OCEAN THERMAL ENERGY CONVERSION (OTEC) SYSTEMS: LOSSES DUE TO HEAT TRANSFER IN COLD WATER PIPES

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Summary Ocean Thermal Energy Conversion (OTEC) systems harness the natural temperature difference (ΔT) between surface water and deep seawater to generate power in the form of electricity. The temperature of seawater however fluctuates based on the geographical location, as well as on factors such as ocean depth and proximity to the coastline. Therefore, the distance from shore might be very long where the requisite ΔT is met for the system to function optimally. This study assesses scenarios where the required ΔT is at a high distance from shore, by evaluating the heat transfer losses in the cold-water pipe (CWP). A computational investigation was conducted on the CWP to evaluate a more precise ΔT, as in most literature studies, the ΔT used in the performance assessments does not consider any heat transfer losses.

INTRODUCTION

Renewable Energy systems (RES) related to the ocean and marine environment have seen a significant advancement in recent years, due to the promotion of such RES for the reduction of fossil fuels and CO2 emissions in general. Ocean Thermal Energy Conversion (OTEC) systems can be categorized as RES, as they exploit the stored solar thermal energy in the ocean surface. The natural temperature difference ΔT between the surface of the sea and at high sea depths, of approximately 1 km, gives rise to such exploitation potential. To assess the efficiency of OTEC systems, researchers commonly assume a seawater temperature difference of around 20-24°C [1], depending on the location data. A ΔT of 20°C or higher is estimated to provide a Carnot efficiency of 6.7% [2]. However, due to the heat transfer loss at the CWP, the actual temperature difference (ΔT) used for estimating the thermodynamic cycle may deviate from the assumed ΔT. In the current paper a computational investigation of the heat transfer losses that occur when cold seawater flows from the deep sea to the sea surface through a cold-water pipe (CWP) is performed.

METHODS AND INITIAL RESULTS

COMSOL Multiphysics software serves as the primary computational tool, where the necessary equations, as suggested in the literature, are implemented. A verifications of the model and boundary conditions is conducted using existing literature studies. The diameter and the CWP lengths are directly related to the required pumping power and the pressure losses and consequently have an impact on the overall performance of the OTEC system. The current study assumes a usage of HDPE pipes with a fixed diameter, allowing the flowrate and the total length to vary. The computational analysis, although it can be generalised, it is focused here on the data pertaining to the precise location of an operational OTEC system situated in Hawaii, namely at the Natural Energy Laboratory in Kailua-Kona, HI, USA. The length of the pipe and the distance from the shore are also influenced by the characteristics of the ocean floor. An example of this is the KRISO project, which is located within a 5km horizontal distance from the shore and requires a depth of 3.5-4 km [3], [4]. The geometry of the CWP is analysed using both a realistic scenario method and a simplified approach, where the pipe is laid in a straight manner toward the seabed.

The convection-diffusion heat transfer equation is used with a one-dimensional simplified [5] domain on the pipes as described below:

$$\rho A c_p \frac{\partial T}{\partial t} + \rho A c_p u e_t \cdot \nabla_t T = \nabla_t \cdot (A \lambda \nabla_t T) + \frac{1}{2} f_D \frac{\rho A}{d_i} |u| u^2 + Q_{wall} \tag{1}$$

where ρ is the density of the fluid [kg m^{-3}], , A is the area of the pipe [m^2], c_p is the specific heat capacity [J kg–1 K–1], T is the temperature [K], t is time [s], ue_t is the tangential velocity [m s^{-1}], λ the thermal conductivity [W m^{-1} K^{-1}], f_D the Darcy's friction factor where in this study the Colebrook approximation is used, d_i is the inner diameter of the pipe [m], and Q_{wall} is the heat conduction given by the effective heat transfer coefficient of the pipe $(\mathrm{h_p})_{\mathrm{eff}}$, and the wall perimeter of the pipe (Z), described as:

$$\mathrm{Q_{wall}} = (\mathrm{h_p Z})_{\mathrm{eff}} (\mathrm{T_{ext}} - \mathrm{T_f}) \tag{2}$$

At the boundary of the intersection between the fluid and the pipe, a convective heat flux boundary condition is applied, which depends on the Nusselt number (Nu). The Nu is assumed at 3.66 for laminar flows, whereas for the turbulence flows, the Nu is estimated by COMSOL using the Gnielinski correlation [6]. A two-dimensional domain is constructed in this study.

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RESULTS AND DISCUSSION

An initial investigation was performed to compare the realistic geometry of the CWP against the simplified straight-line geometry. Only a 20m disparity is projected in the length of the CWP. The results demonstrate that with flowrates of 0.5 m^3 s^{-1} or higher, or when the velocity is at 1 m s^{-1} or higher, there is minimal to no variation (2.1% or less) in the resulting temperature of the CWP. For flowrates below 0.05 m^3 s^{-1}, a percentage difference of at least 16.8% is seen in terms of ΔT compared to the initial temperature.

Figure 1 presents the relationship between the mass flowrate (on the left y-axis) and the length of the CWP (on the right y-axis), against the temperature difference between the condenser inlet and the CWP deep seawater inlet. The values used for the CWP diameter and the cold seawater density are 0.7608m and 1028 kg m^{-3}, respectively. It can be observed that at lower flowrate values, a temperature difference is noticed, which will be a disadvantage for the system performance. Nevertheless, the flowrate is a variable that is contingent upon the necessary heat exchange rate and is predetermined by the system designer. It is evident that the temperature difference reduces with flowrate for all four distinct CWP lengths, specifically 10km, 7km, 5km, and 3km. Specifically, when the flowrates are sufficiently high, low temperature differences of less than 1°C are recorded along the CWP, resulting in minimal heat losses and maximising efficiency. However, achieving these low temperature differences may be challenging because of the chosen pipe diameter size. The y-axis on the right side of Figure 1 shows the scenario with a mass flowrate of 50 kg s^{-1}. It is evident that significant temperature variations (>10°C) can be achieved, resulting in substantial heat losses along the CWP. The failure to achieve the necessary temperature difference of approximately 20°C poses a limitation in choosing an onshore structure for an OTEC system designed for power generation. Therefore, in such instances, it could be advisable to opt for an offshore structure, such as a floating one. Furthermore, when considering a mass flowrate of 50 kg s^{-1}, the temperature disparities between a 10km and a 7km, a 5km, and a 3km CWP in an onshore OTEC system are decreased by 22%, 39%, and 59% respectively. It should be noted that, although not displayed in this context, the corresponding percentages for a mass flowrate of 1050 kg s^{-1} are 29%, 48%, and 68%. When comparing the 10km and 1km CWP length cases, which could represent an onshore or offshore system respectively, there is an 86% decrease in temperature difference. This further highlights a significant benefit of offshore systems, since they offer superior performance in situations when low flowrates and a considerable distance from the shore are present.

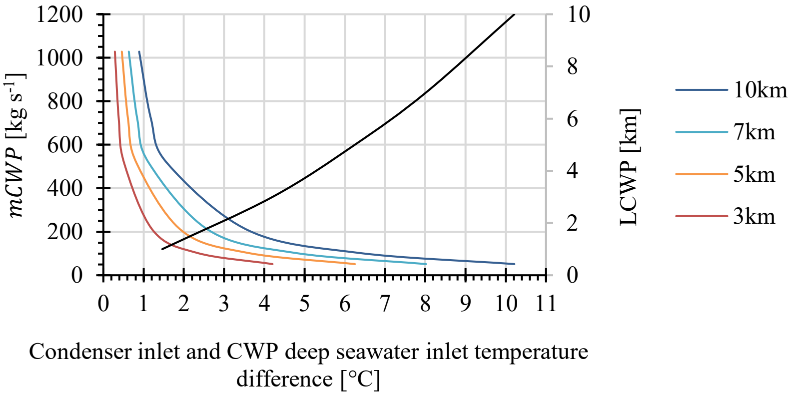

Figure 1. Temperature difference between condenser inlet and CWP deep sweater inlet for different mass flowrates and CWP lengths.

The current computational investigation is further expanded into the different type of pipes such as the fibre reinforced polymer (FRP) that can be used for higher diameters. A sensitivity analysis could also be performed on the seabed characteristics and the effect on the overall CWP length, as this could influence the results further and the simplified approach may be not applicable in some cases.

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