Technical note

Maximum output of an OTEC power plant

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Abstract

This paper theoretically investigates the effects of the temperature and flowrate of cold seawater on the net output of an OTEC plant. Parameters of pipe length, pipe diameter, seawater depth, and the flowrate of seawater are considered. It shows that a maximum output of the net work exists at a certain flowrate of cooling seawater. The output work is higher for a larger ratio of warm to cold seawater flow rate. For a lower cold seawater temperature, the maximum net output and the corresponding required flow rate of cold seawater becomes larger. The pipe diameters corresponding to maximum net output increase with decreasing cold seawater temperature for fixed flow velocity in pipes. In addition, a larger maximum net output can be obtained by employing a higher temperature of the surface warm seawater.

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1. Introduction

Due to the remarkable economic growth in Taiwan, the demand for electric power increases and the quality of life and health for the people deteriorates. For this reason, more and more researches are devoted to investigating the inexhaustible source of non-pollution energy. The water along the east coast of Taiwan is in the route of Kuroshio’s major flow. The temperature difference between the surface water and that at 800 m deep is greater than...
Nomenclature

A  heat transfer area
\(c_p\)  specific heat
D  pipe diameter
F  correction factor, Eq. (1)
f  friction factor or fouling factor
h  heat transfer coefficient or enthalpy
g  acceleration of gravity
k  thermal conductivity
L  pipe length
m  mass flow rate
P  pressure
Pr  Prandtl number
Q  heat transfer rate
Re  Reynolds number
r  radius
s  entropy
T  temperature
U  overall heat transfer coefficient
V  velocity
\(v\)  specific volume
W  work
x  quality

Greek letters

\(\alpha\)  ratio of liquid ammonia to mixture in the evaporator
\(\beta\)  ratio of liquid ammonia to the whole mixture of the working fluid
\(\eta\)  efficiency
\(\nu\)  kinematic viscosity of working fluid
\(\rho\)  density

Subscripts

a  ammonia
c  condenser or condensing
ci  inlet of the condenser
c0  outlet of the condenser
cs  deep cold seawater
e  evaporator or evaporating
ei  inlet of the evaporator
eo  outlet of the evaporator
g  generator
h  hydraulic diameter of heat exchanger (condenser and evaporator)
20 °C most of the time through a year in this area. It makes this area possessing superior potential to develop OTEC (Ocean Thermal Energy Conversion) energy resources.

It is known that the deeper the cooling water is pumped from the sea, the more efficient will the OTEC plant work. Besides, an additional benefit of this system will be an enormous amount of pathogen-free and nutrient-rich cold seawater pumped up from the deep sea. This water can be used for the culture of high value marine-cultural cops, refrigeration, air-conditioning, and production of potable water (Dylan, 1995). However, the riskiest part of construction for the seawater system will be the cold water pipe. Laying pipes to greater depth may lead to higher pumping head requirements and needs higher costs (Avery and Wu, 1994).

The OTEC power plant uses the heat in the surface water of the tropical oceans to generate electricity and uses cold water drawn from deep sea to cool and liquefy the vapor emerging from the turbine. Since D’Arsenoval conceived the OTEC concept in 1881, and Laude first implemented the data and idea in 1930, there has been a wide range of interest in OTEC technology.

A lot of research efforts were engrossed in the performance improvement of an OTEC plant due to its low efficiency. Using freon 22 as a working fluid, Uehara et al. (1979) studied the performance of an OTEC plant. Later, Nakaoka and Uehara (1988a,b) conducted experiments to test the performance of an OTEC system with shell-and-plate-type evaporator and condenser. In addition, Ganic and Moeller (1980) carried out the performance study of an OTEC plant. Owens (1980) investigated the optimization of closed-cycle OTEC plants. Using ammonia as the working fluid, an optimization study by Power method was presented by Uehara and Ikegami (1990) for a closed-cycle OTEC plant. Both constant and variable turbine efficiencies were considered in their study. Later, Rabas et al. (1990) proposed a comprehensive study of the non-condensible gas removal system for a particular 10 MW hybrid power plant. Dividing the power plant into three major subsystems, Tseng et al. (1991) utilized an optimal design concept to find the best design for a complex and large-scale OTEC plant. Using the total heat transfer area of heat exchangers per net power as an objective function, an OTEC cycle with plate-type heat exchange would be the most suitable.
exchangers and ammonia as working fluid was investigated numerically (Uehara et al., 1996). Recently, based on plate exchangers with surface coating on the ammonia side, Abraham et al. (1999) have proposed the design of 1 MW closed-cycle floating OTEC plant.

The difference in seawater temperature between the surface and deep ocean off eastern Taiwan is around 20 °C for the entire year. In addition, the east coast of Taiwan drops off rapidly so that the OTEC plant can be constructed on shore and still have a cold seawater pipe of reasonable length. The purpose of this study is to develop a computerized calculation method for the performance analysis of an OTEC plant. Parameters of pipe length, pipe diameter, warm and cold seawater temperature, mass flow rate of seawater are considered to obtain the generated net power. For practical applications, the seawater temperature–pipe length relationship (Yeh et al., 1999) of Taitung county, located in eastern Taiwan, is employed in the analysis.

2. Mathematical analysis

For a given evaporator or condenser, a shell-and-tube heat exchanger, the heat transfer rate between the working fluid and seawater can be expressed as

\[ Q = UAF\Delta T_{lm} \tag{1} \]

where \( Q \) is the total heat transfer, \( U \) stands for overall heat transfer coefficient, \( A \) is entire surface area, \( F \) is a correction factor, and \( \Delta T_{lm} \) represents a log mean temperature difference of the evaporator or condenser. The overall heat transfer coefficient may be determined from knowledge of the hot and cold fluid convection coefficients, fouling factors, and appropriate geometric parameters. For a \( U \) based on the outside surface area of the heat exchanger, it may be written as

\[ U = \frac{1}{[(A_o/A_i)(1/h_w) + (A_o/h_s) + (1/h_m) + f_o]} \tag{2} \]

Note that subscripts \( i \) and \( o \) refer to inner and outer tube surfaces, which may be exposed to either the working fluid or the seawater, and \( f_o \) is the fouling factor of heat transfer surface. In Eq. (2), the heat transfer coefficient in the seawater side, \( h_w \), for the transition case, \( 2300 < Re_w < 10,000 \), can be calculated from the empirical expression (Gnielinski, 1976)

\[ h_w = \left( \frac{k_w}{D_h} \right) \frac{(Re_w - 1000)Pr_w(f_w/2)}{1 + 12.7(f_w/2)^{0.5}(Pr_w^{2/3} - 1)} \tag{3} \]

where

\[ f_w = [1.58 \ln(Re_w) - 3.28]^{-2} \tag{4} \]

and for the fully developed turbulent flow, i.e. \( Re_w > 10,000 \), the Petukhov correlation (Petukhov et al., 1973) is used

\[ h_w = \left( \frac{k_w}{D_h} \right) \frac{Re_w Pr_w(f_w/2)}{1.07 + 12.7(f_w/2)^{0.5}(Pr_w^{2/3} - 1)} \tag{5} \]
where $k_w, f_w, Re_w,$ and $Pr_w$ are the thermal conductivity, fanning friction factor, Reynolds number and Prandtl number of the seawater. In addition, $D_h$ denotes the hydraulic diameter of the heat exchanger.

The conductance, $h_s$, of the tube wall may be written in the form

$$h_s = \frac{2\pi k_t L}{\ln \left( \frac{r_o}{r_i} \right)}$$

where $k_t$ is the thermal conductivity, $L$ is the length, $r_o$ and $r_i$ are the outer and inner radii of the tubes in the heat exchangers.

The heat transfer coefficient, $h_m$, in the side of working fluid has two different values depending on condensation or evaporation. In condenser, it is calculated as

$$h_m = 0.725 \left[ \frac{(\rho_l - \rho_v)g h_{fg} k_v}{D_h v(T_m - T_c)} \right]^{0.25}$$

where $\rho$, $k$ and $v$ are density, thermal conductivity and kinematic viscosity of working fluid at condensing temperature. Also, the average temperature outside the tubes is denoted as $T_m$, the condensing temperature is abbreviated as $T_c$, and $h_{fg}$ is the enthalpy difference between saturated vapor and saturated liquid of the working fluid. As for an evaporator, it is given as

$$h_m = 0.62 \left\{ \frac{(\rho_l - \rho_v)g [h_{fg} + 0.4 C_{pv}(T_m - T_c)]k_v}{D_h v(T_m - T_c)} \right\}^{0.25}$$

Note that the above thermal properties are obtained at evaporating temperature, and $T_e$ is the evaporating temperature of the working fluid.

The total heat flow of the evaporator can be evaluated as

$$Q_e = m_{ws} c_{pw} (T_{ei} - T_{eo})$$

In Eq. (9), $m_{ws}$ and $c_{pw}$ are mass flow rate and specific heat of the warm (surface) seawater temperatures, and $T_{ei}$ and $T_{eo}$ are the inlet and outlet seawater temperatures of the evaporator, respectively. From energy balance, the quantity of circulating working fluid can be obtained as

$$m_a = \frac{Q_e}{(h_{eo} - h_{ei})}$$

where $h_{eo}$ and $h_{ei}$ are the enthalpies of working fluid at the outlet and inlet temperatures of the evaporator.

To simulate the real cases, it is assumed that a certain portion of the working fluid returns to the storage tank and the rest goes to the generator and enters the condenser. Suppose that the working fluid changes from saturated vapor into saturated liquid isothermally. The heat transfer rate of the condenser can then be calculated as

$$Q_c = m_a (h_{ei} - h_{eo})$$
or alternatively

\[ Q_c = m_c c_{pw} (T_{co} - T_{ci}) \]  \hspace{1cm} (12)

The output of the turbine work can be evaluated by assuming that the saturated vapor of the working fluid expands isentropically into low pressure’s mixture of saturated vapor and liquid. In this case, the quality of ammonia vapor leaving the turbine can be obtained as

\[ x = (s_{to} - s_{co})/(s_{ci} - s_{co}) \]  \hspace{1cm} (13)

where \( s_{to} \) is the entropy of the working fluid at the exit of the turbine, and \( s_{co} \) and \( s_{ci} \) are the entropies of saturated liquid and saturated vapor at condensing temperature. The enthalpy of the working fluid vapor at the outlet of the turbine therefore becomes

\[ h_{to} = (1 - x)h_{ci} + xh_{co} \]  \hspace{1cm} (14)

In the above equation, \( h_{ci} \) and \( h_{co} \) are the enthalpies of saturated liquid and saturated vapor at the temperature of condensation. Since the enthalpies at the entrance and exit of the turbine are known, the total electric work resulting from turbine can be calculated as

\[ W_t = m_a (h_{eo} - h_{to})\eta_t \eta_g \]  \hspace{1cm} (15)

where \( h_{eo} \) is the enthalpy of saturated vapor at evaporating temperature, and \( \eta_t \) and \( \eta_g \) are efficiencies of turbine and generator, respectively.

For an isentropic process, the power consumption of the working fluid circulating pump can be expressed as

\[ W_a = m_a v_a [(P_{ei} - P_{eo}) + (\alpha + \beta)(P_{ei} - P_s)]/\eta_p \]  \hspace{1cm} (16)

In Eq. (16), \( v_a \) is the specific volume of saturated liquid working fluid at condensing temperature, \( P_{ei} \) and \( P_{eo} \) are the pressure of working fluid at evaporating and condensing sides, respectively, \( P_s \) is the pressure of storage tank of working fluid and \( \eta_p \) is the efficiency of circulating pump. In addition, \( \alpha \) is the ratio of liquid ammonia to mixture as the working fluid flow through the evaporator whereas \( \beta \) is ratio of liquid ammonia, which did not evaporate in the evaporator, to the whole mixture of the working fluid. In this study, both \( \alpha \) and \( \beta \) are assumed to be 0.125.

The OTEC system utilizes cold seawater in the deep sea for cooling and warm seawater from the sea surface as a heating source. Therefore, some of the generating electric works are required for pumping water from sea to the plant. The power consumption of the pump is mainly to overcome the friction loss as seawater goes through the pipes, fittings, and heat exchangers. The summation of all the pressure losses caused by the friction between seawater and pipe wall or fitting can be given as

\[ \Delta P_s = \sum_{i=1}^{n} f_i V^2 L_i/(2gD_i), \quad i = 1, 2, 3, \ldots, n \]  \hspace{1cm} (17)

where \( f_i \) is the friction factor of pipe or fitting, \( V \) is the velocity of seawater, \( L_i \) is the pipe length, and \( D_i \) is the diameter of pipe. Considering the core friction, the pressure drop
associated with seawater flow through an evaporator or a condenser is given by

\[ \Delta P_h = \rho V^2 \left( \frac{f_h}{2} \right) \left( \frac{D_h}{A_h} \right) \left( \frac{L_h}{g} \right) \]  \hspace{1cm} (18)

In Eq. (18), \( \rho \) and \( V \) are the density and velocity of seawater, \( f_h, D_h, A_h, \) and \( L_h \) are the friction factor, diameter, cross-sectional area and length of the tubes used in evaporators or condensers.

The total power required for the surface warm as well as the deep cold seawater pump then becomes

\[ W_s = (\Delta P_s + \Delta P_h) m_s g/\eta_s \] \hspace{1cm} (19)

where \( m_s \) is the flow rate of the seawater and \( \eta_s \) signifies the efficiency of seawater pump. The net power \( W_n \) is calculated from

\[ W_n = W_t - W_{ws} - W_{cs} - W_p \] \hspace{1cm} (20)

In the above equation, \( W_t \) is the generated turbine power, \( W_{ws} \) is the warm seawater pumping power, \( W_{cs} \) is the cold seawater pumping power, and \( W_p \) is the pumping power of the working fluid.

3. Solution procedures

In this study, several parameters are known and are listed as follows:

1. the temperature of the surface warm seawater \( T_{ei} \) and deep cold seawater \( T_{ci} \).
2. the diameter \( D \), length \( L \), wall thickness \( t \) and number of tubes \( N \) used in evaporators or condensers.
3. the efficiencies of working fluid circulating pump, seawater pumps, and turbine generator.
4. friction factor, fouling factor, and liquid ratios of the working fluid.

With the above given values, the total surface area of heat exchangers and some dimensionless parameters can be calculated. Subsequently, the heat transfer coefficients inside and outside the tubes in evaporator are obtained. The overall heat transfer coefficient is then determined from Eq. (2). This analysis initiates with the assumption of the outlet temperature of warm seawater in evaporator. With the aid of Eq. (1), the heat transfer rate of the evaporator can be computed. Also, from Eq. (9), the heat dissipated by warm seawater can be evaluated. An iteration method with updating \( T_{eo} \) continues until a specified tolerance of 0.001 (relative error) is met. As the heat duty of evaporator is calculated, the quantity of working fluid can be obtained from Eq. (10). In a similar manner, the heat flow of the condenser and the outlet of cold seawater are found from Eqs. (1) and (12). In addition, the pressure drops of pipes, fitting, and heat exchanger can be determined from Eqs. (17) and (18). The power consumed by the seawater pumps and working fluid pump can then be calculated. The net output of the turbine work is finally found from Eq. (20). Detailed flow chart of the calculation is given in Fig. 1.
Fig. 1. Flow chart of calculation.
4. Results and discussion

In this study, it is assumed that the OTEC plant is situated in Taitung county, east coast of Taiwan. For the first part, the surface warm water temperature is 25 °C and the cold seawater temperatures are specified as 5–8 °C. In the latter part, the cold seawater temperature is assumed 5 °C whereas the warm seawater temperatures are 20, 25, 30, and 35 °C, respectively. The relationship between deep cold water temperatures and actual pipe lengths are shown in Table 1. The OTEC system converts heat energy into electricity by using the temperature difference between water at the surface of the ocean and cold water of the depths. The larger the temperature difference is, the higher the plant efficiency will be. However, in practical applications, it is observed that the lower the seawater temperature is, the longer the piping is needed in Table 1. The increase in the turbine work is offset by the pumping power of cold seawater. Thus, a computerized calculation method is developed to find the appropriate design conditions with which maximum net output power is produced. Table 2 lists the design parameters used in computation. Ammonia is selected as the working fluid. Both evaporator and condenser of this plant are shell-and-tube exchangers, in which the tubes are made of stainless steel. The heat transfer area of the condenser is the same as that of the evaporator. Although various seawater pipe diameters are given for calculation, identical pipe diameters for evaporators and condensers are used in each run. For practical applications, the surface warm seawater pipe is much shorter than the deep cold seawater pipe.

Table 1
Depth–temperature relationship and actual pipe length used in this study

<table>
<thead>
<tr>
<th>Depth of sea (m)</th>
<th>Temperature (°C)</th>
<th>Horizontal distance (m)</th>
<th>Actual pipe length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>500</td>
<td>8</td>
<td>2900</td>
<td>3200</td>
</tr>
<tr>
<td>600</td>
<td>7</td>
<td>3100</td>
<td>3400</td>
</tr>
<tr>
<td>700</td>
<td>6</td>
<td>3600</td>
<td>4000</td>
</tr>
<tr>
<td>800</td>
<td>5</td>
<td>4100</td>
<td>4500</td>
</tr>
</tbody>
</table>

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Table 2
Parameters of the proposed OTEC plant

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Warm seawater pipe length (m)</td>
<td>250</td>
</tr>
<tr>
<td>Thermal conductivity of heat exchangers (W/m K)</td>
<td>14</td>
</tr>
<tr>
<td>Turbine efficiency (%)</td>
<td>89</td>
</tr>
<tr>
<td>Seawater pump efficiency (%)</td>
<td>85</td>
</tr>
<tr>
<td>Working fluid pump efficiency (%)</td>
<td>85</td>
</tr>
<tr>
<td>Generator efficiency (%)</td>
<td>96</td>
</tr>
<tr>
<td>Warm seawater inlet temperature (°C)</td>
<td>20, 25, 30 and 35</td>
</tr>
<tr>
<td>Cold seawater inlet temperature (°C)</td>
<td>5, 6, 7 and 8</td>
</tr>
<tr>
<td>Seawater pipe diameter (Cold and warm) (m)</td>
<td>0.5, 1, 1.5 and 2</td>
</tr>
</tbody>
</table>
4.1. Fixed pipe diameters

Fig. 2 depicts the dependence of powers on the mass flow rate of cold seawater for $T_{ci} = 8\,^{\circ}C$ and $D_{cs} = 1\,m$. On the whole, $W_t$, $W_n$, $W_{cs}$, $W_{ws}$, and $W_p$ increase with $m_{cs}$. Note that the pumping power of the working fluid and the warm seawater pumping power in an OTEC plant system play an insignificant role in the determination of net power. This is because the required pipe lengths are smaller for warm seawater and working fluid in this system. The pressure loses caused by the friction between fluids and pipe walls or fittings are thus smaller. For a fixed pipe diameter, the mass flow rate of the cooling water increases with the flow velocity of cold seawater. Since the pumping power is proportional to the third power of $V_{cs}$, $W_{cs}$ rises very quickly with $m_{cs}$. For this reason, a maximum net power can be obtained at an appropriate mass flow rate of seawater. A further increase in $m_{cs}$, the total sum of $W_{cs}$, $W_{ws}$, and $W_p$ may be larger than $W_t$ and $W_n$ will even become negative. In this case, the generated turbine power is not enough to sustain the total pumping powers of the seawater and working fluid. The dependence of powers on the mass flow rate of cold seawater for $T_{ci} = 5\,^{\circ}C$ and $D_{cs} = 1\,m$ is shown in Fig. 3. Viewing the two figures, similar phenomenon is observed. For the same system, as the cooling seawater temperature is changed from 8 to 5 $^{\circ}C$, apparent increases in $W_t$, $W_n$, $W_{cs}$, $W_{ws}$, and $W_p$ can be observed. Also, note that the increasing rate of the working fluid pumping power for $T_{ci} = 5\,^{\circ}C$ is larger than that for $T_{ci} = 8\,^{\circ}C$. This is due to the fact that the evaporating and condensing temperatures of the working fluid are fixed, for energy balance a larger mass flow rate of the working fluid is needed for a lower temperature of seawater temperature. The working fluid thus has an obvious increase in pumping power especially at a larger $m_{cs}$. In addition, the maximum net power of the OTEC plant increases from 112 kW for $T_{ci} = 8\,^{\circ}C$ to 248 kW for $T_{ci} = 5\,^{\circ}C$ whereas the corresponding seawater flow rate merely
increases from 2800 to 3200 tons/h. It is interesting to see the effects of $T_{ci}$ on $W_n$ for $D_{cs} = 1$ m and $T_{ci} = 25 \, ^\circ C$. Fig. 4 plots the dependence of $W_n$ on $m_{cs}$ for $T_{ci} = 5, 6, 7$ and $8 \, ^\circ C$ and for $D_{cs} = 1$ m and $T_{ci} = 25 \, ^\circ C$. In this figure, $(W_n)_{max} = 112, 160, 204$, and $248 \, kW$ for $T_{ci} = 8, 7, 6$, and $5 \, ^\circ C$, respectively. Although apparent increases in net powers are obtained, no pronounced increase in the corresponding $m_{cs}$ is found. This can be explained

Fig. 3. Dependence of $W_t$, $W_n$, $W_{cs}$, $W_{ws}$, and $W_p$ on $m_{cs}$ for $T_{ci} = 5 \, ^\circ C$.

Fig. 4. Dependence of $W_n$ on $m_{cs}$ for $T_{ci} = 5, 6, 7$, and $8 \, ^\circ C$. 
that a larger temperature difference between heat source and sink results in a high efficiency of OTEC plant. The loci of \((W_n)_{\text{max}}\) is also included in this figure. Fig. 5 displays the dependence of \(W_n\) on \(m_{cs}\) for \(D_{cs} = 0.5, 1, 1.5,\) and \(2\) m and \(T_{ci} = 5\) °C. By increasing \(V_{cs}\) of cooling seawater, the mass flow rate can be increased for a fixed pipe diameter. It is shown that the maximum net power increases from 63 kW for \(D_{cs} = 0.5\) m to 1334 kW for \(D_{cs} = 2\) m. Thus, it is effective to increase \((W_n)_{\text{max}}\) by utilizing a larger pipe. However, the required corresponding \(m_{cs}\) to \((W_n)_{\text{max}}\) also increases enormously. The dependence of \((W_n)_{\text{max}}\) on \(T_{ci}\) for \(D_{cs} = 0.5, 1, 1.5,\) and \(2\) m is given in the lower portion of Fig. 6. As a whole, \((W_n)_{\text{max}}\) decreases with \(T_{ci}\). At a specified \(T_{ci}\), the maximum net power is larger for a larger pipe of seawater. The required corresponding mass flow rate of cooling seawater to \((W_n)_{\text{max}}\) is shown in upper portion of Fig. 6. No pronounced variation of \(m_{cs}\) is observed for \(T_{ci}\) below 7 °C. The mass flow rate of seawater tends to decrease for \(T_{ci}\) greater than 7 °C. Nevertheless, the maximum net power becomes lower for a higher \(T_{ci}\). This figure also sets up a guideline for piping design. It is suggested that the mass flow rate of cooling seawater should not exceed the proposed limit for a specified pipe diameter used otherwise \(W_n\) will decrease with increasing \(m_{cs}\). This can be understood from the observation of Figs. 2–5. Fig. 7 describes the effects of \(m_{cs}\) on \(W_n\) for various \(T_{ci}\), when \(m_{cs}\) is lower than the design limit of \(m_{cs}\), i.e. the mass flow rate of seawater corresponding to \((W_n)_{\text{max}}\). As expected, the net power decreases with increasing \(T_{ci}\). In addition, \(W_n\) is larger for a larger \(m_{cs}\) at a given \(T_{ci}\).

There is still an alternative way to increase net power if the mass flow rate of cold seawater is fixed. Fig. 8 shows the dependence of \(W_n\) on \(m_{ws}/m_{cs}\) for \(T_{ci} = 5\) °C and \(D_{cs} = 2\) m. It is apparent that \(W_n\) increases with \(m_{ws}/m_{cs}\). From Figs. 7 and 8, \(W_n\)
increases from 500 kW for $m_{ws} = 4000$ tons/h to 540 kW for $m_{ws} = 4400$ tons/h. This is because the evaporating temperature rises due to the increase in the mass flow rate of warm seawater. The performance of the OTEC plant is, therefore, improved.

Fig. 6. The effects of seawater pipe diameter on $(W_n)_{\text{max}}$ and $m_{cs}$.

Fig. 7. Dependence of net power on the temperature of cold seawater.
In any case, the expense in increasing \( m_{ws} \) is lower than \( m_{cs} \) because the pipe for warm seawater is shorter than that of cold seawater. The pumping power, by all means, for the former is smaller due to lower friction loss.

4.2. Fixed flow velocities

To increase the mass flow rate, it can be achieved by accelerating the flow velocity for a fixed pipe diameter; or alternatively, by increasing the pipe diameter for a fixed flow velocity. For convenience, it is assumed that \( V_{cs} = 1 \text{ m/s} \) herein. Fig. 9(a) and (b) give the dependence of \( W_n \) on \( D_{cs} \) and \( m_{cs} \) for \( T_{ei} = 25 \text{ °C} \) and \( V_{cs} = 1 \text{ m/s} \). Comparing Fig. 9(b) with Fig. 4, the maximum net power turns out to be a little lower for this case. It is clarified that the data in Fig. 4 is presented under the condition of \( D_{cs} = 1 \text{ m} \) and the flow velocities are all smaller than \( V_{cs} = 1 \text{ m/s} \). Also, it should be emphasized that flow velocity plays a dominant role in pressure loss. The pumping power is thus higher for a larger \( V_{cs} \). In Fig. 4, the seawater velocities corresponding to the maximum net powers are 0.38 \( \text{m/s} \) for \( T_{ei} = 8 \text{ °C} \), 0.57 \( \text{m/s} \) for \( T_{ei} = 7 \text{ °C} \), 0.77 \( \text{m/s} \) for \( T_{ei} = 6 \text{ °C} \), and 0.97 \( \text{m/s} \) for \( T_{ei} = 5 \text{ °C} \), respectively. From Fig. 9(a), it is shown that \( (W_n)_{max} \) is obtained at a larger pipe diameter for a colder cooling seawater used for \( V_{cs} = 1 \text{ m/s} \).

It is desirable to learn that the effect of \( T_{ei} \) on \( W_n \) for \( V_{cs} = 1 \text{ m/s} \). Fig. 10(a) and (b) present the dependence of \( W_n \) on \( D_{cs} \) and \( m_{cs} \) for \( T_{ei} = 5 \text{ °C} \). The lower the temperature of warm seawater is, the smaller the seawater pipe diameter is needed to obtain the maximum net power. Nevertheless, \( (W_n)_{max} \) is larger for a higher \( T_{ei} \).
Fig. 9. Dependence of $W_n$ on $D_{cs}$ and $m_{cs}$ for $T_{ci}=5$–8 °C.

Fig. 10. Dependence of $W_n$ on $D_{cs}$ and $m_{cs}$ for $T_{ei}=20$, 25, 30, and 35 °C.
5. Conclusions

In this study, a theoretical approach is proposed to compute the output powers of an OTEC plant. Assuming either fixed pipe diameter or fixed flow velocity, a maximum generated net power is obtained at a certain mass flow rate of the cold seawater. From the foregoing results, it can be concluded as follows:

1. For fixed $D_{cs}$ and $T_{ei}$, the lower the temperature of cold seawater is, the higher the maximum net power is obtained.
2. For fixed $T_{ei}$ and $T_{ci}$, a larger $(W_n)_{\text{max}}$ is found for a larger diameter of seawater pipe.
3. For a fixed $m_{cs}$, the generated net power is increased by increasing $m_{ws}$.
4. For fixed $V_{cs}$ and $T_{ei}$, the maximum net power increases with the decrease of $T_{ci}$. The colder the seawater is, the larger the pipe diameter is used to obtain maximum net power.
5. For fixed $V_{cs}$ and $T_{ci}$, the maximum net power and its corresponding pipe diameter increase with the temperature of surface warm seawater.

References