

Proposal of a Solar Thermal Power Plant at Low Temperature Using Solar Thermal Collectors

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Abstract

To this day, only two types of solar power plants have been proposed and built: high temperature thermal solar one and photovoltaic one. It is here proposed a new type of solar thermal plant using glass-top flat surface solar collectors, so working at low temperature (*i.e.*, below 100°C). This power plant is aimed at warm countries, *i.e.*, the ones mainly located between -40° and 40° latitude, having available space along their coast. This land based plant, to install on the seashore, is technologically similar to the one used for OTEC (Ocean Thermal Energy Conversion). This plant, apart from supplying electricity with a much better thermodynamic efficiency than OTEC plants, has the main advantage of providing desalinated water for drinking and irrigation. This plant is designed to generate electricity (and desalinated water) night and day and all year round, by means of hot water storage, with just a variation of the power delivered depending on the season.

Keywords

Power Plant, Solar Thermal, Solar Energy, Renewable Energy, Low Temperature, Solar Collectors, Electric Power Generation, Desalinated Water

1. Introduction

1.1. Goal of the Presentation

The goal of this article is to describe a new type of power plant, taking its source in the difference in temperatures between hot water heated, up to about 77°C, by glass-top flat surface solar collectors and the surface sea (or ocean) water. So a pipe is used to extract sea water from low depth. Another pipe forces the water back to the sea near the surface.

The power generation plant itself is similar to an OTEC plant.

In **Figure 1**, a simplified diagram of the plant layout is presented.

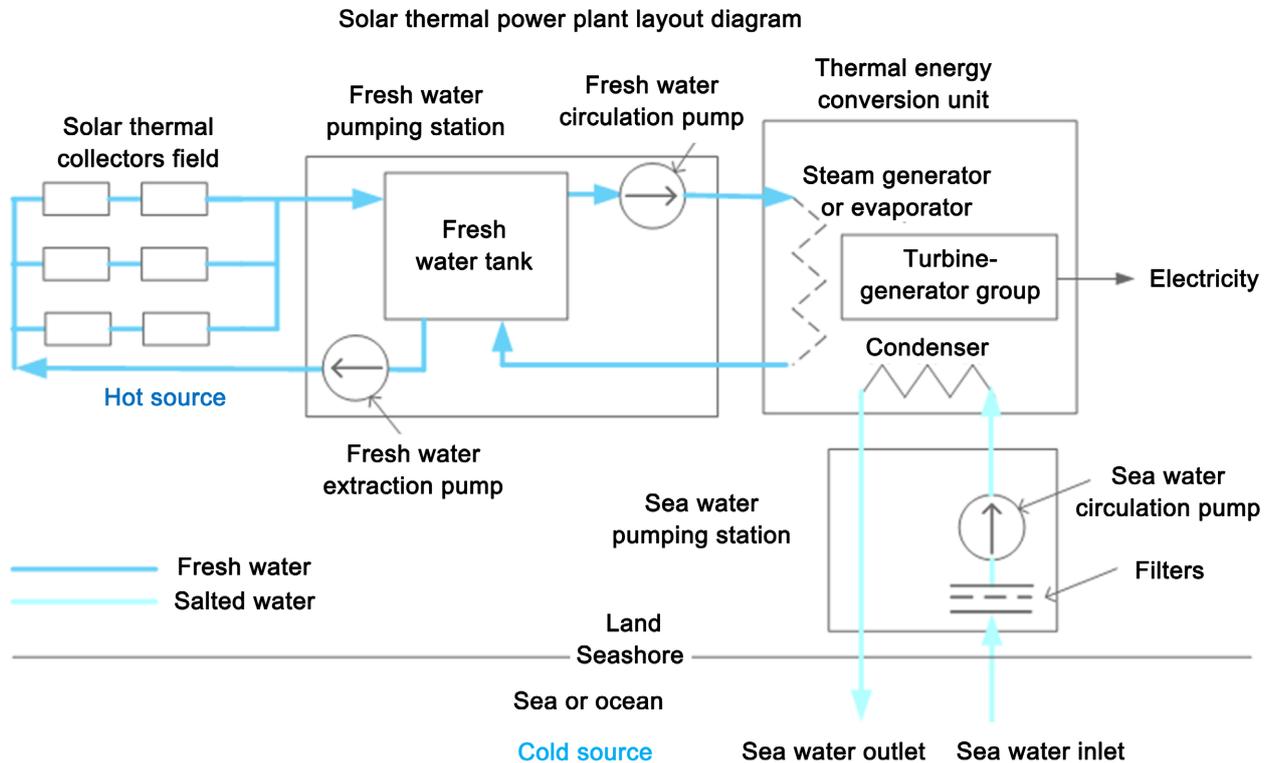


Figure 1. Solar thermal power plant layout diagram.

The plant is installed on the seashore. A large ground surface must be available to install solar surface collectors. A road and a power grid close to the plant must be available. It will work in countries where the global horizontal irradiance is good, *i.e.*, countries located at low latitude (let's say between -40° to 40°) and having a high clearness index.

A close domain is the "Ocean Thermal Energy Conversion" (OTEC), for which energy is extracted between the warm surface water (about 26°C) and the cold water (about 4°C) at about 1000 m depth. The final difference of temperatures for the thermodynamic cycle is around 11°C , whereas for the way proposed here, the mean difference is higher and around 42°C .

SI units, multiples ("bar" for example) and sub-multiples ("mm" for example) are only used. "bar" is often used for the pressure rather than Pa, with $1 \text{ bar} = 1\text{E}5 \text{ Pa}$.

In this document, all pressure values are absolute ones (*i.e.*, no gauge pressures) and all temperatures are, by default, in $^\circ\text{C}$ (Celsius degrees).

This domain could be called "WATEC" for "Water—Atmosphere Thermal Energy Conversion", by reference to "OTEC".

1.2. Method of the Work

Step 1: The proposed power plant is first described, at the level of principle, in Section 2.

Step 2: The two main cycles (Rankine and Claude) are studied in Section 3.

The goal of this chapter is to determine, with a simple equation, the net efficiency η_{net} (net electric power/heat transfer power) of both cycles, according to the hot and cold sources' temperatures.

Step 3: For the solar collectors (Section 4), it is determined the thermal collector efficiency η_{co} (thermal power generated by the collectors/horizontal irradiation power received by the collectors).

Step 4: From these two efficiencies (η_{net} and η_{co}), in Section 5 it is determined the ideal mean hot temperature at collectors output (Thi). Finally, it is determined the mean net electric power delivered by 1 m² of a collector.

Step 5: From the mean values calculated before, it is described and assessed:

- The way to produce continuously by means of water storage (Section 6),
- The way to configure solar collectors (Section 7),
- The way to produce desalinated water (Section 8).

Step 6: From all the results found in previous chapters, in Section 9, it is assessed an example of the solar power plant whose goal would be to generate a net electric power of 2.9 MW and 350 tons of desalinated water daily.

Moreover, it is proposed, in Appendix, a small utility program designed to roughly estimate the interest of such plant for a given place.

1.3. List of the Notations and Variables Used in the Article

Notations

“§”	for “subsection”.
“≈”	for “about”.
“~”	for “proportional”.
“[<i>x</i>]”	for “reference number <i>x</i> ”, the references list being available at the end of this article.
“ <i>f</i> (<i>x</i>)”	for “function of <i>x</i> ”.

List of the main variables:

ΔTh	Difference between Thi and $T3$ (fixed to 5.46°C).
ΔTc	Difference between $T4$ and Tci (fixed to 6.75°C).
ΔTco	Difference between Tm and Ta .
η_{net}	Net thermodynamic efficiency (= Pen/Ph).
η_{co}	Thermal collector efficiency.
Cl_i	Clearness index (from 0 to 1, in %).
Cp	Specific heat capacity at constant pressure in J/(kg·°K).
Dp	Daylight period (hour).
GHI	Global horizontal irradiance in kWh/m ² .
h	Hour.
I_{er}	Horizontal irradiance at the top of the atmosphere (W/m ²).
I_r	Horizontal irradiation power (W/m ²).
Lv	Heat of vaporization (J/kg).
Mt	Mass of water in the tank (in kg, with $Mt = Mt1 \times Sco$).
$Mt1$	Mass of water in the tank, for 1 m ² of heat-absorbing surface col-

lector (in kg/m^2).

P_1 Liquid fluid pressure at the condenser outlet.

P_2 Liquid fluid pressure at the compressor (circulation pump) outlet.

P_3 Steam pressure at the turbine inlet.

P_4 Saturated fluid pressure at the turbine outlet.

P_{eadu} Electric power consumed by the desalination unit (in W, $P_{eadu} = P_{eadu1} \times S_{co}$).

P_{eadu1} Electric power consumed by the desalination unit, for 1 m^2 of heat-absorbing surface collector (W/m^2).

P_{ea} Difference between the rough and the net electric power (in W, $P_{ea} = P_{erough} - P_{en}$).

P_{en} Net electric power provided to the grid net (in W, $P_{en} = P_{en1} \times S_{co}$).

P_{en1} Net electric power provided to the grid net, for 1 m^2 of heat-absorbing surface collector (in W/m^2).

P_{erough} Rough electric power generated by the alternator (W).

P_h Heat power consumed from the hot source by the thermal energy conversion unit (in W, $P_h = P_{h1} \times S_{co}$).

P_{h1} Heat power consumed from the hot source by the thermal energy conversion unit, for 1 m^2 of heat-absorbing surface collector (in W/m^2).

P_{hav} Mean available heating power (in W, $P_{hav} = P_h + P_{hcodu} = P_{hav1} \times S_{co}$).

P_{hav1} Mean available heating power, for 1 m^2 of heat-absorbing surface collector (in W/m^2).

P_{hco} Heat power generated by a heat-absorbing surface collector S_{co} (in W, $P_{hco} = P_{hco1} \times S_{co}$).

P_{hco1} Heat power generated by 1 m^2 of heat-absorbing surface collector (in W/m^2).

P_{hcodu} Heat power consumed from the hot source by the desalination unit (in W, $P_{hcodu} = P_{hcodu1} \times S_{co}$).

P_{hcodu1} Heat power consumed from the hot source by the desalination unit, for 1 m^2 of heat-absorbing surface collector (in W/m^2).

Q_{mdcw} Daily production of condensed water (in $\text{kg}/\text{day}/\text{m}^2$).

R_{cw} Ratio between the heating power used for the desalination unit (P_{hcodu}) and the available heat power (P_{hav}).

S_{co} Heat-absorbing surface collector (m^2).

T_1 Liquid fluid temperature at the condenser outlet.

T_2 Liquid fluid temperature at the compressor (circulation pump) outlet.

T_3 Steam temperature at the turbine inlet.

T_4 Saturated fluid temperature at the turbine outlet.

T_a Mean ambient air temperature for the daylight period.

T_{ci} Cold temperature (at the condenser inlet) coming from the cold source (ocean).

T_{co} Temperature (at the condenser outlet) returning to the cold source (ocean).

T_{hi} Hot temperature (at the steam generator or evaporator inlet) coming from the hot source.

T_{ho} Temperature (at the steam generator or evaporator outlet) returning to the hot source.

T_m Mean water temperature inside the collectors.

V_t Volume of water in the tank (in m^3 , with $V_t = V_{fl} \times S_{co}$).

V_{fl} Volume of water in the tank, for $1 m^2$ of heat-absorbing surface collector (in m^3/m^2).

The other variables are explained locally, but their mantissa (first letter) is, in general, generic:

Δ (Delta) for a difference.

ρ for a mass density in kg/m^3 .

η for an efficiency (between 0 and 1).

m or M for a mass (kg).

P for a pressure in Pa or in bar.

P_x for a power (W) or a power by surface unit (W/m^2), with the “ x ” relative to the source, as it can be a heat transfer rate from a hot or a cold source, or an electric power.

Q_m for a mass flow rate (kg/s).

Q_v for a volume flow rate (m^3/s).

s for specific entropy in $J/(^\circ K \cdot kg)$.

S for a surface area (m^2).

t for time (s).

T for a temperature in $^\circ C$ (note that $^\circ K = ^\circ C + 273.15$).

v for the specific volume in m^3/kg .

Several suffixes are used in the variables naming:

l for $1 m^2$ of heat-absorbing surface collector (for power variables only).

a for “auxiliaries”.

av for “available”.

c for “cold” (sea water through the condenser).

co for (solar thermal) “collectors”.

cw for “condensed water” (*i.e.* desalinated water).

d for “daily”.

du for “desalination unit”.

e for “electric”.

h for “hot”.

i for “input” or “inlet”.

m for “mass” or for “mechanic”.

n for “net”.

o for “output” or “outlet”.

- r for “rough”.
- s for “solar” (thermal collectors field).
- t for (fresh water) “tank”.
- v for “volume”.

2. Principle and General Working

2.1. Generalities and Concerned Areas

In different zones of the world, there are relatively great differences in temperature between the ambient air and the sea or ocean surface water for coastal areas. Moreover, if the latitude is low and the luminosity is high, the global horizontal irradiance is good, *i.e.*, let's say superior or equal to 1500 kWh/m² per year. See [1] for estimates of the global horizontal irradiance in different places of the world. A priori, in many zones of the world, conditions would be favorable for such solar thermal power plants, *i.e.*, part of Africa, Oceania and South America, Central America, south of California and South Asia coasts.

A high mean ambient air permits to heat water by means of solar collectors with high efficiency.

Due to the difference in temperatures between the hot water at solar collectors output and the ocean temperature, it can be envisaged to set up plants generating electricity through a thermodynamic cycle.

See **Figure 2**, for the principle.

Below, in **Table 1**, is an example taken from Nouakchott, capital of Mauritania where the global horizontal irradiance is very high, between 2150 and 2200 kWh/m² per year according to [2].

The ambient air and ocean temperatures (in °C) are issued from [3] and [4]. This example is very favorable because the mean ambient air temperature is high and the ocean is relatively cold.

Note that the surface ocean temperature and its temperature amplitude throughout the year depend on the latitude but also on the oceanic currents (cold or warm). Of course coasts swept by cold current are favorable. See the **Figure 3** below.

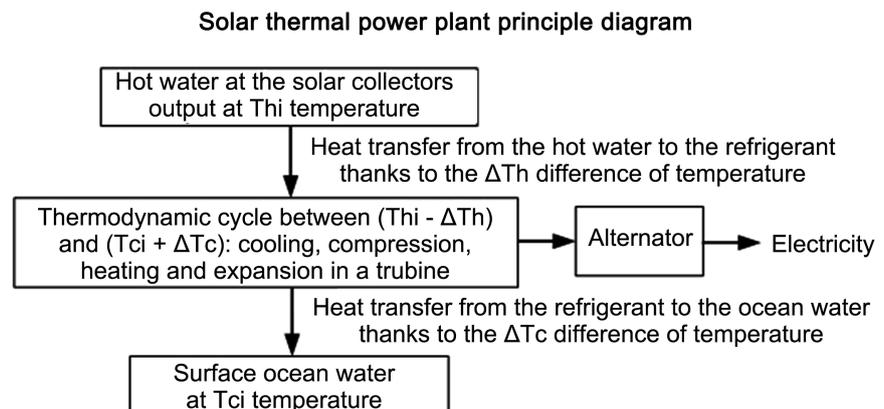


Figure 2. Solar thermal power plant principle diagram.

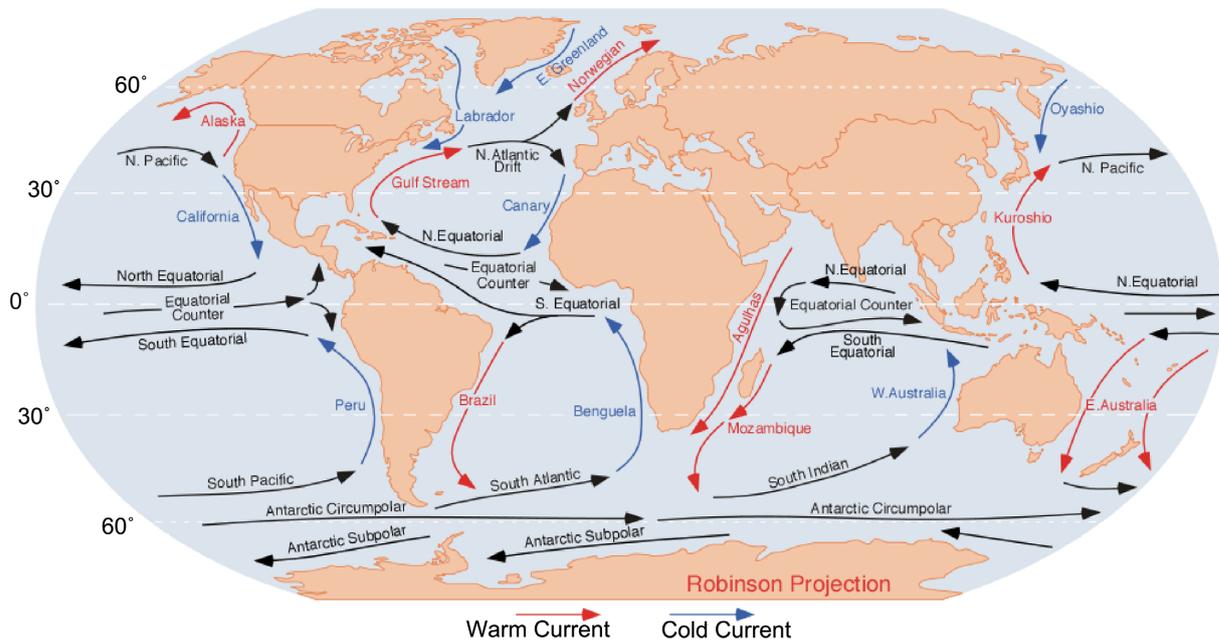


Figure 3. Ocean surface currents.

From Wikipedia, Author Dr. Michael Pidwirny, Public domain, U.S. government publication.

Table 1. Temperatures at Nouakchott.

Month	J	F	M	A	M	J	J	A	S	O	N	D
Max air temperature	29	29	31	32	33	33	32	32	34	35	32	29
Min air temperature	18	19	20	21	23	24	25	27	28	28	24	20
Ocean temperature	18.3	18.0	18.5	18.6	19.1	21.8	25.6	27.5	28.3	25.9	22.4	20.1
Difference: Tmax air-Tocean	11	11	12	13	14	11	6	5	6	9	10	9

From now on, this article will concentrate on the Nouakchott area, which is very favorable for such a type of power plant.

2.2. General Working

Based on **Figure 1** where a simplified diagram of the plant layout is presented, the general working principle is the following:

- Fresh water at about 71.2°C extracted from the fresh water tank is heated by the solar irradiation inside the solar collectors. Then, the heated water at about 76.7°C returns to the fresh water tank.

This circuit is only in operation for the daylight period (cf. Section 6).

- The fresh water at 76.7°C is pumped towards the thermal energy conversion unit where it leaves heat to the refrigerant through a steam generator or an evaporator according to the cycle used (cf. Section 3). It is the “hot source” of the thermodynamic cycle.
- In the thermal energy conversion unit, due to this hot source, steam is

produced which makes work a turbine-generator group generating electricity. The steam is condensed in a condenser cooled by the cold sea water (“cold source”).

- The sea water is filtered and pumped towards the condenser to cool the steam. Afterwards, the sea water returns to the sea. The sea water is pumped at a depth of about 20 or 30 m so to have an approximate constant sea temperature and to be sufficiently below the swell trough. The sea water is rejected just below the surface but far away from the water intake (at least 200 m, according to the current) to avoid to re-pump the rejected sea water.

3. Types of Thermal Energy Conversion Units and Determination of the Thermodynamic Efficiency of These Units

3.1. Generalities about the Thermal Energy Conversion Units

See the location of the “Thermal energy conversion unit” in **Figure 1**. It works between the hot and the cold source to provide mechanical work which is transformed into electricity.

There are two types of units used in OTEC technology and used here:

- One uses ammonia as a working fluid and works at relatively high pressure (*i.e.*, around 8 bar) in a closed cycle. This cycle is called “Rankine” or “Rankine without superheating” or “Anderson”. It will be called “Rankine” in what follows.
- The other uses sea water as a working fluid and works in an open cycle at low pressure (*i.e.*, under vacuum). This cycle is called “Claude”. Note that for the proposed plant, the working fluid (refrigerant) will be fresh water and not sea water, this to avoid salt deposit inside the solar collectors.

For information about OTEC, see [5] and/or [6].

The goal here is to develop a simple equation to determine the net thermodynamic efficiency “ η_{net} ” of these units, function of:

- The hot temperature of the hot source called T_{hi}
- The cold temperature of the cold source called T_{ci} .

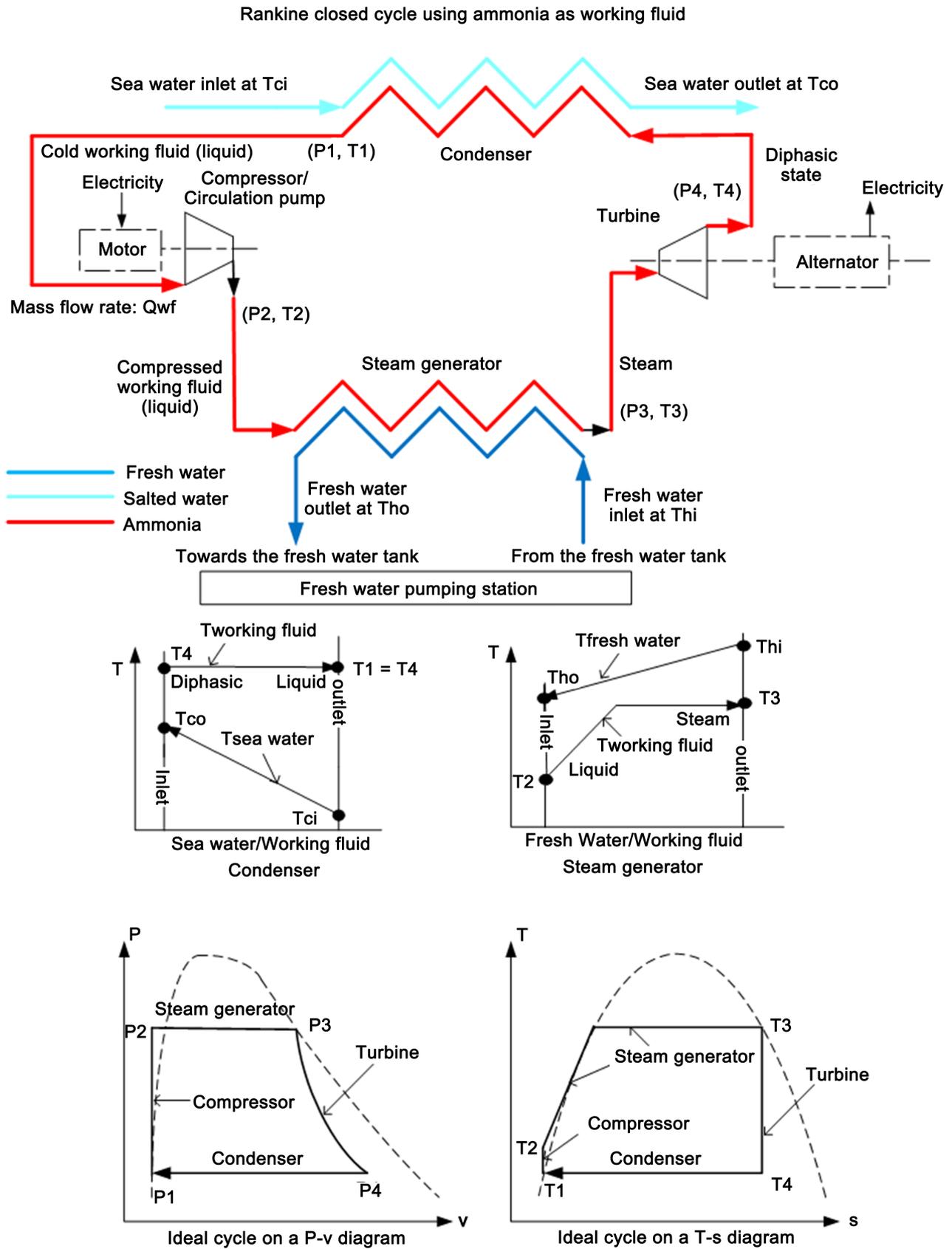
Let’s call “ P_{en} ” the net electric power provided to the grid net and “ P_h ” the heat power consumed from the hot source. So

$$\eta_{net} = \frac{P_{en}}{P_h} \quad (1)$$

3.2. Diagram of the Thermal Energy Conversion Unit for a Rankine Cycle

The diagram of a machine based on the Rankine closed cycle is proposed below, in **Figure 4**. The refrigerant is usually ammonia (NH₃, R717).

Note 1: in **Figure 4**, “ P ” is worth for “Pressure”, “ v ” for “specific volume”, “ T ” for “Temperature” and “ s ” for “specific entropy”.



Note 2: the cycle is shown as ideal. However, there is irreversibility; for example the compression and the expansion are not really isentropic, there are losses of heat along the cycle, etc.

So the real efficiency is not as well as the ideal efficiency.

Compared to a true Rankine cycle, in **Figure 4**, it misses equipment such as a separator, different tanks and pumps, different systems relative to the turbine and the compressor, steam tapping from the turbine directed towards a heater, etc. It is not necessary to take into account this equipment as it will be indirectly considered in the real efficiency estimate (see §3.3).

Note: for this chapter and for all the calculations in this document, the thermodynamic data for R717 (NH₃) comes from [7]. Note that the sole saturated state is given, so the enthalpy and density of the liquid in non-saturated state is taken from the liquid in saturated state considering the sole temperature. This introduces a very small error. The thermodynamic data for water comes from [8].

3.3. Determination of the Net Rankine Thermodynamic Efficiency

To determine this net efficiency, we are going to consider the study of an OTEC Rankine unit done by Creusot-Loire in 1980 and exposed in [6] pages 33 and 34. The previous **Figure 4** represents **Figure 8** of [6] (page 34), from which the temperatures and pressures can be extracted:

- $T_{ci} = 4.8^\circ\text{C}$, $T_{co} = 9.05^\circ\text{C}$.

$T_{co} - T_{ci} = 4.25^\circ\text{C}$, which depends on the condenser type, is supposed constant.

- $T_{hi} = 26.8^\circ\text{C}$, $T_{ho} = 23.94^\circ\text{C}$ (so $T_{hi} - T_{ho} = 2.86^\circ\text{C}$).
- $P_1 = 6.45$ bar, $T_1 = 11.32^\circ\text{C}$.
- $P_2 = 8.96$ bar, $T_2 = T_1 + \varepsilon \approx T_1$.
- $P_3 = 8.96$ bar, $T_3 = 21.34^\circ\text{C}$, with $P_3 = f(T_3)$ in saturated state.
- $P_4 = 6.50$ bar, $T_4 = 11.55^\circ\text{C}$, with $P_4 = f(T_4)$ in saturated state.
- The fresh water volume flow rate Q_{vh} is equal to $60 \text{ m}^3/\text{s}$ so, with a density $\rho_h = 997 \text{ kg}/\text{m}^3$ at 25°C , the mass flow rate $Q_{mh} = 59,820 \text{ kg}/\text{s}$.
- The sea water volume flow rate Q_{vc} is equal to $39 \text{ m}^3/\text{s}$ so, with a density ρ_c supposed equal to $\rho_c = 1000 \text{ kg}/\text{m}^3$ (the salinity not being given), the mass flow rate $Q_{mc} = 39,000 \text{ kg}/\text{s}$.

The heat transfer rate Ph (in W) from the hot source is equal to

$$Ph = Q_{mh} \times C_p \times (T_{hi} - T_{ho}) \quad (2)$$

with C_p the specific heat capacity = $4200 \text{ J}/(\text{kg} \cdot ^\circ\text{K})$ at 25°C . So $Ph = 7.186\text{E}8 \text{ W}$.

The theoretical Carnot efficiency η_{Carnot} between the high temperature at the steam generator output (T_3 in $^\circ\text{K}$) and the low temperature at the condenser temperature input (T_4 in $^\circ\text{K}$) is equal to:

$$\eta_{Carnot} = \frac{T_3 - T_4}{T_3} = 3.324\text{E}-2 \quad (3)$$

The rough electric power P_{erough} generated is equal to $15\text{E}6 \text{ W}$ according to

[6]. So the rough efficiency η_{rough} of this cycle is equal to

$$\eta_{rough} = \frac{P_{rough}}{P_h} = 2.087E-2 \quad (4)$$

(which is a standard value for this OTEC machine).

η_{rough} can be expressed as

$$\eta_{rough} = \eta_{Carnot} \times \eta_{Rankine} \times \eta_{Machine} \quad (5)$$

(see [6] page 28).

With $\eta_{Rankine}$ the Rankine efficiency (compared to the Carnot efficiency).

$\eta_{Machine}$ is mainly the efficiency of the turbo generator but more generally the machine efficiency compared to an ideal Rankine efficiency. Here

$$(\eta_{Rankine} \times \eta_{Machine}) = \frac{\eta_{rough}}{\eta_{Carnot}} = 0.628 \quad (6)$$

So $\eta_{rough} = \eta_{Carnot} \times 0.628$

Now according to [6], the net electric power P_{en} generated is equal to 10E6 W. The difference (P_{ea}) between the rough and the net power

$$P_{ea} = P_{rough} - P_{en} = 5E6 \text{ W (5 MW)} \quad (7)$$

corresponds to the electric power mainly (*i.e.*, 4 MW) consumed by the pumps making circulate the sea water and the fresh water. The rest of these 5 MW (about 1 MW) is consumed by the compressor and the different pieces of equipment of the plant (see §3.3.1 below).

It will be supposed that:

- The thermal performance of the steam generator and the condenser, as exchangers, will be constant whatever the heat transfer rate P_h . This means that the difference in temperature $\Delta T_h = T_{hi} - T_3 = 5.46^\circ\text{C}$ and $\Delta T_c = T_4 - T_{ci} = 6.75^\circ\text{C}$ are supposed constant. So:

$$T_3 = T_{hi} - (T_{hi} - T_3) = T_{hi} - \Delta T_h = T_{hi} - 5.46$$

$$T_4 = T_{ci} + (T_4 - T_{ci}) = T_{ci} + \Delta T_c = T_{ci} + 6.75$$

With T_{hi} and T_{ci} in $^\circ\text{C}$, $\eta_{Carnot} = \frac{T_{hi} - 5.46 - T_{ci} - 6.75}{T_{hi} - 5.46 + 273.15}$

$$\eta_{Carnot} = \frac{T_{hi} - T_{ci} - 12.21}{T_{hi} + 267.69} \quad (8)$$

So

$$\eta_{rough} = \eta_{Carnot} \times 0.628 = 0.628 \times \frac{T_{hi} - T_{ci} - 12.21}{T_{hi} + 267.69} \quad (9)$$

- P_{ea} is proportional to P_h , because the different pumping powers (see §3.3.1 below) mainly depend on the volume flow rates Q_{vh} and Q_{vc} , which are proportional to P_h (for a supposed constant pressure drop). Indeed P_h is proportional to:

○ “Qmh” because $P_h = Q_{mh} \times C_p \times (T_{hi} - T_{ho})$.

$$Q_{mh} = \rho h \times Q_{vh} \quad (10)$$

with ρh supposed constant, so $P_h \sim Q_{vh}$

- “ Q_{mc} ” because $Ph - P_{erough} \approx Q_{mc} \times C_p \times (T_{co} - T_{ci})$ with P_{erough} very inferior to Ph and proportional to Ph ($\frac{P_{erough}}{Ph} = \eta_{rough}$).
 $Q_{mc} = \rho_c \times Q_{vc}$ with ρ_c supposed constant, so $Ph \sim Q_{vc}$ (and incidentally $Q_{vc} \sim Q_{vh}$ at η_{rough} constant)

So from $Pea = 5$ MW at $Ph = 7.186E8$ W, for a given Ph ,

$$Pea = Ph \times \left(\frac{5E6}{7.186E8} \right) = 6.958E-3 \times Ph \tag{11}$$

So for a given Ph ,

$$\begin{aligned} Pen &= Ph \times \eta_{rough} - Pea \\ &= Ph \times \eta_{rough} - Ph \times 6.958E-3 \\ &= Ph \times (\eta_{rough} - 6.958E-3) \\ Pen &= Ph \times \left(\frac{0.628 \times (Thi - Tci - 12.21)}{Thi + 267.69} - 6.958E-3 \right) \end{aligned} \tag{12}$$

Finally the net efficiency η_{net} is equal to $\eta_{net} = \frac{Pen}{Ph}$ so:

$$\eta_{net} = \frac{0.628 \times (Thi - Tci - 12.21)}{Thi + 267.69} - 6.985E-3 \tag{13}$$

with Thi and Tci in °C.

Note: for OTEC plants where the difference of temperature ΔThc between warm and cold sea temperatures ($\Delta Thc = Thi - Tci$) is about 22°C, 12.21°C represents a loss of a bit more than half of the initial ΔThc , which is relatively high. This relative loss in the proposed plant is only about the half due to a higher ΔThc (about 42°C).

Estimation of the Different Components of Pea

It is interesting to calculate how Pea (=5 MW) is composed.

About the working fluid (refrigerant) circuit

Let’s call Q_{mwf} the working fluid mass flow rate. The heat transfer of the steam generator from the hot source to the working fluid is such that

$Ph = Q_{mwf} \times (H_{sv}(T3) - Hl(P2, T2))$, with $H_{sv}(T3)$ the saturated steam enthalpy function of $T3$ and $Hl(P2, T2)$ the liquid enthalpy function of $P2, T2$.

So

$$Q_{mwf} = \frac{Ph}{H_{sv}(T3) - Hl(P2, T2)}$$

In our example, $Q_{mwf} \approx 585$ kg/s. So with a liquid volume mass $\rho_{wf} \approx 622.8$ kg/m³, the liquid volume flow rate:

$$Q_{vwf} = \frac{Q_{mwf}}{\rho_{wf}} = 0.939 \text{ m}^3/\text{s} \tag{14}$$

for the example.

The mechanical power P_{mwf} necessary for the compressor (**Figure 4**) to pump the working fluid in liquid state is equal to

$$P_{mwf} = Q_{vwf} \times (P_2 - P_1) \approx 2.36E5 \text{ W} \quad (15)$$

for the example.

About the fresh water circuit

It is supposed that the total pressure drop ΔPh through the fresh water circuit (hot source) is equal to 0.3 bar. So, the mechanical power P_{mh} necessary for the fresh water circulation pump (**Figure 1**) to pump the fresh water through the steam generator is equal to: $P_{mh} = Q_{vh} \times \Delta Ph = 1.80E6 \text{ W}$ for the example.

About the sea water circuit

It is supposed that the pressure drop ΔPc through the sea water circuit (cold source) is equal to 0.4 bar. So, the mechanical power P_{mc} necessary for the sea water circulation pump (**Figure 1**) to pump the sea water through the condenser is equal to: $P_{mc} = Q_{vc} \times \Delta Pc = 1.56E6 \text{ W}$ for the example.

Electric pumping power (P_{ep})

The sum of these 3 mechanical pumping powers is equal to

$$P_{mp} = P_{mwf} + P_{mh} + P_{mc} = 3.60E6 \text{ W}$$

Given a global efficiency of 0.85 for these pumps, the expected electric pumping power P_{ep} is equal to: $P_{ep} = \frac{P_{mp}}{0.85} = 4.23E6 \text{ W}$.

Reminder power (P_{er})

The reminder power equal to $P_{er} = P_{ea} - P_{ep} = 0.77E6 \text{ W}$ for the example, is intended to the different auxiliaries, control, light of the plant, etc.

Note: the P_{mp} power (and so P_{ep} and P_{ea}) could be reduced by increasing the expected pipes diameter (and their cost) and so decreasing the pressure drop. The P_{ea} power limited to 5 MW for $Ph = 7.186E8 \text{ W}$ must be considered as a technical-economical objective.

3.4. Diagram of the Thermal Energy Conversion Unit for a Claude Cycle and Determination of the Claude Thermodynamic Efficiency

The diagram of a machine based on the Claude cycle is proposed below in **Figure 5**.

The working fluid circuit is connected by the fresh water circuit (in broad lines) instead of the sea (in OTEC plants). It is considered as an “open-cycle” contrary to the Rankine “closed-cycle” for which the same fluid circulates along the cycle.

The working fluid is condensed water under vacuum because under the pressure of one atmosphere, the boiling temperature is 100°C, superior to the hot temperature targeted (T_3 around 71°C).

As for the Rankine cycle, this one is shown as ideal. Now, the cycle as shown is a pseudo cycle because the fluid circulation in the equipment is not always the same, as the fresh water entering the evaporator can become working fluid in the form of steam or can return to the fresh water tank. However, the matter being the same, the cycle can be represented as it was a closed cycle, so with the same flowrate all along the cycle.

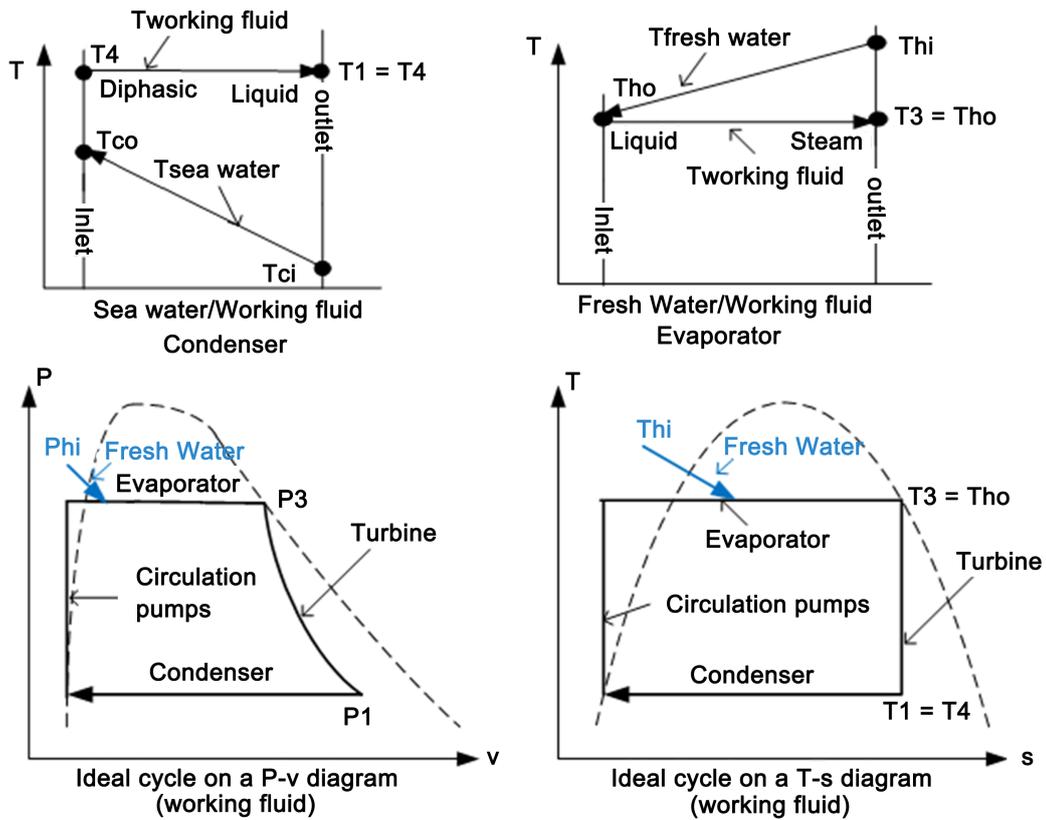
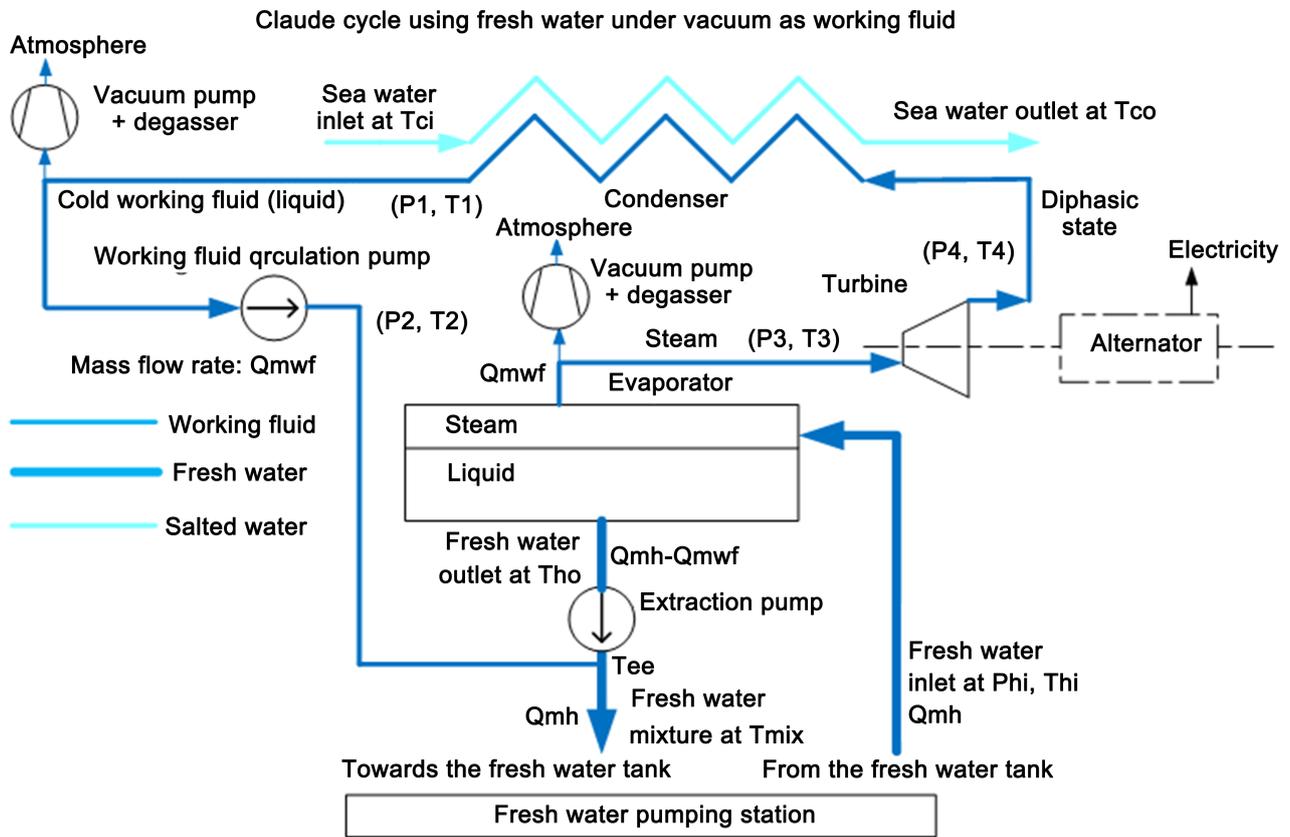


Figure 5. Claude cycle using fresh water under vacuum as working fluid.

Let's suppose that the temperatures of this Claude cycle be the same as the Rankine cycle (§3.2), except for Tho which is now equal to $T3$, this to keep the same final efficiency.

- $Tci = 4.8^\circ\text{C}$, $Tco = 9.05^\circ\text{C}$.

$Tco - Tci = 4.25^\circ\text{C}$, which depends on the condenser type, is supposed constant.

- $Thi = 26.8^\circ\text{C}$, $Tho = 21.34^\circ\text{C} (=T3)$, Phi is a bit superior to $P3$, to permit a flow rate (a precise determination being outside the scope of this article).
- $T1 = 11.32^\circ\text{C}$, $P1 \approx P4$.
- $T2 = T1 + \varepsilon \approx T1$ ($P2$ is roughly equal to $P3 + \Delta Pwf$ to compensate the pressure drop through the pipes leading to the fresh water tank, a precise determination of $P2$ being outside the scope of this article).
- $Thi = 26.8^\circ\text{C}$ (as for the Rankine cycle).
- $T3 = Thi - (Thi - T3) = Thi - 5.46 = 21.34^\circ\text{C}$ (as for the Rankine cycle), so $P3 = 0.0254$ bar, with $P3 = f(T3)$ in saturated state.
- $T4 = 11.55^\circ\text{C}$ so $P4 = 0.0136$ bar ($P4 = f(T4)$ in saturated state).

The heat transfer rate Ph (in W) from the hot source is the same as for the Rankine cycle (§3.2), equal to $Ph = 7.186 \text{ E8 W}$.

Note: the altitude of the equipment (evaporator, turbine and condenser) under vacuum ($P3$ or $P4$) is such that the barometric static pressure of the working fluid corresponds to the vacuum pressure (so around 9 m).

In broad blue lines on **Figure 5**, it is symbolically displayed the evolution of the fresh water exchanging heat to make boil part of the flowrate. So the fresh water transfers, per second, the heat power.

$$Ph' = Qmh \times (Hl(Thi, Phi) - Hls(Tho)) \approx Qmh \times Cp \times (Thi - Tho) \quad (16)$$

with Hl for the liquid enthalpy in non saturated state and Hls for the liquid enthalpy in saturation state.

The steam flowrate $Qmwf$ will be such that $Ph' = Qmwf \times Lv(Tho)$ with Lv the heat of vaporization at Tho .

So

$$Qmh \times Cp \times (Thi - Tho) \approx Qmwf \times Lv(Tho) \quad (17)$$

In our example, $Thi = 26.8^\circ\text{C}$, $Tho = 21.34^\circ\text{C}$, $Lv(21.34^\circ\text{C}) = 2451 \text{ kJ/kg}$ and $Cp = 4200 \text{ J/kg}$.

It can be deduced that

$$\frac{Qmh}{Qmwf} \approx \frac{Lv(Tho)}{Cp \times (Thi - Tho)} = 106.9 \quad (18)$$

So a big flow rate of hot water is necessary to produce a small flow rate of steam.

At the outlet (*i.e.* towards the fresh water tank), it appears a mixture at $Tmix$ (**Figure 5**), so that (considering a constant Cp):

$$(Q_{mh} - Q_{mwf}) \times C_p \times T_{ho} + (Q_{mwf} \times C_p \times T_2) = Q_{mh} \times C_p \times T_{mix}$$

So

$$T_{mix} = \frac{(Q_{mh} - Q_{mwf}) \times T_{ho} + Q_{mwf} \times T_2}{Q_{mh}} \quad (19)$$

For the example, with $\frac{Q_{mh}}{Q_{mwf}} = 106.9$, $T_{mix} = 21.25^\circ\text{C}$ which is very close to T_{ho} (21.34°C).

To be equivalent to the Rankine cycle, the total heat transfer (Ph) must be equal to $7.186\text{E}8$ W, for our example. The heat is extracted from the evaporator and from the working fluid circuit, so that:

$$Ph = Q_{mh} \times C_p \times (T_{hi} - T_{mix}) \quad \text{and} \quad Q_{mh} = \frac{Ph}{C_p \times (T_{hi} - T_{mix})} \quad (20)$$

For our example, $Q_{mh} = 30,807$ kg/s, so equal to about the half of the Q_{mh} necessary for the Rankine cycle ($Q_{mh} = 59,820$ kg/s).

So the volume flow rate $Q_{vh} = \frac{Q_{mh}}{\rho_h} = 30.9$ m³/s with $\rho_h = 998.3$ kg/m³.

Then $Q_{mwf} = \frac{Q_{mh}}{106.9} = 288.2$ kg/s

The volume flow rate pumped by the extraction pump (Figure 5) will be equal to:

$$Q_{vep} = \frac{Q_{mh} - Q_{mwf}}{\rho_h} = 30.6$$
 m³/h (21)

The Claude cycle is thermodynamically very close to the Rankine cycle, the slight difference being between the different behavior of the steam generator and the evaporator. The temperatures across the turbine (T_3 and T_4) being the same for both cycles, it is considered that Pen , Pea , η_{Carnot} , η_{rough} and η_{net} are the same as the ones calculated for the Rankine cycle (cf. §3.3).

Consequently, the net efficiency η_{net} calculated for the Rankine cycle (cf. §3.3) is applicable to the Claude cycle:

$$\eta_{net} = \frac{0.628 \times (T_{hi} - T_{ci} - 12.21)}{T_{hi} + 267.69} - 6.985\text{E}-3 \quad (13)$$

with T_{hi} and T_{ci} in $^\circ\text{C}$.

Estimation of the Different Components of Pea

As for the Rankine cycle (§3.3.1), it is interesting to calculate how Pea (=5 MW) is composed.

About the working fluid circuit towards the fresh water tank

In our example, $Q_{mwf} = 288.2$ kg/s. So with a liquid volume mass $\rho_{wf} = 1000$ kg/m³, the liquid volume flow rate: $Q_{vw} = \frac{Q_{mwf}}{\rho_{wf}} = 0.288$ m³/s for the example.

It is supposed that the pressure drop ΔP_{wf} through the pipes leading to the

fresh water tank through the Tee (see **Figure 5**) is equal to 0.15 bar. Because $P_2 \approx P_3 + \Delta P_{wf}$, the mechanical power P_{mwf} necessary for the working fluid circulation pump (**Figure 5**) to pump the working fluid in a liquid state is equal to $P_{mwf} = Q_{v_{wf}} \times (P_2 - P_1) = Q_v \times (\Delta P_{wf} + (P_3 - P_1)) = 4663 \text{ W}$ for the example, which is very weak, compared to the compressor in the Rankine cycle (§3.3.1).

About the fresh water circuit towards the fresh water tank

It is supposed that the pressure drop ΔP_{ho} through the pipes leading to the fresh water tank through the Tee (see **Figure 5**) is equal to 0.15 bar. The mechanical power P_{mho} necessary for the extraction pump (**Figure 5**) to pump the fresh water is equal to: $P_{mho} = Q_{v_{ep}} \times \Delta P_{ho} = 4.59\text{E}5 \text{ W}$ for the example.

About the fresh water circuit from the fresh water tank

It is supposed that the pressure drop ΔP_{hi} through the pipes coming from the fresh water tank towards the evaporator is equal to 0.15 bar. The mechanical power P_{mhi} necessary for the fresh water circulation pump (**Figure 1**) to pump the fresh water is equal to: $P_{mhi} = Q_{vh} \times \Delta P_{hi} = 4.63\text{E}5 \text{ W}$ for the example.

About the sea water circuit

The mechanical power P_{mc} necessary for the sea water circulation pump (**Figure 1**) to pump the sea water through the condenser is the same as for the Rankine cycle (§3.3.1) (*i.e.*, $P_{mc} = 1.56\text{E}6 \text{ W}$ for the example).

Electric pumping power (P_{ep})

The sum of these 4 mechanical pumping powers is equal to

$$P_{mp} = P_{mwf} + P_{mho} + P_{mhi} + P_{mc} = 2.49\text{E}6 \text{ W}$$

Given a global efficiency of 0.85 for these pumps, the expected electric pumping power P_{ep} is equal to:

$$P_{ep} = \frac{P_{mp}}{0.85} = 2.92\text{E}6 \text{ W}.$$

Power for vacuum (P_{ev})

The necessary vacuum pumps and degassing units to trap dissolved gasses are supposed to consume $P_{ev} = 0.14\%$ of P_h , so 1 MW in the example.

Reminder power (P_{er})

The reminder equal to $P_{er} = P_{ea} - P_{ep} - P_{ev} = 1.08\text{E}6 \text{ W}$ for the example, is intended to the different auxiliaries, control, light of the plant, etc.

3.5. Comparison of the Rankine and the Claude Cycles

3.5.1. Advantages of the Rankine Cycle Compared to the Claude Cycle

- The Rankine cycle is better mastered than the Claude cycle, due to the various OTEC studies and small plants built. Note that none OTEC big plants (*i.e.*, net power $\geq 100 \text{ MW}$) has been built to date.
- In the OTEC domain, the turbine is much smaller and much less costly than for a Claude cycle. The difference of pressure across the turbine ($P_3 - P_4$) is about 2.4 bar for a Rankine cycle versus 0.015 bar for a Claude cycle, which explains the different of size.

Now for the proposed plant, the expected mean temperatures T_3 and T_4 will

respectively be equal to 71°C and 29°C. So the different expected pressures will be equal:

- For a Rankine cycle (R717 ammonia): $P_3 = 33.9$ bar, $P_4 = 11.3$ bar, $P_3 - P_4 = 22.6$ bar
- For a Claude cycle (water under vacuum): $P_3 = 0.325$ bar, $P_4 = 0.0400$ bar, $P_3 - P_4 = 0.285$ bar

Compared to OTEC turbines, it is obvious that, in both cases, the turbines will be much smaller (and cheaper) due to the increase of the difference of pressure across the turbine ($P_3 - P_4$), by a factor of 9 for the Rankine cycle and 19 for the Claude cycle.

- In the Rankine cycle there is no need to create vacuum as for the Claude cycle, for which it is necessary to add vacuum pumps and degassing units to trap dissolved gasses and mainly air. However, in the OTEC domain, the minimum vacuum pressure is equal to about 1300 Pa (P_1 or P_4). For the proposed plant, the mean saturation pressure P_4 is equal to about 4000 Pa, which is less penalizing.

3.5.2. Advantages of the Claude Cycle Compared to the Rankine Cycle

- With water as a working fluid, there is no toxic risk as with ammonia. Even if the ammonia is well mastered by industry, the population could be reluctant toward this gas.
- The cost of an evaporator is very inferior to the cost of a steam generator, due to a much smaller contact surface needed (see [6] p 73).
- In the OTEC domain, the total cost of a Claude thermal energy conversion unit is a bit cheaper than the Rankine one according to [6] p 73.
- In fact, the evaporator needs a difference of temperature $T_{hi} - T_3$ of about 3.5°C versus 5.5°C for a steam generator, so the rough efficiency of a Claude cycle is better for an OTEC unit: 2.7% versus 2.11%, according to [6] page 70. However, for the plant proposed, this gain would be very small due to the yet relatively elevated difference in temperature $T_3 - T_4$ (about 42°C).
- The pressure being much smaller for the Claude cycle compared to the Rankine cycle (<1 bar versus 34 bar as given in §5.3), the mechanical sizing of pipes and equipment will probably be simpler and the cost weaker, due to a smaller thickness of the metal layer.

3.5.3. Common Advantage and Disadvantage

- For the sole thermal energy conversion unit, it is obvious that for the same electric power delivered, the cost will be much smaller than the cost of such a unit in the OTEC domain, simply because the net efficiency η_{net} is roughly 5 times better. It means that for the same heat power from the hot source, it is delivered to the grid net, an electric power 5 times higher than an OTEC unit.
- Now there will be the same problem of microbial fouling of the condenser

as for all the plants (OTEC, nuclear...) located on the seashore. See for example [5] for more details. It is a classical problem solved with filters of different kinds at the sea water inlet, the maintenance of the condenser being done, among others, by periodic mechanical and chemical (chlorination) treatments. This problem is outside the scope of this article.

4. Selection of the Best Solar Thermal Collector Type and Determination of the Mean Thermal Efficiency

4.1. Selection of the Best Solar Thermal Collector Type

There are 3 types of non-concentrating solar thermal collector heating water (see [9] and [10] for general information):

- Unglazed liquid collectors are mainly used to heat water for swimming pools. They are simple and the cheapest among the collectors. The problem is their poor efficiency, which prevents any use for an application of solar thermal plant.
- Evacuated tube collectors have the best efficiency. However, they use sophisticated technology (high vacuum) and they are the most expensive among the collectors. Their sophistication is an interrogation of competitive lifetime. Moreover, their relatively high cost will also prevent us from using them as a hot source for a solar thermal plant.
- Glazed flat plate collectors (also called “glass-top flat surface solar collectors”). It is the best choice as they are simple, relatively cheap and efficient (with cost and efficiency between unglazed and evacuated collectors).

4.2. Description of the Hot Source

In **Figure 6**, it is shown the hot source formed by these collectors.

Fresh water is extracted at the temperature T_{si} (about 71°C), by the extraction pump from the inferior layer of the fresh water tank, a tank which is intended to supply the thermal energy conversion unit with hot water during the night period so as to provide a continuous working day and night, all year round.

Note 1: this tank is supposed thermally insulated to limit heat loss to the minimum, and covered by a roof to avoid evaporation. For the calculations, this tank is supposed thermally stratified, *i.e.*, hot water (about 77°C) remains in the superior layer and the “warm” water (about 71°C) remains in the inferior layer. However, it is not a requirement, as the water could be totally mixed.

To remain simple, the tank will have a small depth (let's say ≤ 10 m), and will look like a covered in-ground pool.

Note 2: the fresh water extraction pump is stopped during the night period. It is a variable flow rate pump, because according to the horizontal irradiation power (I_r), to have a constant ΔT_s across the solar thermal collectors field (5.46°C in our example, according to §4.3), the flowrate must be controlled (see also Section 7).

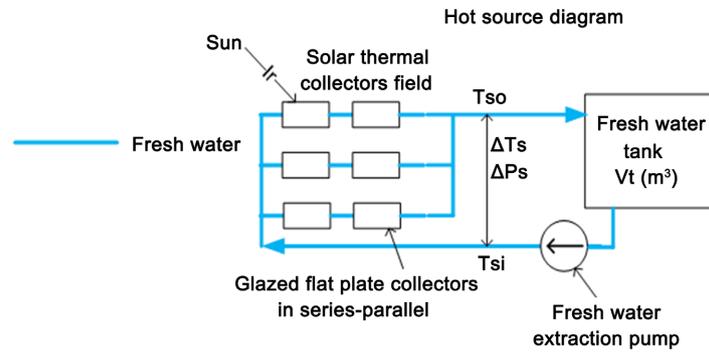


Figure 6. Hot source diagram.

After heating by the collectors (up to T_{so} , about 77°C) during the daylight period, the fresh water returns to the superior layer of the tank.

4.3. Mean Thermal Efficiency of the Glazed Flat Plate Collectors

From now on, only glazed flat plate collectors will be considered. They will be simply called “collectors”.

The thermal power Ph_{co} generated by a heat-absorbing surface collector Sc_o (m^2) is equal to:

$$Ph_{co} = \eta_{co} \times Sc_o \times Ir \tag{22}$$

With η_{co} the thermal collector efficiency and Ir the horizontal irradiation power (W/m^2).

According to [10] pages 121 and 122,

$$\eta_{co} = \eta_{c0} - \frac{a1 \times \Delta T_{co}}{Ir} - \frac{a2 \times \Delta T_{co}^2}{Ir} \tag{23}$$

- η_{c0} is the thermal efficiency without heat loss (ideally it might be equal to 1).
- $a1$ is the linear coefficient of heat transfer (ideally it might be equal to 0).
- $a2$ is the quadratic coefficient of heat transfer (ideally it might be equal to 0).

$$\Delta T_{co} = T_m - T_a \quad (^\circ\text{C}) \tag{24}$$

T_m is the mean temperature inside the collectors. For all collectors, it is supposed that T_m is the mean temperature between the inlet (T_{si}) and the outlet (T_{so}) temperatures of the solar thermal collectors field:

$$T_m = \frac{T_{si} + T_{so}}{2} \tag{25}$$

Moreover, due to the supposed stratification of the tank, it can be written:

- $T_{so} = Th_i$, i.e., the temperature at the outlet of the collectors is equal to the temperature at the inlet of the steam generator or the evaporator (see Figure 1).
- $T_{si} = Th_o$, i.e., the temperature at the inlet of the collectors is equal to the temperature at the outlet of the steam generator or the evaporator.

See **Figure 4** for the Rankine cycle. For the Claude cycle (**Figure 5**) T_{mix} would have to be considered. However, T_{mix} is very close to T_{ho} (see Note of §3.4), so taking T_{ho} (instead T_{mix}) only generates a very slight error.

To simplify, it will be done a sole calculation for both the Rankine and Claude cycles. The lowest T_{ho} will be considered. Now $T_{hi} - T_{ho} = 2.86^\circ$ for the Rankine cycle (§3.3) and $T_{hi} - T_{ho} = 5.46^\circ\text{C}$ for the Claude cycle (§3.4). T_{hi} being supposed constant, the lowest T_{ho} is the Claude one, with $T_{ho} = T_{hi} - 5.46^\circ\text{C}$, so $T_{si} = T_{hi} - 5.46^\circ\text{C}$.

$\Delta T_s = T_{so} - T_{si} = 5.46^\circ\text{C}$, which introduces a small error for the Rankine cycle.

So

$$T_m = \frac{T_{hi} + T_{ho}}{2} = T_{hi} - 2.73$$

T_a is the ambient air temperature. So

$$\Delta T_{co} = T_{hi} - 2.73 - T_a \quad (26)$$

Among these collectors there are a wide variety of equipment, from very efficient and expensive (about 200 Euros/m² in 2022) down to not so efficient but cheap (about 30 Euros/m² in 2022).

For the example, it will be chosen a collector with an intermediate efficiency: $\eta_{c0} = 0.78$, $a_1 = 3.3$ and $a_2 = 0.014$. So:

$$\eta_{co} = 0.78 - \frac{3.3 \times \Delta T_{co}}{I_r} - \frac{0.014 \times \Delta T_{co}^2}{I_r} \quad (27)$$

Roughly, the best equipment would be 15% better in term of efficiency (η_{co}) and the worst equipment would be 15% worse.

5. Mean Net Electric Power (Pen_1) Delivered on a Continuous Working over One Year, for 1 m² of Collector

5.1. Goal and Hypothesis

The goal of this estimate is to assess the mean net electric power (Pen) delivered by this plant over one year, if located in the Nouakchott area.

It will be supposed 1 m² of heat-absorbing surface collector, so the variable “ Pen ” will be replaced by “ Pen_1 ”.

Note: the heat-absorbing surface collector is equal to about 91% of the total collector surface.

Of course, as the net electric power (Pen) is proportional to the heat power transferred (Ph , see §3.3 and §3.4) and, finally, to the effective surface of collectors (S_{co}), it will be, afterwards, enough to determine S_{co} from a targeted net electric power (Pen).

The first possibility of assessment would be to share the year in minutes, then to calculate the net electric power delivered during each minute and finally to calculate the mean value over the year. This would be possible but complex and

over the scope of this article.

To limit the complexity of such estimate, it will be calculated a set of mean values, and on this basis, the estimate will be done. This method is simple but pessimistic.

5.2. Mean Values for the Continuous Working Estimate

It will be first determined by a meteorological set of data. From the Nouakchott data in §2.1, it can be deduced that:

- The average “max air temperature” is equal to 31.75°C.
- The average “min air temperature” is equal to 23.08°C.
- The average “ocean temperature” (T_{ci}) is equal to 22.0°C. So according to §3.3, $T_{co} = T_{ci} + 4.25 = 26.25^\circ\text{C}$ and $T_4 = T_{ci} + 6.75^\circ\text{C} = 28.75^\circ\text{C}$.

In saturated state for NH₃ (Rankine cycle), $P_4 = f(T_4) = 11.24$ bar and for water (Claude cycle), $P_4 = f(T_4) = 0.0395$ bar. Note that $P_1 \approx P_4$.

As the air temperature is minimum at the beginning of the day and maximum a bit before the end of the day, it will be considered that the average air temperature for the daylight period (T_a) is intermediate between both temperatures so

$$T_a = \frac{31.75 + 23.08}{2} = 27.4^\circ\text{C}.$$

Even if the latitude of Nouakchott is not equatorial (18.1° North) it will be supposed that the mean daylight duration (Dp) is equal to 12 h (in fact very slightly superior to 12 h), and so corresponds to the March and September equinoxes (*i.e.*, 20th of March and 23th of September in 2022).

According to [2], the *GHI* (global horizontal irradiance) is between 2150 and 2200 kWh/m² per year in the Nouakchott area. It will be considered 2175 kWh/m² per year.

So the mean horizontal irradiation power during daylight.

$$I_r = \frac{1000 \times GHI}{365 \times 12} = 496.6 \text{ W/m}^2$$

The mean values and the values deduced are listed below:

- $T_{ci} = 22.0^\circ\text{C}$, $T_{co} = 26.25^\circ\text{C}$, $T_4 = 28.75^\circ\text{C}$, $T_2 \approx T_1 \approx T_4$.

From T_4 , $P_4 = 11.24$ bar ($\approx P_1$) for a Rankine cycle and $P_4 = 0.0395$ bar ($\approx P_1$) for a Claude cycle.

- $T_a = 27.4^\circ\text{C}$
- Mean daylight duration (Dp) = 12 h
- $I_r = 496.6 \text{ W/m}^2$ during the 12 h of daylight period.

5.3. Determination of the Ideal Mean Hot Temperature Thi

Using the mean values, the mean hot temperature Thi at the steam generator or evaporator inlet (see **Figure 4** and **Figure 5**) must be determined, using as criteria the maximum net electric power (Pen).

Below are reminded the Equation (13) (§3.3 and §3.4), (26) and (27) (§4.3):

$$\eta_{net} = \frac{0.628 \times (Thi - T_{ci} - 12.21)}{Thi + 267.69} - 6.958E-3 \quad (13)$$

$$\Delta T_{co} = T_{hi} - 2.73 - T_a \quad (26)$$

$$\eta_{co} = 0.78 - \frac{3.3 \times \Delta T_{co}}{I_r} - \frac{0.014 \times \Delta T_{co}^2}{I_r} \quad (27)$$

Note: for 1 m² of heat-absorbing surface collector, it will now be referred as:

- $Ph1$ instead Ph , with $Ph = Ph1 \times Sco$
- $Pen1$ instead Pen , with $Pen = Pen1 \times Sco$
- $Phco1$ instead $Phco$, with $Phco = Phco1 \times Sco$

Now the thermal power $Phco1$ (W/m²) generated by 1 m² of heat-absorbing surface collector is equal to:

$$Phco1 = \frac{Phco}{Sco} = \eta_{co} \times I_r \quad (28) \text{ (cf. §4.3)}$$

Because this plant, by means of water storage (Section 6), is supposed to generate electricity day and night (24 h) and not only during the daylight period Dp (12 hours according to our hypothesis in §5.2), the heat transfer rate $Ph1$ (in W/m²) (see §3.3 for Ph) to supply to the thermal energy conversion unit will be equal to:

$$Ph1 = Phco1 \times \frac{Dp}{24} \text{ so } Ph1 = \frac{\eta_{co} \times I_r \times Dp}{24} \quad (29)$$

$$\text{Because } \eta_{net} = \frac{Pen}{Ph} \text{ (§3.4), } \eta_{net} = \frac{Pen1}{Ph1}$$

So

$$Pen1 = Ph1 \times \eta_{net} \text{ and } Pen1 = \frac{\eta_{co} \times I_r \times \eta_{net} \times Dp}{24} \quad (30)$$

Dp , T_{ci} , T_a , I_r are known, T_{hi} is unknown. In a loop written in a computer program, it is easy to make slowly increase the T_{hi} value. For each T_{hi} , it will be successively found η_{net} , ΔT_{co} , η_{co} , $Pen1$.

The calculation will be stopped when the maximum value of $Pen1$ will be found.

It is found an ideal $T_{hi} = 76.7^\circ\text{C}$. For this temperature, $\eta_{co} = 40.85\%$, $\eta_{net} = 7.05\%$, $Phco1 = 203.3 \text{ W/m}^2$ and $Pen1 = 7.169 \text{ W/m}^2$. As $T_{hi} - T_3 = 5.46^\circ\text{C}$ (§3.3 and §3.4) then $T_3 = T_{hi} - 5.46 = 71.24^\circ\text{C}$.

Note: this value of $Pen1 = 7.169 \text{ W/m}^2$ can be compared to the mean power delivered over one year by the Seikh Zayed photovoltaic plant located at Nouakchott. From [11] and [12], it can be determined that this plant is composed of 29,826 photovoltaic modules of 5.72 m² each (so a total of 1.706E5 m²) generates 25,409 MWh per year. Let's suppose that it is a net electric energy and not a rough one.

It means that, over a year, the mean net generated power by m² ($Pen1$) is equal

$$\text{to } Pen1 = \frac{25409E6}{365 \times 24 \times 1.706E5} = 17.00 \text{ W/m}^2 .$$

So 1 m² of photovoltaic panel generates as much electricity as $\frac{17.00}{7.169} = 2.37 \text{ m}^2$ of thermal collector (only 2 m² with a more precise estimation in §6.4).

The values deducted are listed below:

- $T_{hi} = 76.7^\circ\text{C}$, $T_{ho} = T_{hi} - (T_{hi} - T_{ho}) = 76.7 - 2.86$ (cf. §3.3) = 73.84°C for the Rankine cycle
- $T_3 = 71.24^\circ\text{C}$ (= T_{ho} for the Claude cycle).

From T_3 , $P_3 = P_2 = 34.07$ bar for a Rankine cycle and $P_3 = 0.3287$ bar (with $P_2 \approx P_3 + \Delta P_{wf}$, cf. §3.4) for a Claude cycle.

6. Continuous Working by Means of the Fresh Water Storage

6.1. Principle

The principle of continuous working is to store heat during the daylight in the form of a temperature rise of the water inside the fresh water tank, and to make “consume” half of this heat by the thermal energy conversion unit during the night; the other half being “consumed” during the day, as shown in **Figure 7**.

From the §5.3, it has been determined that the ideal T_{hi} is equal to 76.7°C for $Pen1 = 7.169 \text{ W/m}^2$.

Now it can be observed that at $T_{hi} = 71.2^\circ\text{C}$, $Pen1 = 7.027 \text{ W/m}^2$ and at $T_{hi} = 82.2^\circ\text{C}$ $Pen1 = 7.029 \text{ W/m}^2$.

The loss in $Pen1$ compared to the nominal value (7.169 W/m^2) is inferior to 2%, so $Pen1$ is almost constant if T_{hi} remains between 71.2°C (minimum) and 82.2°C (maximum).

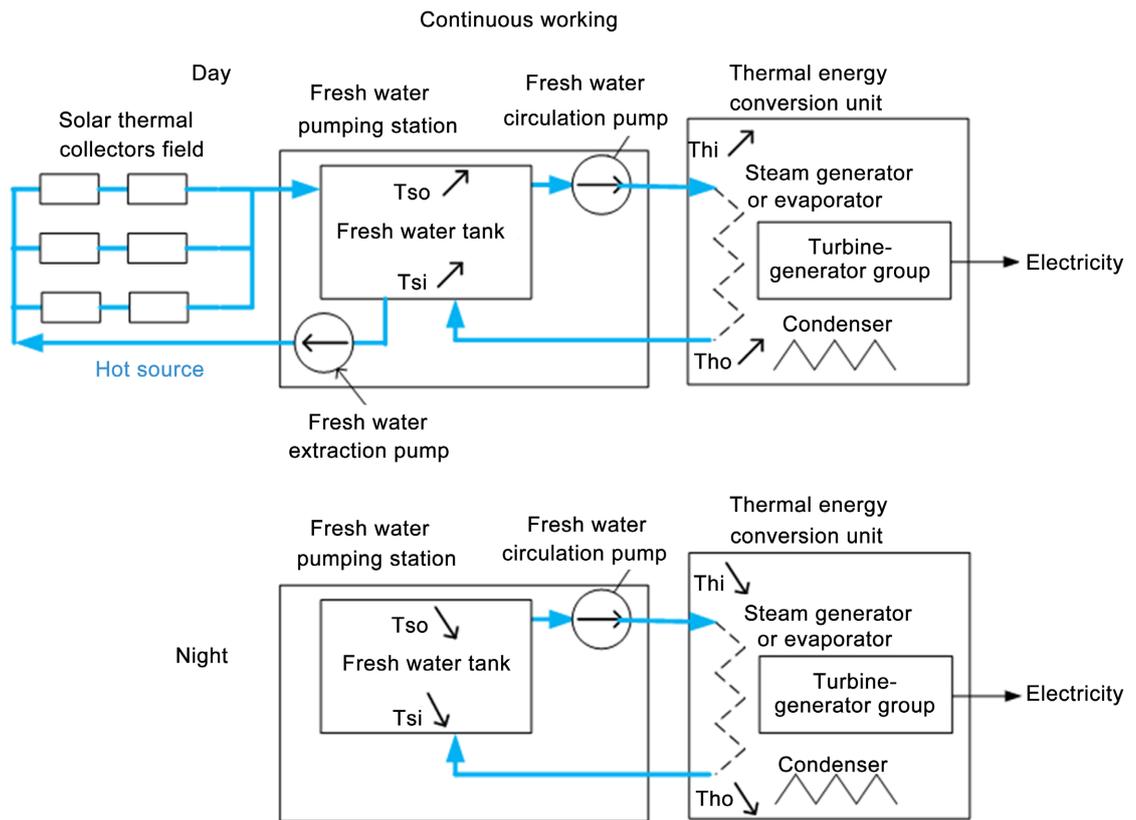


Figure 7. Continuous working.

So the principle of continuous working will be (see **Figure 7**):

- During the daylight (6 AM to 6 PM), the fresh water tank is heated so as to switch from $T_{so} = T_{hi} = 71.2^\circ\text{C}$ at the beginning of the daylight (6 AM) to $T_{so} = T_{hi} = 82.2^\circ\text{C}$ at the end of the daylight (6 PM). Meanwhile $T_{si} = T_{ho}$ will switch from $71.2 - 5.46 = 65.7^\circ\text{C}$ to $82.2 - 5.46 = 76.7^\circ\text{C}$. So the tank temperature will increase by $\Delta Tt = T_{hi}$ at 6 PM $- T_{hi}$ at 6 AM $= 82.2 - 71.2 = 11^\circ\text{C}$
- During the night (6 PM to 6 AM), the fresh water tank is slowly refreshed by extraction of heat from the thermal energy conversion unit so as to switch from $T_{so} = T_{hi} = 82.2^\circ\text{C}$ at the beginning of the night (6 PM) to $T_{so} = T_{hi} = 71.2^\circ\text{C}$ at the end of the night (6 AM). Meanwhile $T_{si} = T_{ho}$ will switch from $82.2 - 5.46 = 76.7^\circ\text{C}$ to $71.2 - 5.46 = 65.7^\circ\text{C}$. So the tank temperature will decrease by $\Delta Tt = T_{hi}$ at 6 AM $- T_{hi}$ at 6 PM $= 71.2 - 82.2 = -11^\circ\text{C}$

6.2. Determination of the Fresh Water Tank Mass ($Mt1$) and Volume ($Vt1$) for 1 m² of Collector

In the §5.3 the thermal power generated by 1 m² of the collector ($Phco1$) has been determined: $Phco1 = 203.3 \text{ W/m}^2$, at the ideal temperature $T_{hi} = 76.7^\circ\text{C}$.

During the 12 hours of daylight the heat produced $Eh1$ (J/m²) can be approximated by:

$$Eh1 \approx Phco1 \times (12 \times 3600) \quad (31)$$

Half of this heat will be directly consumed by the thermal energy conversion unit and the other half will heat the fresh water inside the tank. To increase the tank temperature by $\Delta Tt = 11^\circ\text{C}$, the mass of water $Mt1$ by m² of the collector will be such that

$$\frac{Eh1}{2} = Mt1 \times Cp \times \Delta Tt \quad \text{and} \quad Mt1 = \frac{Eh1}{2 \times Cp \times \Delta Tt} \quad (32)$$

with $Cp = 4196.8 \text{ J/kg}$, it is found $Mt1 = 95.1 \text{ kg/m}^2$ so a tank volume $Vt1$ (by m² of collector) equal to $0.0975 \text{ m}^3/\text{m}^2$ with $\rho = 975.3 \text{ kg/m}^3$

6.3. A More Precise Calculation of $Phco1$ and ΔTt

Horizontal irradiation power (I_r) according to the time angle (w)

From [13] pages 127 to 130, it can be deduced that, at equinoxes (*i.e.* declination of the Sun $\delta = 0$), the horizontal irradiance at the top of the atmosphere during the daylight is

$$I_{er}(w, \rho) \approx TSI \times \text{Cos}(w) \times \text{Cos}(\rho) \quad (33)$$

TSI (Total Solar Irradiation) = 1360.8 W/m^2 . The latitude ρ is equal to 18.1° at Nouakchott.

w is the time angle between $-\text{Pi}/2$ rd at 6h, 0 rd at 12 h and $\text{Pi}/2$ rd at 18 h.

So

$$I_{er}(w) = 1293.5 \times \text{Cos}(w) \quad (34)$$

From this equation, supposing a constant clearness index (“*Cl_i*”, from 0 to 1) for the luminosity during the daylight, it can be written:

$$I_r(w) = I_{er}(w) \times Cl_i = 1293.5 \times Cl_i \times \cos(w) = I_{r0} \times \cos(w) \quad (35)$$

between 6 h and 18 h.

After some calculations, the mean value *I_r* can be written

$$I_r = \frac{2 \times I_{r0}}{\pi} \quad (36).$$

So $I_{r0} = \frac{\pi}{2} \times I_r$. *I_r* is known and equal to 496.6 W/m² (§5.2) so *I_{r0}* = 780.1 W/m² and

$$I_r(w) = 780.1 \times \cos(w) \quad (37).$$

Note: a more general but more complex equation $I_r(w) = f(w)$, could be calculated for any latitude ρ and any declination of the Sun δ (between -23.43° and 23.43°), taking into account the relative Earth orbit eccentricity factor (between -3.344% and 3.344%). It’s not necessary here.

Daily ambient air temperature (*T_a*) according to the time (*t*)

The daily temperature *T_a* can be approximated by a sinus function evolving between:

- The average “max air temperature”: 31.75°C (§5.2), supposed to be obtained at 18 h.
- The average “min air temperature”: 23.08°C (§5.2), supposed to be obtained at 6 h.

The temperature *T_a* can be written as, with *t* in minute:

$$T_a(t) = -4.335 \times \sin\left(\frac{\pi \times t}{720}\right) + 27.415 \quad (38)$$

The time *t* varies from 0 to 1440 minutes at midnight.

On **Figure 8**, both functions are shown.

Ta and I_r functions of the time of day

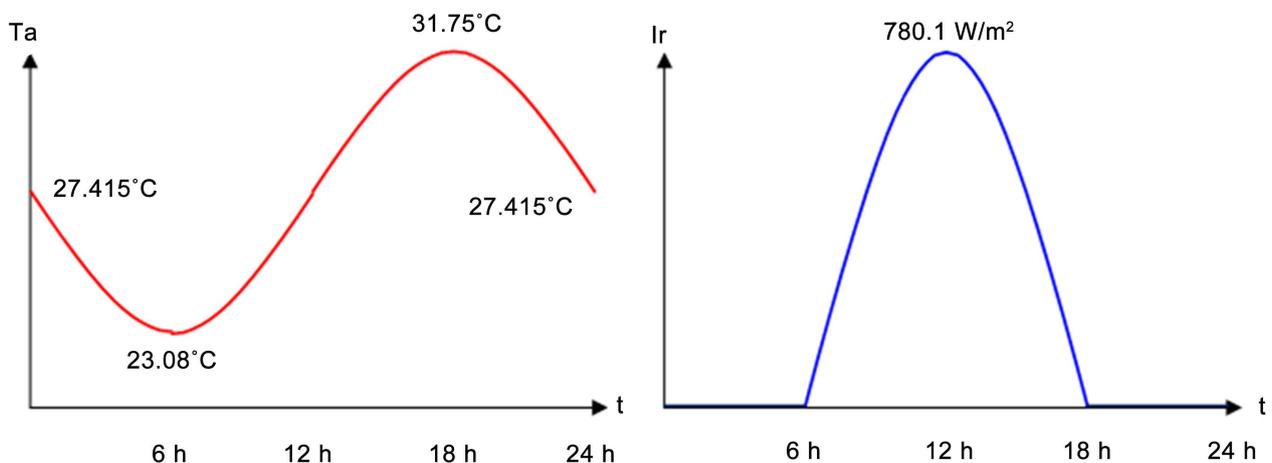


Figure 8. Ta and *I_r* functions of the time of day.

Precise calculation of $Phco1$ and ΔTt

With these two approximations, it is possible to calculate a more precise value of the thermal power $Phco1$ (§5.3).

From the minimum temperature $Thi = 71.2^\circ\text{C}$ at 6 AM (§6.1), a calculation can be done each minute from 6 h to 18 h to determine the heat energy supplied by the collector, knowing $Ir(w)$ and $Ta(t)$. Afterwards, after addition of these elementary energies, it is determined the mean thermal power $Phco1$ and the increase of temperature (ΔTt).

Finally, it is found that $Phco1 = 219.4 \text{ W/m}^2$ instead 203.3 W/m^2 , as determined in §5.3. With this new $Phco1$ value, Thi increases up to 83.1°C instead 82.2°C .

Note: the ambient temperature Ta approximation has a very weak influence on the result ($Phco1 = 219.0$ instead 219.4 W/m^2 without this approximation). A constant Ta would be sufficient.

It can be concluded that the method using means values (Sections 5 and 6) applicable for all the day, is pessimistic on $Phco$ and consequently on $Pen1$ by about 8%.

6.4. Estimates of the Minimum and the Maximum Net Electric Power per m^2 ($Pen1$) during the Year

From §5.3, it has been determined that at the ideal $Thi = 76.7^\circ\text{C}$, $Pen1 = 7.169 \text{ W/m}^2$. This corresponds to the mean estimate over the year. Let's estimate the ideal Thi at the moments of the year where $Pen1$ is either minimum or maximum.

To simplify the calculation by ignoring the Ta and Tci influence, it will be considered that the worst and the best $Pen1$ are obtained respectively for the worst Ir and the best Ir . Ir depends on the horizontal irradiance at the top of the atmosphere (extraterrestrial solar irradiation) Ier and the luminosity (clearness index Clf). From [14] page 6, it can be observed that, at Nouakchott:

- Ir is minimal in December with solar energy of $5023.07 \text{ Wh/m}^2/\text{day}$,
- Ir is maximal in April with solar energy of $7117.18 \text{ Wh/m}^2/\text{day}$.

Now, from Nouakchott data in §2.1, it can be observed that:

- In December, the “max air temperature” is equal to 29°C , the “min air temperature” is equal to 20°C and the ocean temperature (Tci) is equal to 20.1°C .
- In April, the “max air temperature” is equal to 32°C , the “min air temperature” is equal to 21°C and the ocean temperature (Tci) is equal to 18.6°C .

In the same way as explained in §5.2, it will be considered that the average air temperature (Ta) for the daylight period is intermediate between the “min air temperature” and the “max air temperature”. So:

- $Ta = 24.5^\circ\text{C}$ in December,
- $Ta = 26.5^\circ\text{C}$ in April.

From equations issued from [13] pages 97 to 99, it can be determined that the daylight period (Dp) lasts 10.925 hours on the 15th of December and 12.417 hours on the 15th of April.

So the mean horizontal irradiation power

$$Ir = \frac{Se}{Dp} \quad (39)$$

is equal to $5023.07/10.925 = 459.8 \text{ W/m}^2$ in December and $7117.18/12.417 = 573.2 \text{ W/m}^2$ in April. These pieces of data show that the mean clearness index ($Cl_i \approx 0.66$) is superior to the one implicitly given by [2] ($Cl_i \approx 0.625$), which will improve the results.

Starting from these Ir values and using the same method as the one described in §5.3, it can be determined that:

- On the 15th of December, it is found an ideal $Th_i = 71.8^\circ\text{C}$. For this temperature, $\eta_{co} = 39.86\%$, $\eta_{net} = 6.61\%$, $Ph_{co1} = 183.75 \text{ W/m}^2$ and $Pen1 = 5.53 \text{ W/m}^2$
- On the 15th of April, it is found an ideal $Th_i = 79.5^\circ\text{C}$. For this temperature, $\eta_{co} = 42.80\%$, $\eta_{net} = 8.11\%$, $Ph_{co1} = 245.83 \text{ W/m}^2$ and $Pen1 = 10.32 \text{ W/m}^2$

The mean $Pen1$ for all year round is probably around $(5.53 + 10.32)/2 = 7.92 \text{ W/m}^2$.

The mean $Pen1$ determined in §5.3 was equal to 7.169 W/m^2 . So, due to an implicit mean clearness index inferior to the measured one by about 5.3%, the mean $Pen1$ result is found pessimistic by about 9.5%.

With a probable mean $Pen1 = 7.92 \text{ W/m}^2$, the mean variation of $Pen1$ around this mean value is equal to about: $\pm 37\%$ maximum, throughout the year.

Note 1: this variation would be much less important if the collector was inclined according to the latitude, rather than installed horizontally (see first note in Section 7).

Note 2: taking into account the previous pessimistic margin (8%) determined in §6.3, the real mean $Pen1$ must be probably equal to $7.92 \times 1.08 = 8.55 \text{ W/m}^2$. So 1 m^2 of photovoltaic panel generates as much electricity as about $17.00/8.55 = 2 \text{ m}^2$ of thermal collector (see §5.3 for the 17.00 W/m^2 value).

7. Solar Collectors Field Configuration

As shown in Figure 6, the collectors can be configured in series-parallel. They are supposed to be installed on a horizontal support, to remain with a simple (but pessimistic) calculation.

Note: of course, the installation of collectors could advantageously be done on inclined support, the best angle depending on the latitude, so as to improve the heat power output and to reduce the variation of net power output between the cold and the hot periods of the year. However, a calculation taking into account the azimuth and the inclination angle is beyond the objective of this article. Now

it is reminded that for the North hemisphere, the best configuration (relatively to I_r) for an inclined collector is full southern for the azimuth and an inclination angle equal to the latitude. It is the reverse for the South hemisphere.

It comes to multiply I_r by $\frac{1}{\cos(\rho)}$ with ρ the latitude.

In the case of Nouakchott, I_r becomes 522.4 W/m^2 instead 496.6 W/m^2 (so a gain of 5.2% on I_r). From the same type of calculation as in §5.3, it is found $Pen1 = 7.89 \text{ W/m}^2$ instead 7.17 W/m^2 (so a gain of 10.0% on $Pen1$). Moreover, the variation of the generated net power throughout the year will be smaller, which is an advantage for the thermal energy conversion unit, because it will have to work with a reduced variation of the physical parameters.

Let's calculate the necessary volume flowrate Q_{vlco} through a standard collector to have the necessary ΔTs , such that $\Delta Ts = T_{so} - T_{si} = 5.46^\circ\text{C}$ (according to §4.3) with a mean $T_{s0} = 76.7^\circ\text{C}$ (cf. §5.3). So $T_{si} = 71.2^\circ\text{C}$ and

$$T_m = \frac{T_{so} + T_{si}}{2} = 74.0^\circ\text{C}.$$

Now, the mean heat power $Phco1$ delivered by 1 m^2 of heat-absorbing surface collector is equal to 203.3 W/m^2 (cf. §5.3).

A standard collector has a heat-absorbing surface collector Sco of about 2 m^2 .

So the real heat power Pco generated by a collector is equal to

$$Pco = 2 \times Phco1 = 406.6 \text{ W} \quad \text{for our example.}$$

The transfer heat through the collector is equal to $Pco = Q_{mlco} \times Cp \times \Delta Ts$ with Cp about 4197 J/kg at T_m , and Q_{mlco} the mass flow rate through a collector.

$$Q_{mlco} = Q_{vlco} \times \rho_{co} \quad \text{with } \rho_{co} \text{ the volume mass equal to } 975.3 \text{ kg/m}^3 \text{ at } T_m.$$

$$\text{So } Q_{vlco} = \frac{Pco}{\rho_{co} \times Cp \times \Delta Ts} = 1.819\text{E-}5 \text{ m}^3/\text{s}, \text{ or } 65.5 \text{ l/h ("l" for a liter), for}$$

the example.

Q_{vlco} is the volume flow rate through the collector.

One calculation condition is to have constant ΔTs across the solar thermal collectors field (5.46°C for our example).

Note 1: A ΔTs constant and equal to 5.46°C is not an absolute requirement. It depends on the way stratification and mixing occurs in the fresh water tank. Now if ΔTs is too important, there is a risk of vaporization inside the collectors, which would prevent any working. T_{so} must be widely below 100°C , let's say 90°C maximum.

Note 2: The collectors field as a hydraulic net must be balanced in term of pressure loss, so to have about the same flow rate through all the collectors. This will prevent any vaporization due to a local weak flow rate.

In the example, it is found $Q_{vlco} = 65.5 \text{ l/h}$ for $Phco1 = 203.3 \text{ W/m}^2$ and the mean value $I_r = 496.6 \text{ W/m}^2$ during the 12 h of daylight period (cf. §5.2). But I_r is variable according to the day time (see §6.3) and to the moment on the year (see §6.4). So Q_{vlco} can vary from several l/h up to let's say 200 l/h , by means of

the fresh water extraction pump which is supposed to be a variable flow rate pump (cf. §4.2).

Let's suppose that $I_r(w) = 780.1 \times \cos(w)$ as calculated in §6.3. From I_r maximum = 780.1 W/m², a calculation done as in §5.3 gives $\eta_{co} = 0.544$, $Ph_{co1} = 424$ W/m² and $Q_{v1co} = 136.7$ l/h

In these conditions, the pumping mechanical power necessary (P_{p1co}) for one collector is equal to: $P_{p1co}(W) = Q_{v1co}(m^3/s) \times \Delta P_{1co}(Pa)$

About ΔP_{1co} , let's suppose that, roughly, for a standard collector

$$\Delta P_{1co}(Pa) = 1E12 \times Q_{v1co}^2 \quad (40)$$

with Q_{v1co} in m³/s. For a supposed maximum flowrate Q_{v1co} equal to 136.7 l/h (i.e. 3.80E-5 m³/s):

- $\Delta P_{1co}(Pa) = 1E12 \times Q_{v1co}^2 = 1442$ Pa
 - $P_{p1co} = 1E12 \times Q_{v1co}^3 = 0.055$ W
- (41)

A simulation on a whole daylight (12 h) gives a mean $P_{p1co} = 0.019$ W, which corresponds to

$$Q_{v1co} = \sqrt[3]{\frac{P_{p1co}}{1E12}} = 2.66E-5 \text{ m}^3/\text{s} \quad \text{and to} \quad \Delta P_{1co} = 709 \text{ Pa} \quad (42)$$

Let's suppose two collectors in parallel (4 m²). The total volume flowrate will be the double for the same ΔP_{1co} . So the mean mechanical power necessary will be the double (0.038 W for the example).

Let's suppose two collectors in series (4 m²), as shown in **Figure 6**. In this case, for one collector:

- The flowrate $Q_{v1cos} = 2 \times Q_{v1co}$ (=5.32E-5 m³/s for the example), to keep the same ΔT_s ,
- $\Delta P_{1cos} = 4 \times \Delta P_{1co}$ (2836 Pa for the example),
- $P_{p1cos} = 8 \times P_{p1co}$ (0.15 W for the example).

For both collectors in series, the mean $\Delta P_{2co} = 2 \times \Delta P_{1cos}$ (5672 Pa for the example) and the mean mechanical power necessary $P_{p2co} = 2 \times P_{p1cos}$ (0.30 W for the example).

Let's suppose a configuration similar to the one shown in **Figure 6**:

- All the collectors are grouped two by two in series.
- All these groups of two collectors are in parallel.

Moreover, it will be supposed that the pressure loss in the pipes upstream and downstream the group of two collectors (4 m²) will be equal to the pressure loss across this group. So $\Delta P_s = 2 \times \Delta P_{2co}$ (11,344 Pa for the example) and the mean mechanical power (given by the fresh water extraction pump) necessary for a group of two collectors becomes $P_{ps} = 2 \times P_{p2co}$ (0.60 W for the example).

Of course, this power is consumed only during the 12 hours of daylight.

Note: the pipes must be heat-insulated to limit the heat loss.

Now, it can be of economic interest to have several collectors in series to mi-

The hot fresh water at $T_{so} = T_{hi} = 76.7^\circ\text{C}$ (cf. §5.3) enters the desalination unit by the SG (steam generator), which makes boil the sea water under vacuum at T_{sat} . The fresh water leaves the SG at for example $T_{ho} = T_{hi} - 2.85^\circ\text{C}$ as for the Rankine SG (cf. §3.3). Let's suppose that $T_{hi} - T_{sat} = 5.46^\circ\text{C}$ as for the Rankine SG (cf. §3.3).

So $T_{sat} = 71.24^\circ\text{C}$ and $P_{sat} = 0.3287$ bar. This steam is condensed in the condenser cooled by the sea water. The condensed water (at 71.24°C) is extracted by the condensed water extraction pump towards either the fresh water tank (initial filling or level control) or for other uses such as drinking water, irrigation, etc.

Let's suppose that a ratio $R_{cw} = 20\%$ of the mean available heating power (Ph_{av} in W) is used for the desalination unit and 80% is used for electricity production. For 1 m^2 of heat-absorbing surface collector, the mean heat power supplied Ph_{col} is equal to 203.3 W/m^2 during $Dp = 12$ hours (§5.3).

With Ph_{av1} equal to Ph_{av} for 1 m^2 of heat-absorbing surface collector

$$(Ph_{av} = Ph_{av1} \times S_{co}), \quad Ph_{av1} = Ph_{col} \times \frac{Dp}{24} = \frac{Ph_{col}}{2} = 101.65\text{ W/m}^2$$

Consequently, the power

$$Ph_{codu} = Ph_{av} \times R_{cw} \quad (43)$$

and for 1 m^2 of heat-absorbing surface collector:

$$Ph_{codu1} = \frac{Ph_{codu}}{S_{co}} = Ph_{av1} \times R_{cw} = 20.33\text{ W/m}^2 \quad (44)$$

will be used, day and night, to produce condensed water by the desalination unit, whereas

$$Ph_1 = Ph_{av1} \times (1 - R_{cw}) = 81.32\text{ W/m}^2 \quad (45)$$

will be used, day and night, to produce electricity by the thermal energy conversion unit.

The heat of vaporization L_v at $T_{sat} = 71.24^\circ\text{C}$ is equal to 2331 kJ/kg . So the condensed water mass flow rate for 1 m^2 of collector is equal to

$$Q_{mcw} = \frac{Ph_{codu1}}{L_v + (C_p \times (T_{sat} - T_{ci}))} \quad (46)$$

$T_{ci} = 22.0^\circ\text{C}$ (cf. §5.2). C_p for a temperature between T_{ci} and T_{sat} is equal to about 4183 J/kg .

For the example, $Q_{mcw} = 8.013\text{E-}6\text{ kg/s}$ and the daily production of condensed water for 1 m^2 of collector (Q_{mdcw}) will be equal to:

$$Q_{mdcw} = Q_{mcw} \times (3600 \times 24) = 0.692\text{ kg/day/m}^2 \quad (47)$$

8.2. Estimation of the Electric Power Consumed for 1 m^2 of Heat-Absorbing Surface Collector (Pe_{adu1})

About the fresh water circuit through the steam generator

The fresh water mass flow rate through the desalination unit steam generator (Q_{mdu}) is such that:

$Phcodu1 = Qmdu \times Cp \times (Thi - Tho)$ with $Thi = 76.6^\circ\text{C}$, $Thi - Tho = 2.85^\circ\text{C}$ (cf. §3.3 and §8.1) and Cp about 4198 J/kg. So

$$Qmdu = \frac{Phcodu1}{Cp \times (Thi - Tho)} = 1.699\text{E}-3 \text{ kg/s}$$

The volume flow rate $Qvdu = \frac{Qmdu}{\rho du}$ with ρdu the volume mass equal to 974.7 kg/m³. So $Qvdu = 1.743\text{E}-6 \text{ m}^3/\text{s}$

The mechanical power $Pmdu1$ necessary for the fresh water circulation pump (**Figure 9**) to pump this flow rate is equal to $Pmdu1 = Qvdu \times \Delta P du$ with the pressure drop $\Delta P du$ through the pipes leading the fresh water to the steam generator supposed equal to 0.3 bar. So $Pmdu1 = 5.23\text{E}-2 \text{ W/m}^2$.

About the condensed water circuit

The mechanical power $Pmcw1$ necessary for the condensed water extraction pump to pump the condensed flow rate $Qvcw$ is equal to $Pmcw1 = Qvcw \times \Delta P cw$; $Qvcw = \frac{Qmcw}{\rho cw}$ with ρcw the volume mass equal to 976.9 kg/m³.

So $Qvcw = 8.20\text{E}-9 \text{ m}^3/\text{s}$

With the pressure drop $\Delta P cw$ through the pipes leading the condensed water to the fresh water tank or to the consumer circuits supposed equal to 0.3 bar, $Pmcw1 = 2.46\text{E}-4 \text{ W/m}^2$.

About the sea water circuit through the condenser

The sea water mass flow rate through the desalination unit condenser ($Qmcd u$) is such that: $Qmcd u \times Cp \times \Delta Tcd u = Qmcw \times Lv(Tsat)$

So

$$Qmcd u = \frac{Qmcw \times Lv(Tsat)}{Cp \times \Delta Tcd u} \quad (48)$$

It will be considered that the $\Delta Tcd u$ across the condenser is equal to 20°C. Cp is equal to 4194 J/kg. So $Qmcd u = 2.23\text{E}-4 \text{ kg/s}$

$Qvcdu = \frac{Qmcd u}{\rho cdu}$ with ρcdu the volume mass equal to 995 kg/m³

So $Qvcdu = 2.24\text{E}-7 \text{ m}^3/\text{s}$

The mechanical power $Pmcd u1$ necessary for the sea water circulation pump (**Figure 1**) to pump the sea water flow rate through the condenser is equal to $Pmcd u1 = Qvcdu \times \Delta Pcd u$.

With the pressure drop $\Delta Pcd u$ through the sea water circuit supposed equal to 0.4 bar, $Pmcd u1 = 8.96\text{E}-3 \text{ W/m}^2$.

Electric pumping power ($Pepdu1$)

The sum of these 3 mechanical pumping powers is equal to $Pmpdu1 = Pmdu1 + Pmcw1 + Pmcd u1 = 3.15\text{E}-2 \text{ W/m}^2$. Given a global efficiency of 0.85 for these pumps, the expected electric pumping power $Pepdu1$ is equal to: $Pepdu1 = \frac{Pmpdu1}{0.85} = 7.24\text{E}-2 \text{ W/m}^2$.

Power for vacuum ($Pevdu1$)

The necessary vacuum pump and degassing unit to trap dissolved gasses are supposed to consume $P_{evd1} = 0.14\%$ of $Phcod1$ (as in §3.4.1), so $2.85E-2$ W/m² in the example.

Note: this value is pessimistic due to the high vacuum pressure ($Psat = 0.3287$ bar).

Total expected consumed electric power (P_{ead1})

Consequently, the total expected consumed electric power $P_{ead1} = P_{epd1} + P_{evd1}$ is equal to 0.101 W/m² (for 1 m^2 of heat-absorbing surface collector, with the ratio $R_{cw} = 20\%$).

9. Example of Solar Thermal Power Plant

Further, it will be considered, as an example, a solar thermal power plant equivalent, in terms of electricity production, to the Seikh Zayed photovoltaic plant located at Nouakchott, which electricity production is equal to 25,409 MWh per year (cf. [11]).

9.1. Generalities about a Solar Thermal Power Plant Equivalent to the Seikh Zayed Photovoltaic Plant

The plant is supposed to be located at Nouakchott, it will be found for 1 m^2 of heat-absorbing surface collector, as shown in §5.3, that the ideal temperature $Thi = 76.7^\circ\text{C}$. For this temperature, $\eta_{co} = 40.85\%$, $\eta_{net} = 7.05\%$, $Phcol = 203.3$ W/m² and the net electric power $Pen1 = 7.169$ W/m².

Note: this $Pen1$ value is probably pessimistic and rather equal to 8.55 W/m² (see §6.4).

Let's consider the ratio $R_{cw} = 20\%$, as chosen in §8.1. Consequently, for 1 m^2 of heat-absorbing surface collector:

- 20% of the mean heating power ($Phav1$) is used for the desalination unit, so as to produce 0.692 kg/day/m² of condensed water (cf. §8.1),
- 80% is used for electricity production. Now for $Ph1 = Phav1$ (at $R_{cw} = 0$), $Pen1$ (at $R_{cw} = 0$) = 7.169 W/m² (§5.3), so for $R_{cw} = 20\%$, $Pen1(R_{cw}) = Pen1(\text{at } R_{cw} = 0) \times (1 - R_{cw}) = 7.169 \times 0.8 = 5.73$ W/m² of surface collector.

25,409 MWh per year being the supposed net electricity production, the mean net electric power (Pen) to generate over a year is equal to:

$$Pen = \frac{25409E6}{365 \times 24} = 2.90E6 \text{ W} .$$

So, to generate $Pen = 2.90E6$ W, it would be necessary to install a heat-absorbing surface collector Sco : $Sco = \frac{Pen}{Pen1} = 5.06E5 \text{ m}^2$ (i.e. 0.506 km^2).

According to §8.1, the daily production of condensed water will be equal to $Q_{mdcw} = 0.692 \times Sco = 3.50E5$ kg/day, so 350 tons per day.

According to §6.2, the mass of water in the tank will be equal to $Mt = 95.1 \times Sco = 4.81E7$ kg or about a volume $Vt = 49,328 \text{ m}^3$. The surface of

the tank could, for example, be equal to 80 m × 80 m for a depth of 7.71 m.

According to §3.3, $\eta_{net} = \frac{Pen}{Ph}$, so $Ph = \frac{Pen}{\eta_{net}} = 4.11E7$ W. Ph is the heating power for the thermal energy conversion unit. It can also be calculated as:

$$Ph = \frac{Ph_{col}}{2} \times S_{co} \times (1 - R_{cw}) \quad (49)$$

9.2. Application to the Rankine and Claude Cycles

Peat corresponds to the electric power consumed by all the pumps and auxiliaries of the plant (called “*Pea*” in §3.3 and §3.4). Now, Peat includes the desalination unit and the fresh water extraction pump. Hypothetically, *Peat* will be limited to $6.958E-3 \times Ph$ as for *Pea* in §3.3 and §3.4. This maximum (*Peat max*) must be considered as a reasonable objective. So in the example, for $Ph = 4.11E7$, *Peat max* = 2.86E5 W (0.286 MW).

The expected $Pen = 2.90E6$ (§9.1) will be really available if *Peat* (calculated) ≤ *Peat max*. If it is not the case (i.e., *Peat* > *Peat max*), Pen will have to be reduced by the difference *Peat* - *Peat max*.

9.2.1. Calculation of Pea for the Thermal Energy Conversion Unit Supposing a Rankine Cycle

It is reminded that:

- For $Thi = 76.7^\circ\text{C}$, $T3 = 71.24^\circ\text{C}$ and $P3 = P2 = 34.07$ bar (see §5.3).
- For $Tci = 22.0^\circ\text{C}$, $Tco = 26.25^\circ\text{C}$, $T4 = 28.75^\circ\text{C}$, $T2 \approx T1 \approx T4$, $P4 = 11.24$ bar, $P1 \approx P4$ (see §5.2).

Below, *Pea* is calculated in the same way as in §3.3.1.

About the working fluid circuit

The mechanical power P_{mwf} necessary for the compressor (**Figure 4**) to pump the working fluid in liquid state is equal to: $P_{mwf} = Q_{vwf} \times (P2 - P1)$,

with $Q_{mwf} = \frac{Ph}{H_{sv}(T3) - H_l(P2, T2)}$. For this example, $Q_{mwf} \approx 35.6$ kg/s. So

with a liquid volume mass $\rho_{wf} \approx 597.1$ kg/m³, the liquid volume flow rate:

$Q_{vwf} = \frac{Q_{mwf}}{\rho_{wf}} = 0.060$ m³/s for the example. So the mechanical power P_{mwf} is

equal to 1.36E5 W for the example.

About the fresh water circuit

The mechanical power P_{mh} necessary for the fresh water circulation pump (**Figure 1**) to pump the fresh water through the steam generator is equal to: $P_{mh} = Q_{vh} \times \Delta Ph$, Q_{vh} is proportional to Ph (with C_p and ρ_h supposed constant) so, referenced to §3.3, $Q_{vh} = 3.43$ m³/s. However, a more precise calculation

based on $Q_{mh} = \frac{Ph}{C_p \times (Thi - Tho)}$ with $Thi - Tho = 2.86^\circ\text{C}$ (§3.3 and §5.3) and

on $Q_{vh} = \frac{Q_{mh}}{\rho_h}$ gives $Q_{vh} = 3.52$ m³/s. ΔPh is supposed equal to 0.3 bar.

Therefore $P_{mh} = 1.06E5$ W for the example.

About the sea water circuit

The mechanical power P_{mc} necessary for the sea water circulation pump to pump the sea water through the condenser is equal to: $P_{mc} = Q_{vc} \times \Delta P_c$. Q_{vc} is approximately proportional to Ph (with C_p and ρ_c supposed constant) so $Q_{vc} = 2.23 \text{ m}^3/\text{s}$. ΔP_c is supposed equal to 0.4 bar. So $P_{mc} = 8.93\text{E}4 \text{ W}$ for the example.

Electric pumping power (P_{ep})

The sum of these 3 mechanical pumping powers is equal to

$$P_{mp} = P_{mwf} + P_{mh} + P_{mc} = 3.31\text{E}5 \text{ W} .$$

Given a global efficiency of 0.85 for these pumps, the expected electric pumping power P_{ep} is equal to: $P_{ep} = \frac{P_{mp}}{0.85} = 3.89\text{E}5 \text{ W} .$

Reminder power (P_{er})

Let's suppose that the electric power P_{er} consumed by the different auxiliaries, control, light of the plant, etc., is proportional to Ph . In §3.3 $Ph = 7.186\text{E}8 \text{ W}$ for $P_{er} = 0.76\text{E}6 \text{ W}$. So here for $Ph = 4.11\text{E}7 \text{ W}$, $P_{er} = 4.41\text{E}4 \text{ W}$.

Total expected consumed electric power (P_{ea})

The total expected electric power P_{ea} is equal to $P_{ea} = P_{ep} + P_{er} = 4.33\text{E}5 \text{ W}$

9.2.2. Calculation of P_{ea} for the Thermal Energy Conversion Unit Supposing a Claude Cycle

It is reminded that:

- For $Thi = 76.7^\circ\text{C}$, $T3 = Tho = 71.24^\circ\text{C}$ and $P3 = 0.3287 \text{ bar}$ (with $P2 \approx P3 + \Delta P_{wf}$) (see §5.3).
- For $Tci = 22.0^\circ\text{C}$, $Tco = 26.25^\circ\text{C}$, $T4 = 28.75^\circ\text{C}$, $T2 \approx T1 \approx T4$, $P4 = 0.0395 \text{ bar}$, $P1 \approx P4$ (see §5.2)

Below, P_{ea} is calculated in the same way as in §3.4.1.

About the working fluid circuit towards the fresh water tank

The mechanical power P_{mwf} necessary for the working fluid circulation pump (**Figure 5**) to pump the working fluid in liquid state is equal to: $P_{mwf} = Q_{vwf} \times (P2 - P1)$ (see §3.4.1), with $P2 - P1 \approx P3 - P1 + \Delta P_{wf}$ and $Q_{mwf} \approx \frac{Ph}{L_v(Tho)}$. For this example, $Q_{mwf} \approx 17.6 \text{ kg/s}$. So with a liquid volume mass $\rho_{wf} \approx 976.9 \text{ kg/m}^3$, the liquid volume flow rate:

$$Q_{vwf} = \frac{Q_{mwf}}{\rho_{wf}} = 0.0181 \text{ m}^3/\text{s} \text{ for the example.}$$

It is supposed that the pressure drop ΔP_{wf} through the pipes leading to the fresh water tank through the Tee (see **Figure 5**) is equal to 0.15 bar. So the mechanical power P_{mwf} is equal to 793 W for this example.

About the fresh water circuit towards the fresh water tank

The mechanical power P_{mho} necessary for the extraction pump (**Figure 5**) to pump the fresh water is equal to: $P_{mho} \approx Q_{vh} \times \Delta P_{ho}$.

Now $Q_{mh} = \frac{Ph}{C_p \times (Thi - T_{mix})} \approx \frac{Ph}{C_p \times (Thi - Tho)}$ with $Thi - Tho = 5.46^\circ\text{C}$ (§3.4 and §5.3).

$$Q_{vh} = \frac{Q_{mh}}{\rho h} \quad \text{With } C_p = 4197 \text{ J/kg and } \rho h = 975.3 \text{ kg/m}^3, Q_{vh} = 1.84 \text{ m}^3/\text{s}.$$

It is supposed that the pressure drop ΔP_{ho} through the pipes leading to the fresh water tank through the Tee (see **Figure 5**) is equal to 0.15 bar. The mechanical power P_{mho} is equal to 2.76E4 W for the example.

About the fresh water circuit from the fresh water tank

The mechanical power P_{mhi} necessary for the fresh water circulation pump (**Figure 1**) to pump the fresh water through the evaporator is equal to: $P_{mhi} = Q_{vh} \times \Delta P_{hi}$.

It is supposed that the pressure drop ΔP_{hi} through the pipes coming from the fresh water tank towards the evaporator is equal to 0.15 bar. So the mechanical power $P_{mhi} = P_{mho} = 2.76E4$ W for the example.

About the sea water circuit through the condenser

The mechanical power P_{mc} necessary to pump the sea water through the condenser is the same as for the Rankine cycle (§9.2.1) (*i.e.* $P_{mc} = 8.93E4$ W for the example).

Electric pumping power (P_{ep})

The sum of these 4 mechanical pumping powers is equal to

$$P_{mp} = P_{mwf} + P_{mho} + P_{mhi} + P_{mc} = 1.45E5 \text{ W}.$$

Given a global efficiency of 0.85 for these pumps, the expected electric pumping power P_{ep} is equal to: $P_{ep} = \frac{P_{mp}}{0.85} = 1.75E5 \text{ W}$.

Power for vacuum (P_{ev})

The necessary vacuum pumps and degassing units to trap dissolved gasses are supposed to consume $P_{ev} = 0.14\%$ of Ph , so 5.76E4 W for the example.

Reminder power (P_{er})

Let's suppose that the electric power P_{er} consumed by the different auxiliaries, control, light of the plant, etc is proportional to Ph . In §3.4.1 $Ph = 7.186E8$ W and $P_{er} = 1.08E6$ W.

So here for $Ph = 4.11E7$ W, $P_{er} = 6.18E4$ W.

Total expected consumed electric power (P_{ea})

The total expected electric power P_{ea} is equal to

$$P_{ea} = P_{ep} + P_{ev} + P_{er} = 2.90E5 \text{ W}$$

9.2.3. About the Desalination Unit

As calculated in §8.2, for 1 m² of heat-absorbing surface collector and for the ratio $R_{cw} = 20\%$, the total expected consumed electric power (P_{eadu1}) is equal to 0.101 W. So for $S_{co} = 5.06E5$ m² (§9.1), the total expected consumed electric power P_{eadu} is equal to 5.10E4 W.

9.2.4. About the Fresh Water Pumping Station Associated to the Solar Collectors Field

Let's suppose a solar collectors field configuration similar to the one shown in **Figure 6**. In Section 7, it has been determined that the mean mechanical power necessary for the fresh water extraction pump (**Figure 6**) to pump the fresh wa-

ter through two collectors in series (4 m^2) is equal to 0.60 W , during the $Dp = 12$ hours of the daylight, which is equivalent to $0.60 \times \frac{Dp}{24} = 0.30 \text{ W}$ during 24 hours.

So for $Sco = 5.06E5 \text{ m}^2$ (§9.1), the mean mechanical power is equal to

$$P_{msc} = \frac{0.30 \times Sco}{4} = 3.79E4 \text{ W}$$

Given a global efficiency of 0.85 for the fresh water extraction pump, the expected electric pumping power $Pesc$ is equal to $Pesc = \frac{P_{msc}}{0.85} = 4.46E4 \text{ W}$.

9.2.5. About the Total Auxiliary Electric Power $Peat$ Consumed Supposing a Rankine Cycle and about the Real Pen

From the previous results, it can be determined $Peat$.

$$Peat = Pea(\text{for the Rankine cycle}) + Peadu + Pesc = 5.29E5 \text{ W}$$

$Peat$ is widely superior to the $Peat_{max} = 2.86E5 \text{ W}$ (§9.2)

The expected electric power Pen was supposed equal to $Pen = 2.90E6$ (§9.1), but in fact the real $Pen = 2.90E6 - Peat + Peat_{max} = 2.66E6 \text{ W}$ (so a loss of 8.4%).

9.2.6. About the Total Auxiliary Electric Power $Peat$ Consumed Supposing a Claude Cycle and about the Real Pen

$$Peat = Pea(\text{for the Claude cycle}) + Peadu + Pesc = 3.86E5 \text{ W}$$

$Peat$ is superior to the $Peat_{max} = 2.86E5 \text{ W}$ (§9.2)

The expected electric power Pen was supposed equal to $Pen = 2.90E6$ (§9.1), but in fact the real $Pen = 2.90E6 - Peat + Peat_{max} = 2.80E6 \text{ W}$ (so a loss of 3.4%), which is better than the Pen supposing a Rankine cycle, and close to the objective ($Pen = 2.90E6 \text{ W}$).

10. Results and Discussion

It has been shown (in Section 2) that it exists, in different zones of the world, conditions favorable to a solar thermal power plant at low temperatures using solar thermal collectors. The obvious favorite coast areas are the warm ones (between about -40° to 40° of latitude). Now beyond the ambient air and the sea water temperatures, a very important parameter is the clearness index (Cl_i). See examples in Appendix.

To take profit of the difference of temperatures between the fresh water heated by collectors (around 77°C) and the surface sea water (around 22°C), it has been studied (in Section 3) the OTEC Rankine and Claude cycles. Even if the Rankine cycle is better mastered than the Claude cycle, this last one has advantages, at least in the absence of toxicity risk.

In Section 4, the hot source (solar thermal collectors field) has been described and the type “glazed flat plate collectors” has been selected.

From the previous data, taking as example the Nouakchott area, the mean

values for a continuous working over the year have been determined in Section 5, especially the ideal mean hot temperature Thi ($=76.7^{\circ}\text{C}$).

The continuous working (day and night, all year round) by means of a fresh water storage which acts as a thermal energy reserve is described in Section 6. In this chapter, it is shown that for Nouakchott two m^2 of solar collectors generate about the same average electricity production over the year than one m^2 of photovoltaic panel.

The solar collectors field configuration has been analyzed in Section 7.

The desalinated water production through a vacuum distillation desalination unit has been described in Section 8.

An example of solar thermal power plant has been proposed in Section 9. It takes as objective the same initial electric production as the Seikh Zayed photovoltaic plant located at Nouakchott, which corresponds to a mean net electric power to generate over a year equal to 2.90 MW. Moreover, it is produced 350 tons of desalinated water per day.

The Rankine and the Claude cycles have been considered. Compared to the Rankine cycle, the Claude has a better result, very close to the objective (see §9.2.5 and §9.2.6). Taking also into account the absence of toxicity of its working fluid (fresh water), it must be considered as the preferred cycle for this type of plant.

In Appendix, it is proposed the description of a small program able to calculate the net electric power provided to the grid net, for 1 m^2 of heat-absorbing surface collector. Three examples in different locations are given.

They show that, compared to collectors horizontally installed, the results are better when they are inclined by an angle equal to the latitude, above all for relatively high latitudes (see the third example in Appendix), in both aspects:

- The net electric power averaged over the year is always superior,
- The net electric power varies much less all along the year which is favorable for the machinery and for the electric net production management.

It must also be noted that if the collectors are horizontally installed, the main parameter is the latitude whereas if the collectors are inclined the main parameter is the clearness index.

Now the installation of such inclined collectors is obviously less simple and cheap than the installation of horizontal collectors.

11. Conclusions

It has been shown that this concept could work. Compared to a photovoltaic plant, the solar thermal power plant at low temperature has two main advantages:

- It produces electricity continuously, all year round.
- It produces desalinated water for drinking and irrigation.

According to Section 10, the best configuration is:

- A Claude cycle for the electricity production,
- Collectors are inclined by an angle equal to the latitude.

Moreover, the required technology for most of the equipment is relatively simple, so most of the repairs could be done locally.

Conflicts of Interest

The author declares no conflict of interest regarding the publication of this paper.

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Appendix

A small and simple program (“freeware”) has been written by the author. It is called “Solar_Plant” (version 1.0). It is based on the equations of this article plus an extension of these equations for the inclined collectors.

The results must be considered as rough, but sufficient for a quick evaluation. The program is written in Delphi 6 (*i.e.*, Pascal Object). The Delphi 6 code remains proprietary.

It is intended to calculate the net electric power provided to the grid net (without desalinated water production). It will be enough for the user to multiply by the effective collectors surface (Sc_o) to have the net electric power wished. The goal is to have an idea of the possibility of electricity production for a given place. An average net power over the year superior or equal to 7 W/m^2 with inclined collectors must be considered as interesting. These results apply, qualitatively, to the desalinated water production.

The last version of Solar_Plant (version 1.0 at the publication of this article) is proposed to download from this direct link: http://f6cte.free.fr/Solar_Plant.zip. It is enough to paste this address in your Internet browser. Download the file. Create a folder (C:\Solar for example), unzip the Solar_Plant.zip file in it, start C:\SOLAR\Solar_Plant.exe (the auxiliary file CONFSOLAR.SER will be created automatically).

The first calculation is the clearness index in %, function of GHI and the latitude. Then, for each period (December, June and equinoxes), it is calculated the net electric power provided to the grid net, for 1 m^2 of heat-absorbing surface collector ($Pen1$ in W/m^2) and the ideal Thi , using a calculated mean Ir as in §5.3. Then supposing Ta and Thi fixed, it is determined Ir function of the time of day (in a similar way as in §6.3). So a more precise mean value of $Phco1$ is calculated, as in §6.3. Supposing the same net efficiency η_{net} , it is, afterwards, determined the new value of $Pen1$, by a simple rule of three. The proposed $Pen1$ values are the expected ones, *i.e.*, for $Peat = Peat \text{ max}$, as explained in §9.2.

Note: for December and June, Ir is calculated at the solstice (21th of December and 21th of June in 2022).

To make work this program, the user must, for a given place on the coast:

- Give the GHI (Global horizontal irradiance) in kWh/m^2 (from WEB data, as on [1]).
- Give the latitude, from -65° (South) to 65° (North).
- Give the maximum and the minimum air temperatures plus the sea water temperature for December (from WEB data, as on [3] and [4]).
- Give the maximum and the minimum air temperatures plus the sea water temperature for June (from WEB data, as on [3] and [4]).
- Click on “Calculation” to see the results. First is given the calculated clearness index in % and then the net electric powers per m^2 , as detailed below.
 - First part, net electric power ($Pen1$ in W/m^2) for the collectors in-

stalled horizontally:

- In December and in June, to see the difference,
- Probable mean value over the year, using the average temperatures between June and December.
- Second part, net electric power ($Pen1$ in W/m^2) for the collectors inclined with an angle equal to the latitude.
 - In December and in June, to see the difference,
 - At both equinoxes, using the average temperatures between June and December.
 - Probable mean value over the year, using the three previous $Pen1$ results (December/June/Equinoxes).

Three examples are given below (two in the North hemisphere and one in the South hemisphere). Note that for the third example, the latitude is relatively high (37°).

Example of results in the North hemisphere, for Nouakchott (configuration by default):

Data:

GHI = 2175 kWh/m² Latitude = 18° .

In December: T min air = $20^\circ C$ /T max air = $29^\circ C$ /T sea = $20^\circ C$.

In June: T min air = $23^\circ C$ /T max air = $33^\circ C$ /T sea = $22^\circ C$.

Results: Clearness index = 62.5%.

*Collectors installed horizontally—Net electric power in W/m^2 :

December = 5.16/June = 9.27/Average over the year = 8.29.

*Inclined collectors (angle = latitude)—Net electric power in W/m^2 :

December = 8.32/June = 7.43/Equinoxes = 9.18/Average over the year = 8.53.

Example of results in the South hemisphere, for Lüderitz in the South of Namibia:

Data:

GHI = 2200 kWh/m² Latitude = -27° .

In December: T min air = $14^\circ C$ /T max air = $25^\circ C$ /T sea = $17^\circ C$.

In June: T min air = $11^\circ C$ /T max air = $22^\circ C$ /T sea = $15^\circ C$.

Results: Clearness index = 67.1%.

*Collectors installed horizontally—Net electric power in W/m^2 :

December = 11.85/June = 3.37/Average over the year = 7.99.

*Inclined collectors (angle = latitude) —Net electric power in W/m^2 :

December = 9.11/June = 8.08/Equinoxes = 10.02/Average over the year = 9.31.

Example of results in the North hemisphere, for Almeria in the South of Spain (Andalusia):

Data:

GHI = 1750 kWh/m² Latitude = 37° .

In December: T min air = $10^\circ C$ /T max air = $18^\circ C$ /T sea = $17^\circ C$.

In June: T min air = $19^\circ C$ /T max air = $28^\circ C$ /T sea = $21^\circ C$.

Results: Clearness index = 58.7%.

*Collectors installed horizontally—Net electric power in W/m^2 :

December = 1.29/June = 7.97/Average over the year = 4.66.

*Inclined collectors (angle = latitude)—Net electric power in W/m^2 :

December = 6.45/June = 6.14/Equinoxes = 7.37/Average over the year = 6.83.

Observations

- The best configuration is the inclined collectors one: higher “Average over the year” net electric power and weaker differences between December/June/Equinoxes.
- From these examples, it is obvious that the main parameter in the inclined collectors configuration is the clearness index which must be the highest possible. For example, the result from Lüderitz ($9.31 \text{ W}/\text{m}^2$) is superior to the one from Nouakchott ($8.53 \text{ W}/\text{m}^2$), even if the absolute latitude is not so favorable (27° versus 18°), this because the clearness index is equal to 67.1% for Lüderitz versus 62.5% for Nouakchott. The result for Almeria ($6.83 \text{ W}/\text{m}^2$) is not so good because the clearness index is not elevated (58.7%). Moreover, the advantage of this configuration compared to the horizontal installation increases with the latitude: the difference is small at Nouakchott (8.53 versus $8.29 \text{ W}/\text{m}^2$) but strong at Almeria (6.83 versus $4.66 \text{ W}/\text{m}^2$).
- The two main parameters are the absolute latitude and the clearness index. However, it can be observed that if the collectors are horizontally installed, the most influential parameter is the latitude, *i.e.* for a constant clearness index, the average net electric power decreases when the absolute latitude increases. Reversely if the collectors are inclined the most influential parameter is the clearness index.
- Note that the extraterrestrial solar irradiation (I_{er}) is superior in December than in June, due a weaker distance from the Sun, which explains the better result in December than in June, in the inclined collectors configuration.