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Performance evaluation and parametric optimum design of an updated ocean thermal energy conversion system

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ABSTRACT

A cycle model of the updated ocean thermal energy conversion system is proposed. The cycle system mainly consists of a compressor, a turbine, an evaporator, a condenser, and a regenerator and can absorb heat from high-temperature heat sources such as solar energy, waste heat produced in other equipments, or heat warm sea water. The effects of main irreversible losses on the performance of the system are considered. Expressions for the power output and efficiency of the system are analytically derived. The characteristic curves of the dimensionless power output versus the efficiency are obtained. The optimum values of some of the main parameters at the maximum power output are calculated. The results obtained here can provide some guidance for the optimum design and operation of real ocean thermal conversion systems.

1. Introduction

Energy shortage and environmental pollution are two critical issues in the 21-th century that must be appropriately solved. The low-grade heat from renewable energy sources has been considered to be a good candidate to solve the two issues. Among those energy sources, ocean thermal energy is one of the largest renewable energy sources in the world because oceans cover approximately 70% of the Earth's surface and the majority of the absorbed solar energy is stored in the upper layer of oceans. The utilization of ocean thermal energy has attracted considerable attention in recent decades (Aydin et al., 2014; Lee et al., 2014; Rajagopalan and Nihous, 2013; Semmari et al., 2012; Yeh et al., 2005).

Ocean thermal energy conversion (OTEC) plants are one of the most important equipments in the utilization of ocean thermal energy. They generate electricity by using the temperature difference between the sea surface (20–30 °C) and the deep-sea (3–10 °C). Many researchers have investigated the performance of OTEC systems. For example, Ganic and Moeller investigated the performance of an OTEC plant (Ganic and Moeller, 1980). Wu calculated the performance bound of real OTEC heat engines (Wu, 1987) and optimized the specific power output of closed-cycle OTEC plants (Wu, 1990). Tseng et al. (1991) utilized an optimal design concept to find the best design for a complex and large-scale OTEC plant. Lennard discussed the viability and best locations for OTEC systems around the world (Lennard, 1995). Yeh et al. calculated the maximum work output of an OTEC power plant (Yeh et al., 2005). Moore and Martin presented a general mathematical framework for the synthesis of OTEC power generating systems and performed a technical analysis of an OTEC system (Moore and Martin, 2008). Sun et al. derived and optimized the performance analytical function and exergy efficiency of organic Rankine cycle in the OTEC (Sun et al., 2012). So far many significant results have been obtained for the design and operation of OTEC systems (Nihous, 2007; Faizal and Ahmed, 2011; Soto and Vergara, 2014). However, the available temperature difference is small for OTEC systems so that the corresponding net power output is limited. Thus, under the limited external conditions, the optimum study to achieve the maximum available energy from seawater is one way of the utmost importance for the performance investigation of OTEC systems. On the other hand, the construction and analysis of some novel cycles are also an important topic in the investigation of ocean energy conversion.

In the present paper, an updated model of OTEC systems is proposed. Expressions for the efficiency and power output of the new model are derived. The general performance characteristics of the power output versus the efficiency are revealed. The main parameters are optimized. Some significant results are obtained.

2. The description of an updated OTEC system

Fig. 1 represents the cycle model of an updated OTEC system, which is different from a common OTEC system (Aydin et al., 2014;

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Nomenclature

$A_i (i = h, c, r)$ heat transfer area
c_p specific heat at constant pressure
$\dot{m}$ mass flow rate
P power output
p dimensionless power output
p_{max} maximum dimensionless power output
q_c heat releasing rate
q_h heat input rate
q_r regenerative rate
r_p pressure ratio
T_0 temperature of environment
T_c temperature of cold see water
T_h temperature of heat reservoir
T_i temperature of state point i
$U_i (i = h, c, r)$ heat transfer coefficient

Greek letters

γ ratio of specific heats
η efficiency
η_c compression efficiency
η_e expansion efficiency
η_p efficiency at maximum power output
η_r regeneration efficiency

Subscript

opt optimum

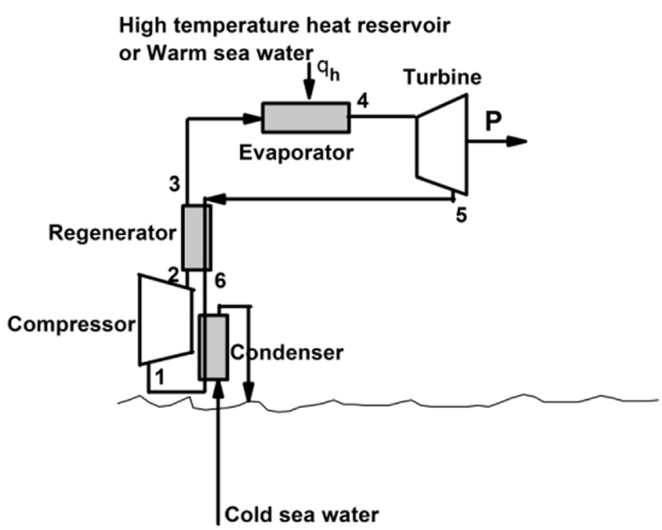

Fig. 1. The schematic diagram of an updated OTEC system.

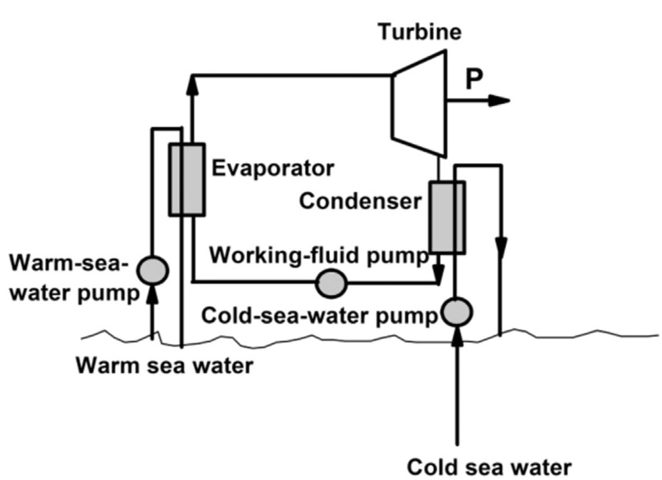

Fig. 2. The schematic diagram of an OTEC system.

Moore and Martin, 2008; Sun et al., 2012; Faizal and Ahmed, 2011; Soto and Vergara, 2014) shown in Fig. 2. In Fig. 1, the cycle system mainly consists of a compressor, a turbine, an evaporator, a condenser, and a regenerator, where the evaporator and condenser are employed as heat exchangers to absorb and release the heat energy, respectively. The system is operated between the two heat

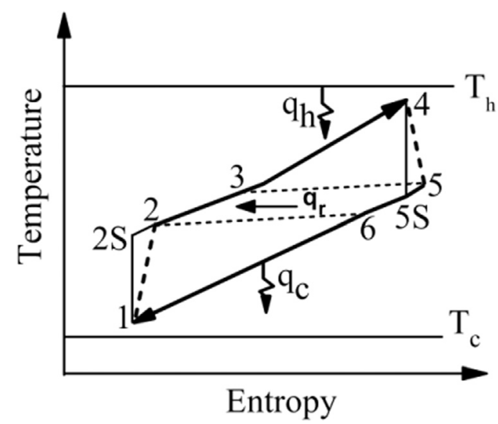

Fig. 3. The temperature-entropy diagram of an updated OTEC system.

reservoirs at temperatures T_h and T_c. q_h may be supplied by high-temperature heat sources such as solar energy, waste heat produced in other equipments, or heat warm sea water. The working fluid at temperature T_1 enters into the compressor and is compressed irreversibly and adiabatically, while the working fluid leaving the compressor, under a given pressure, is heated in the regenerator by the working fluid from the turbine. The heated fluid enters the evaporator and continues to be heated up by the high-temperature heat reservoir, and the hot fluid enters directly into the turbine and expands to produce power via an irreversible adiabatic process. Finally, the working fluid flowing through the regenerator enters the condenser and is cooled to temperature T_1 under another given pressure. It shows that the basic thermodynamic cycle of the system may be modeled as a closed Brayton cycle consisting of two isobaric processes, two adiabatic processes, and a regenerative process and its temperature-entropy diagram is shown in Fig. 3, where $T_i (i = 1, 2, \ldots, 6)$ are the temperatures of the working fluid at state points i, 1–2S and 4–5S are two reversible adiabatic processes; 1–2 and 4–5 are two irreversible adiabatic processes, q_h and q_c are, respectively, the rates of heat absorbed from the heat reservoir at temperature T_h and released to the cold sea water at temperature T_c, and q_r is the rate of heat flow in the regenerator. Because the heat capacities of two heat reservoirs are often much larger than that of the working fluid in the cycle system, the temperatures T_h and T_c of the high-temperature heat reservoir and cold sea water are taken as constant in this cyclic model. The proposed model carries forward the advantages of the various Brayton models adopted in Wu et al., (1996),