OTEC Systems' Heat Losses in Pipes

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Abstract: - Ocean Thermal Energy Conversion (OTEC) systems utilize the temperature difference between surface seawater and deep seawater to produce electricity through a heat engine. A major disadvantage of the OTEC systems is that seawater temperature on the surface and on the seabed varies with the geographical location. This difference is also dependent on the depth and the distance from the coastline, where there are cases with the required temperature difference size to be found at a high distance from shore. This study evaluates the heat losses of the cold water pipe, where such long distances occur, and the subsequent effects for such cases. The current investigation is performed computationally and is based on the accurate estimation of the temperature difference between the deep seawater temperature and the inlet to the condenser temperature. Most literature studies do not consider any heat losses due to the transfer of the seawater when evaluating the performance of the CWP; however, in some cases, even a small temperature change can have a major effect on the output of the system.

Key-Words: - Ocean Thermal Energy Conversion, Cold Water Pipe, OTEC offshore, OTEC onshore, heat transfer, seawater.

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1 Introduction

Renewable Energy systems (RES), relative to the ocean and marine environment, have experienced substantial progress in recent years, mostly driven by the promotion of these systems as a means to decrease reliance on fossil fuels and reduce CO2 emissions. Ocean Thermal Energy Conversion (OTEC) systems fall under the RES category since they harness the accumulated solar thermal energy present on the ocean surface. The exploitation potential arises from the natural temperature difference between the sea surface and high sea depths.

OTEC systems can be classified into two primary categories: open-cycle (OC-OTEC) systems and closed cycle (CC-OTEC) systems. In an OC-OTEC system, seawater serves as the circulating fluid, with the surface seawater undergoing flash evaporation. The vapor consequently is utilized to power a turbine operating at low pressure, which then operates an electric generator. The water vapor is subsequently condensed using the deep cold seawater. Due to the possible production of fresh water as a byproduct, OC-OTEC systems not only generate electricity but also offer the added benefit of producing distilled water. This particular functionality is a characteristic of hybrid systems, which entail the use of two or more technologies or processes. The OC-OTEC systems, however, have not yet been utilized for large-scale power production. The CC-OTEC systems, on the other hand, operate via thermodynamic cycles, which offer the best potential efficiency for converting heat into work. The efficiency of the Carnot thermodynamic cycle, given a temperature difference of 20°C is estimated at only 6.7%; however, more realistic cycles, such as Organic Rankine, Kalina, and Uehara, exhibit lower efficiencies in the ranges of 3-4%. As in the case of the OC-OTEC system, and the CC-OTEC system, the heat from the surface sea level is utilized to cause the evaporation of a circulating fluid with a low boiling point, such as ammonia. The deep cold seawater is utilized for the condenser, and then it is discharged back into the ocean.

In both categories (OC-OTEC and CC-OTEC systems) and in order to evaluate the effectiveness of OTEC systems, researchers often introduce a seawater temperature difference of approximately 20-24°C, [1], which may vary based on the specific site data.

There are various OTEC case studies available in the literature, including pilot cases and theoretical studies, conducted either for onshore or offshore applications. A specific theoretical OTEC plant of 5MW net power would use an offshore structure and operate on the assumption of a steady cold seawater temperature of 4.5°C, [2]. The seawater would flow at a constant rate of 13.9 m^3 s^{-1} and with a pump efficiency of 72%. The selection of a large diameter (2.72 m) necessitated the use of glass fiber-reinforced plastic (FRP) as the material for the cold-water pipe (CWP). The CWP, functioning as part of an offshore system, had a length of 1 km. A separate investigation was conducted to assess the capabilities of an OTEC facility with a greater power output of 100 MW, [3]. The study, in the state of Florida, USA, focused on a rather high mass flow rate of 366000 kg s^{-1}. The optimal depth, for this case study, of 1 km, and the projected maximum temperature variation are situated at a considerable distance from the Florida coastline, around 98 km away. A smaller temperature gradient of around 18°C can be achieved at a considerably closer distance from the shore, specifically at around 12 km. According to the information provided, it was suggested to use an offshore structure having a length of 1 km and a pipe diameter of 10 m. The efficiency of the CWP pump was considered to be 80%. A recent research, [4], proposed a 10 MW OTEC system in Morotai Island, North Maluku, Indonesia, with a mass flow rate of 29000 kg s^{-1}. Based on several assumptions, the temperature of the CWP at the pipe inlet and the condenser inlet can be considered the same, [5], [6]. The identification of possible sites with significant thermal gradients suitable for OTEC application is discussed in a study, which provides a list of places where the temperature difference is equal to or greater than 20°C and the distance from the shore is less than or equal to 10 km, [7]. The mean distance from the shore was 7.7 km, with the smallest distance being 2.3 km and the largest distance being 10.9 km. In the aforementioned (and most) OTEC modeling studies, it is important to note that no heat losses are considered between the CWP inlet and the condenser inlet. It is assumed that the temperatures at these points are similar. This assumption is supported by many studies in the literature, [8], [9], [10].

The efficiency of OTEC systems is mainly determined by the temperature difference between the deep-water temperature, which is circulated by the CWP, and the sea surface water temperature, which is delivered by the warm-water pipe (WWP). As mentioned earlier, a temperature difference of 20°C can result in efficiencies of up to 6.7% for OTEC systems based on the theoretical Carnot cycle, or lower for other more realistic cycles.

Typically, when estimating the performance of OTEC systems, the condenser and evaporator inlet temperatures are the same as the temperatures of the deep seawater and surface seawater at the site. However, this estimation does not take into account any heat losses that may occur between the pipes and the surrounding environment.

The objective of this research is to use computational methods for the CWP sizing (diameter and length) in relation to heat losses, which in turn affect the temperature difference, pumping power, and efficiency of OTEC systems. The results can also determine the feasibility of the implementation of OTEC systems.

2 Methodology

COMSOL Multiphysics software is used as the main computational tool to implement the required equations. An assessment of the model and boundary conditions is performed by references to current literature studies. The length of the CWP has a direct relationship with the required pumping power and the pressure losses, which in turn affect the overall performance of the OTEC system. The present analysis assumes the utilization of HDPE pipes with a constant diameter while allowing for variations in both the flow rate and the overall length. The parameters that have an effect on the CWP length are the morphology of the ocean bed, as well as the distance from shore that the optimum temperature difference is found. For simplicity, the geometry of the CWP is designed as a straight line towards the required depth.

The convection-diffusion heat transfer equation is applied, with a simplified one-dimensional domain for the pipes, [11],

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

where ρ is the fluid density [kg m^{-3}], A is the pipe area [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 is the thermal conductivity [W m^{-1} K^{-1}], f_D is the Darcy's friction factor, d_i is the inner diameter of the pipe [m].

Note that the friction factor in this study was estimated with the Colebrook approximation based on the solution methods provided by COMSOL. The heat conduction Q_{wall} is determined as an internal heat resistance and depends on the effective heat transfer coefficient of the pipe, and the wall perimeter of the pipe, as:

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

where T_{ext} is the temperature at the external of the pipe wall [K] – in this case it is the seawater temperature surrounding the pipe, T_f is the fluid temperature [K], $(h_p Z)_{eff}$ the effective value of the heat transfer coefficient of the pipe [W m^{-2} K^{-1}], and Z the wall perimeter of the pipe [m]. The interface boundary between the fluid and the pipe, is simulated with an internal convective heat flux, where the convective heat transfer coefficient h_{int} is used as and defined by:

$$h_{int} = \mathrm{Nu} \frac{k}{D_{CWP}} \quad (3)$$

where k is the heat transfer rate, D_{CWP} is the diameter of the CWP, and Nu is the Nusselt number set as 3.66 for circular pipes. For further simplification, the model consists of only one domain, the CWP in a 2D environment.

3 Initial Results

A preliminary examination was conducted using the methodology described in Section 2. Initially, the model was set with a short time step of 0.1 h and for a constant run duration of 1 month. This was performed to evaluate if the temperature became steady in this time frame, and it was observed (not shown here) that after one week the temperatures were steady. Therefore, the computational model was adjusted to run continuously for 15 days. The computational recorded outlet temperature (at the inlet of the condenser) was further examined to investigate ΔT (i.e., the temperature difference between condenser inlet and deep seawater), as shown in Figure 1, for a steady flowrate of 50 kg m^{-3}, where the dotted line illustrates the seawater temperature of 4.4°C at 1-km depth (as reported in the literature). It is observed that there is a rapid increase in temperature at a length of 4 km and lower. This is an effect due to the convective heat transfer increase because of the higher temperature difference between the fluid and the seawater at shallower depths. This could therefore have a negative impact on the system's efficiency.

Figure 2 illustrates the correlation between the mass flow rate (left y-axis) and the length of the CWP (right y-axis) in response to the temperature difference between the condenser inlet and the CWP deep seawater inlet (ΔT).

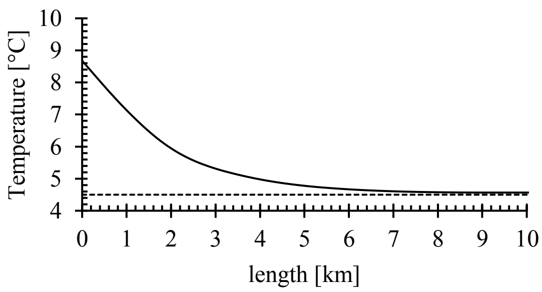

Fig. 1: Temperature variation along the CWP, with the dotted line marking the 4.4°C seawater temperature at 1 km depth

The CWP inner diameter is considered as 0.7608 m with an outer diameter of 0.8 m, and the cold seawater density at 1028 kg m^{-3}. It is evident (Figure 2) that lower mass flow rate values result in a ΔT that will have a negative impact on the system's performance. However, the flow rate is a variable that depends on the required heat exchange rate and is established by the system designer. The relationship between ΔT and mass flow rate is clearly observed for all four varied lengths of CWP, namely 10km, 7km, 5km, and 3km. More precisely, when the rates at which the fluid flows are high enough, in this case study for mass flowrates of over 600 kg s^{-1}, the temperature changes (ΔT) along the CWP are very small, namely less than 1°C. This leads to minimum losses of heat and maximum efficiency. However, attaining these small temperature changes may prove to be difficult due to the selected pipe diameter.

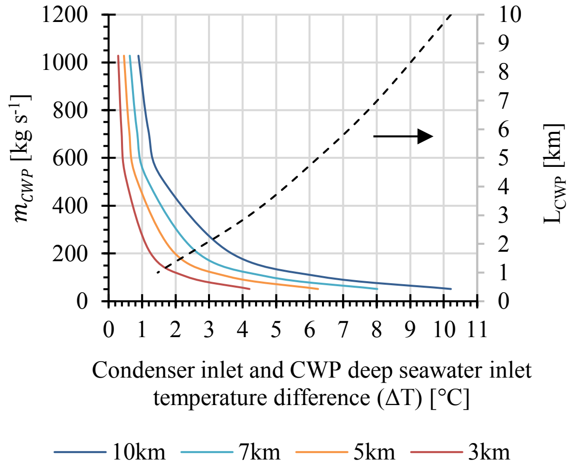

Fig. 2: Temperature difference between condenser inlet and CWP deep sweater inlet (ΔT) for different mass flow rates and CWP lengths

In Figure 2, the right y-axis represents the scenario where the mass flow rate is steady at 50 kg s^{-1} and only the length of the CWP is varied. It is clear that large temperature fluctuations, of over 5°C can occur, leading to considerable heat losses along the CWP. The inability to maintain the required temperature difference of around 20°C presents a constraint when selecting an onshore configuration for an OTEC system intended for electricity production. Therefore, in such cases, it would be beneficial for a selection of an offshore system, such as using a floating platform. However, another solution would be to design a system to efficiently work at higher mass flow rates, since minimal ΔT is observed for flowrates over 600 kg s^{-1} (in the specific case study).

In addition, when examining a mass flowrate of 50 kg s^{-1}, ΔT between a 10km and a 7km, a 5km, and a 3km CWP in an onshore OTEC system is reduced by 22%, 39%, and 59% respectively. It is also important to mention that, even though they are not shown here, the corresponding percentages for a mass flow rate of 1050 kg s^{-1} are 29%, 48%, and 68%. When comparing the scenarios of a 10km and 1km CWP length, which can respectively represent an onshore or offshore system, there is an 86% reduction in ΔT. This emphasizes an important advantage of offshore systems since they provide a better performance in scenarios involving low flow rates and large distances from the shore.

The diameter of the CWP was also compared against the condenser inlet and CWP deep seawater inlet temperature difference (ΔT). Increasing the diameter, the surface area of the pipe wall used for heat exchange is increased; hence, an increase in the temperature increase could be assumed. However, Figure 3 demonstrates that the diameter, D_{CWPo}, of the CWP has a minimal effect (approximately 3-4%) on the condenser inlet temperature. Note that decreasing the CWP diameter does lead to an increased pressure and pumping power for the CWP (not illustrated here). Thus, to reduce the pumping power, it is suggested to increase the pipe diameter. It should be noted that pipes with larger diameters incur higher costs due to the need for increased material required as well as additional labor hours during installation.

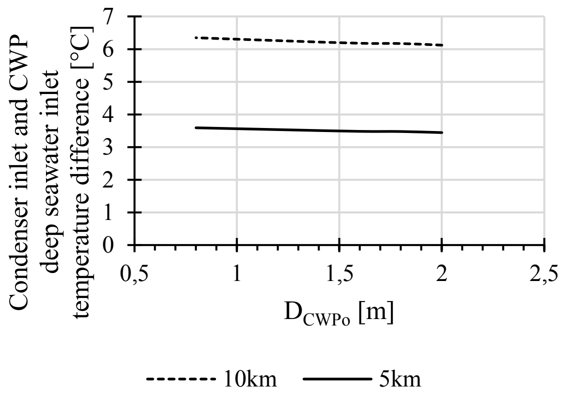

Fig. 3: CWP diameter versus the temperature difference between the condenser inlet and the CWP deep seawater inlet (ΔT), for two different CWP lengths of, 10km and 5km

4 Discussion and Conclusion

The results presented above, as anticipated, have shown that reducing the mass flowrate of the CWP yields a high increase in ΔT (as demonstrated in Figure 2 and Figure 3). The OTEC performance can be reduced by 29% based on the Carnot efficiency when increasing the CWP length from 5km to 10km.

Another notable result is that the CWP diameter does not have a significant effect on the temperature difference between the condenser inlet and CWP deep seawater inlet, and it is estimated at an approximate 3-4%.

Although the CWP diameter does not have a significant effect on the ΔT, as found in this study, it can be anticipated to have an effect on the pumping power required as well as on cost difference, since increasing the diameter, the related costs will increase. This remains a future goal to be studied next.

The interested reader could be further informed on OTEC aspects, such as the Energy, Economy, and Environment (referred to as the 3E aspects), through a comprehensive analysis of each aspect in the available literature, [12]. The energy efficiencies are reported to vary between 2.5% and 5.3%, while the Levelized Cost of Energy ranges from 0.05 to 0.45 USD/kWh. Barriers and technical restrictions, which play a crucial role in the decision-making process, are also addressed. OTEC case studies are additionally reported highlighting the type, positioning, and capacity of each project.

Declaration of Generative AI and AI-assisted Technologies in the Writing Process

During the preparation of this work the author used Grammarly for language editing. After using this tool/service, the authors reviewed and edited the content as needed and take full responsibility for the content of the publication.

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The authors equally contributed in the present research, at all stages from the formulation of the problem to the final findings and solution.

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Conflict of Interest

The authors have no conflicts of interest to declare.