Multi-objective optimization of a closed-cycle OTEC system

Zhihao DING[1], Jean-Luc ACHARD[2], Christophe Corre[1]

[1] LMFA - Laboratoire de Mecanique des Fluides et d'Acoustique
Ecole Centrale de Lyon, 36 avenue Guy de Collongue, 69134 Ecully Cedex, France
christophe.corre@ec-lyon.fr
[2] LEGI - Laboratoire des Ecoulements Géophysiques et Industriels
Domaine Universitaire, CS 40700, 38058 Grenoble Cedex 9, France

An Ocean Thermal Energy Conversion (OTEC) system converts heat energy into electricity by using the temperature difference between the warm water at the ocean surface in tropical areas and the cold water of the depths. Following the initial concept first proposed by D'Arsonval in 1881, an OTEC plant relies on a Rankine cycle in which the hot source is provided by the warm seawater while the cold source corresponds to the cold water of the depths. The major components of an OTEC plant are (see also Fig.1) : i) the heat exchangers in which the working fluid is respectively evaporated and condensed; ii) the turbine where the gaseous working fluid expands to produce a mechanical energy which is then converted into electricity; iii) the pumps which ensure the circulation of the warm seawater, cold seawater and working fluid; iv) the cold pipe through which the cold water is pumped from the ocean depths (about $1000\,m$ deep) to the surface level.
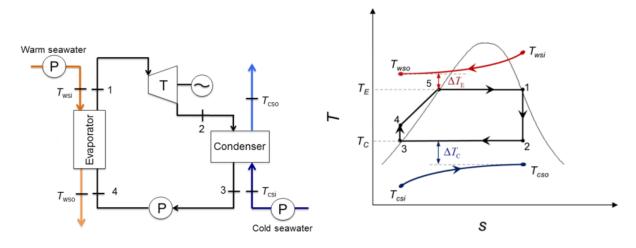

Figure 1: Left : schematic view of a closed-cycle OTEC system. Right : corresponding Rankine cycle.

Due to the small temperature difference (about $20°C$) available for the OTEC plan, the Rankine cycle efficiency remains necessarily very low (a few %) so that it becomes essential to come up with an optimized economical cost to build the OTEC plant. Assuming the plant is built on a prescribed site where the ocean temperature distribution is known and also assuming a given working fluid, the key design parameters of the OTEC plant are : the high and low temperatures of the Rankine cycle, respectively T_E and T_C in the evaporator and in the condenser; the temperature differences $\Delta T_E = T_{wso} - T_E$ and $\Delta T_C = T_C - T_{cso}$ respectively between the cooled warm seawater leaving the evaporator and T_E and between T_C and the heated cold seawater leaving the condenser; the length L_{cs} and diameter D_{cs} of the pipe through which the cold seawater is pumped from the ocean depth to the sea level; the geometrical parameters of the condenser and evaporator. The performance of the OTEC plant can be characterized by the net power output W_e^{net} defined as $W_e^{net} = W_e - (W_{wf} + W_{ws} + W_{cs})$ where the electric power W_e is computed from the working fluid mass flowrate, the enthalpy difference between states 1 and 2 and the turbine efficiency. Note that W_e is prescribed, for instance a $10\,MWe$ plant is targeted, so that for a given turbine efficiency and known values of enthalpies h_1, h_2 from T_E and T_C the working fluid mass flowrate can be computed. The net power output W_e^{net} is obtained by substracting from the electric power the pumping power needed for the circulation of the working fluid, the warm seawater and the cold seawater. This pumping power is needed to balance the head losses through the heat exchangers and the pipes, mostly the long cold seawater pipe. It depends on the seawater mass flowrates and on the geometric parameters of the condenser, evaporator and cold seawater pipe.
The key contributions to the economic cost of the OTEC plant are the heat exchangers and the cold

seawater pipe. The respective cost of these components can be considered as proportional to their total area A_{tot}. The size of the cold seawater pipe, in particular the diameter of this pipe, plays also a key role in the overall robustness of the system, with a large-diameter pipe more sensitive to rough sea states.

Early studies on the design of OTEC systems [1] [2] [3] [4] proposed to minimize the ratio A_{tot}/W_e^{net} and typically found the optimum vector of design parameters thanks to a steepest descent algorithm. These early works also often simplified the optimization problem by assuming for instance fixed head losses in the heat exchangers while these losses are actually dependent upon the evaporator and condenser design, this design being itself determinant to achieve the heat transfer processes required by the Rankine cycle. Numerous studies (see [5] [6]) have been devoted to the optimal design of heat exchangers using global optimization algorithms (genetic or particle swarm optimization algorithms), which make it possible to consider the simultaneous optimization of several objectives, such as for instance the minimization of the heat exchanger total surface (hence cost) and the minimization of the head losses through these exchangers. However, these works decouple the design of the heat exchangers from the global design of the system in which they are to be inserted. Similarly, global optimization method have also been applied to the design of Organic Rankine Cycles (see for instance [7]) but not in the specific context of an OTEC system application. Reference [8] is one of the very few works devoted to the multi-objective optimization of an OTEC plant for several target values of W_e but appears to lack a detailed description of the heat exchangers design. The present contribution investigates the optimal design of an ammonia-based closed-cycle OTEC plant using a numerical modeling which includes the geometry of the heat exchangers among the design parameters, along with the cycle temperatures and temperature differences and the cold seawater pipe geometry. Pareto-optimal designs will be analyzed in the objective (see Fig.2) and parameter space so as to provide guidelines and recommendations for the design of OTEC plants.
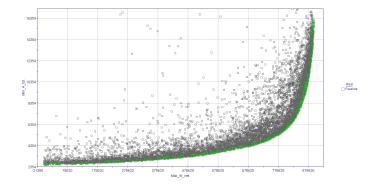

Figure 2: Pareto front for the optimal design of a $10\,MW$ OTEC plant in the space of the objectives : W_e^{net} along the x-axis and A_{tot} along the y-axis.

References

[1] E. N. Ganic and L. Moeller. Performance study of an OTEC system. *Applied Energy*, 6(4):289–299, 1980.

[2] Haruo Uehara and Yasuyuki Ikegami. Optimization of a Closed-Cycle OTEC system. *Journal of Solar Energy Engineering*, 112(June 1990):247–256, 1990.

[3] Rong Hua Yeh, Tar Zen Su, and Min Shong Yang. Maximum output of an OTEC power plant. *Ocean Engineering*, 32(5-6):685–700, 2005.

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

[5] Antonio C. Caputo, Pacifico M. Pelagagge, and Paolo Salini. Heat exchanger design based on economic optimisation. *Applied Thermal Engineering*, 28(10):1151–1159, 2008.

[6] Salim Fettaka, Jules Thibault, and Yash Gupta. Design of shell-and-tube heat exchangers using multiobjective optimization. *International Journal of Heat and Mass Transfer*, 60(1):343–354, 2013.

[7] Leonardo Pierobon, Tuong Van Nguyen, Ulrik Larsen, Fredrik Haglind, and Brian Elmegaard. Multi-objective optimization of organic Rankine cycles for waste heat recovery: Application in an offshore platform. *Energy*, 58:538–549, 2013.

[8] Alireza Najafi, Shahab Rezaee, and Farschad Torabi. Multi-objective optimization of ocean thermal energy conversion power plant via genetic algorithm. In *IEEE 2011 EPEC*, Toronto, 2011.