control theories based on state space model. However, the validity (or applicable range) of the linear approximation should be carefully verified corresponding to the purpose.

Here, let us check the similarity and difference of the proposed model with conventional ones (e.g., in Aosaki et al. (2019), Matsuda et al. (2017-4) etc.). In both models, the dynamics on temperatures and heat flow rates are represented by 1st order systems. This is the similarity. However, in the proposed model, approximated equations (12)-(17) in 2.2 are represented by 1st order polynomials. On the other hand, approximated equations in the conventional models have higher order polynomials. This is the difference.
Although the power output $W(t)$ was selected as the output $y(t)$ for the comparison of the proposed model with conventional one, we can adopt any other quantities corresponding to the purpose.

5.2 Simulation Results

In order to check the behavior of the proposed model, 2 kinds of step responses were considered as shown in Figs. 4 and 5. Fig. 4 is the step response for $m_{ws} = 40-50$ kg/s. This result implies that the simulation result by proposed model was sufficiently close to that by conventional model since the linear approximation was considered for $m_{ws} = 38-84$ kg/s. On the other hand, Fig. 5 is the step response for $m_{ws} = 20-30$ kg/s. This figure indicates that the simulation result (especially, for $T_{wso}$) was not close since the input $m_{ws}$ was out of range of the linear approximation. Therefore, the range of linear approximation should be appropriately chosen.

For comparison of control results, simulations using PI control system were performed as shown in Figs. 6 and 7. The results in Fig. 6 clarifies the usefulness of the proposed model since the power output $W(t)$ reached the target one $W_{ref}$ in both cases. However, in Fig. 7, the power output $W(t)$ did not reach the target one $W_{ref}$ when the proposed model was used. The result may be caused by the linear approximation. Therefore, further investigation of the modeling using state space model with linear approximation should be required.

6. CONCLUSION

In this research, a state space model for OTEC plant using Rankine cycle was proposed by applying linear approximation to an existing model of simple dynamic model. The complexity of the static calculation in the simple dynamic model was confirmed analytically. To obtain a model for the application of advanced control theories, a state space model was constructed by linear approximation for the static calculation. The simulation results of step response for the proposed model showed its usefulness and limitation. Furthermore, the simulation results of control by PI controller indicated that the control system using proposed model could be valid. However, the simulation results also exhibited the necessity of further improvement of the proposed model. In future works, systematic control system design procedure will be investigated by using the proposed model. Furthermore, the control system including the proposed model will be evaluated through experiments using an actual experimental plant.

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