Proposal relative to an aeraulic thermosiphon able to generate electricity using the difference of temperatures between the ambient air and the surface sea water

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Abstract: in this article, it is envisaged different possibilities to generate electric energy from the difference of temperatures between the ambient atmospheric air and the surface sea water, which is a renewable source of energy. Finally, it is proposed a machine based on an aeraulic thermosiphon, installed between the seashore and a high relief (300 m at least). This electric generation would be done without insoluble constraints on the refrigerant. Unfortunately, the cost of such machine is not competitive compared to a wind turbine, which prevents any commercial use. So, at least, it can be hoped that this paper will give news ideas to the readers, for a possible competitive machine.

CONTENTS

1. Goal, history and notations used

The goal of this presentation is to describe a new type of electric energy generation plant taking its source in the difference of temperatures between the ambient atmospheric air and the surface sea (or ocean) water. Such plant based on an aeraulic thermosiphon (see diagram in figure 7) might be installed between the seashore and the top of a hill (300 m of altitude at least) close to the seashore. For a warm water / cold air configuration:

- The turbine alternator group and the refrigerant (dry compressed air) - warm water exchanger will be installed on the seashore, with a pipe extracting sea water from low depth and another pipe forcing the water back to the sea, at the same depth.
- The refrigerant - cold ambient air exchanger will be installed on top of the hill.

For a cold water / warm air configuration, the refrigerant (compressed air) - warm ambient air exchanger and the turbine alternator group would be located on the seashore and the refrigerant – cold water exchanger would be located on top of the hill.

In this paper, it will be only considered the warm water / cold air configuration, because the cold water / warm air configuration is symmetrical and consequently does not need more explanations.

The sole reference found by the author relatively to such use is an article written in 1930 (see reference [7]). This article (including preliminary unfortunate digressions) proposes a closed system working with butane (insoluble in water) as refrigerant. This one, cold and liquid, is vaporized in the boiler by the sea water at 0°C (which is transformed in ice). This butane vapor makes works a turbine. Then it is cooled down in the condenser by “cryh” (salt mixed with ice so that the melting point is -20.5 °C) itself cooled down by the atmosphere supposed at -23.3 °C. This system seems, at least, limited by the capacity to cool the “cryh”.

A close domain is the “Ocean Thermal Energy Conversion” (OTEC), for which energy is extracted between the warm surface water and the cold water (at 4°C) at about 1000 m depth. The difference of temperatures is about 20 °C, whereas for the way proposed here, this difference is lower and around 14 °C

This domain could be called, by reference to OTEC: “Water - Atmosphere Thermal Energy Conversion”.

Notations

- “P” for an absolute pressure in Pa or in bar
- “T” for a temperature in °C or °K (°K=°C+273.15)
- “Z” or “A” for the altitude
- “v” for the specific volume in m^3/kg
- “k” for the isentropic exponent of a perfect gas (=1.4 for air)
- “s” for specific entropy in J/(°K.kg)
- “W_{xyz}” for a power in W
- Qm for mass flow rate in kg/s
- Qv for volume flow rate in m^3/s
- Δ (“delta”) for any difference (altitude, pressure, etc)
- [x] for “reference number x”, the references list being available at the end of this article

SI units, multiples (bar for example) and sub-multiples (mm for example) are only used.

2. Principle

In different zones of the world, it exists relatively strong differences of temperature between the ambient air and the sea or ocean surface water for coastal areas (see §3). This condition can be extended to areas along big river, i.e. with a big flowrate, as for example the Saint-Laurent River in Canada (Quebec province).
Thanks to this difference of temperatures, it can be envisaged to set up plants generating electricity through

a thermodynamic cycle working between the air and the water temperatures. Note that the ambient air can be either warmer or cooler than the sea water.
Even if the energy efficiency of such cycle is very low, the source of heat (i.e. indirectly the solar radiation) is extremely important.
See the figure 1 below, for the principle.

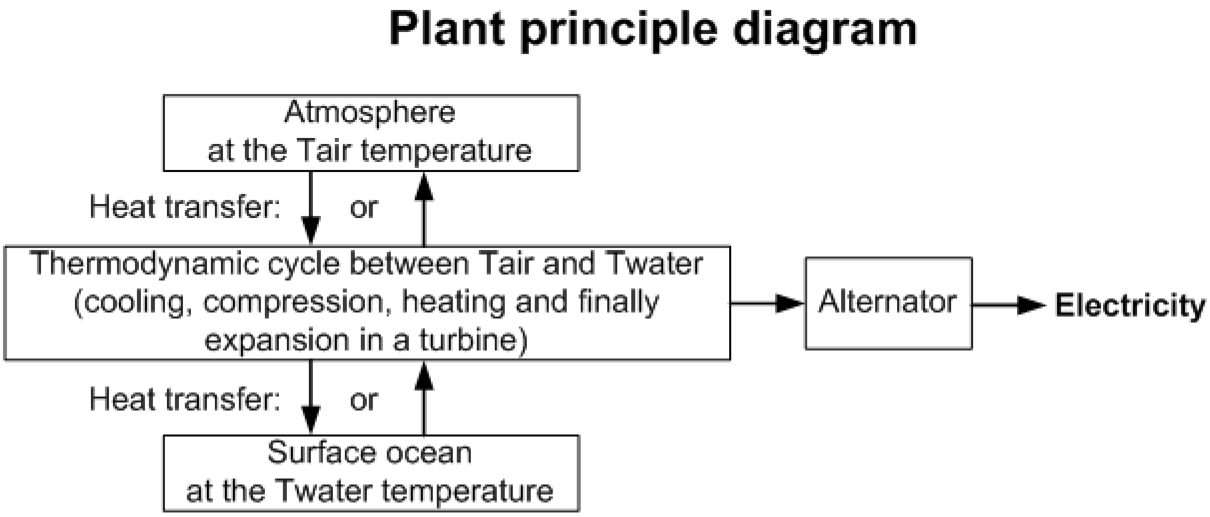

Figure 1

As shown in the figure 1, the plant can work in two configurations:

- Cold air / warm water.
- Warm air / cold water.

The sole first configuration will be addressed in this paper, as the second one is symmetrical of the first one.

3. Concerned areas

Below are given two extreme examples of temperatures.

Here is a first example in a very cold area, i.e. the temperatures (in °C) at Narsaq, in the south of the Greenland (issued from [1] and [2]).

Month	J	F	M	A	M	J	J	A	S	O	N	D
Max air temperature (Tair day)	-21	-23	-22	-15	-5	1	4	2	-5	-10	-15	-19
Min air temperature (Tair night)	-26	-28	-28	-22	-13	-4	0	-1	-7	-13	-18	-23
Ocean temperature (Twater)	-0.6	-1	-0.8	-0.9	-0.5	-0.6	1.7	2.7	1.7	1.2	-0.2	-0.6
Maximum difference (Twater-Tair day or night)	25	27	27	21	12	3	2	3	9	14	18	22

Here is a second example in a very hot area, i.e. the temperatures (in °C) at Nouakchott, capital of Mauritania (issued from [3] and [4]):

Month	J	F	M	A	M	J	J	A	S	O	N	D
Max air temperature (Tair day)	29	29	31	32	33	33	32	32	34	35	32	29
Min air temperature (Tair night)	18	19	20	21	23	24	25	27	28	28	24	20
Ocean temperature (Twater)	18.3	18	18.5	18.6	19.1	21.8	25.6	27.5	28.3	25.9	22.4	20.1
Maximum difference (Twater-Tair day or night)	-11	-11	-12	-13	-14	-11	-6	-5	-6	-9	-10	-9

If a minimum absolute difference of 9° C would be required to get a sufficient yield, it means that a production of electricity, for a part of the day, during 9 months at Narsaq and at Nouatchott would be possible.

In more temperate areas, the maximum difference could be either positive or negative, as for example in Sapporo (Japan) (issued from [5] and [6]):

Month	J	F	M	A	M	J	J	A	S	O	N	D
Max air temperature (Tair day)	-4	-3	1	8	15	19	24	24	20	13	6	-1
Min air temperature (Tair night)	-10	-10	-7	1	6	11	15	16	13	6	0	-6
Ocean temperature (Twater)	6.5	5.3	5.1	7.2	10.2	14.8	18.6	21.6	20.5	16.7	12.4	8.7
Maximum difference (Twater-Tair day or night)	16	15	12	6	-5	-5	-5	6	7	11	12	15

The areas likely to have a sufficient absolute minimum difference of temperatures (let's say 9 °C) are located:

- In cold continental climate, as Pacific coast of Russia, Korea, north of Japan, east of Canada, north-east of USA, Alaska, Greenland and some places in Europe as Lapland, Oslo, etc.
  In fact, these areas could be extended, if a very high relief (let's say ≥ 800 m) dominates the coast, this thanks to the drop of air temperature with the altitude, i.e. about -0.65 °C / 100 m.

- In desert (west of Sahara, for example) or semi-desert (Agadir for example) areas bordering the ocean and having strong differences of temperature between day and night.

Moreover, in all cases, it is necessary that the coast has some relief (300 m at least).

Note that the surface ocean temperature and its temperature amplitude depend on the latitude but also on the oceanic currents (cold or warm). See the figure 2 below.

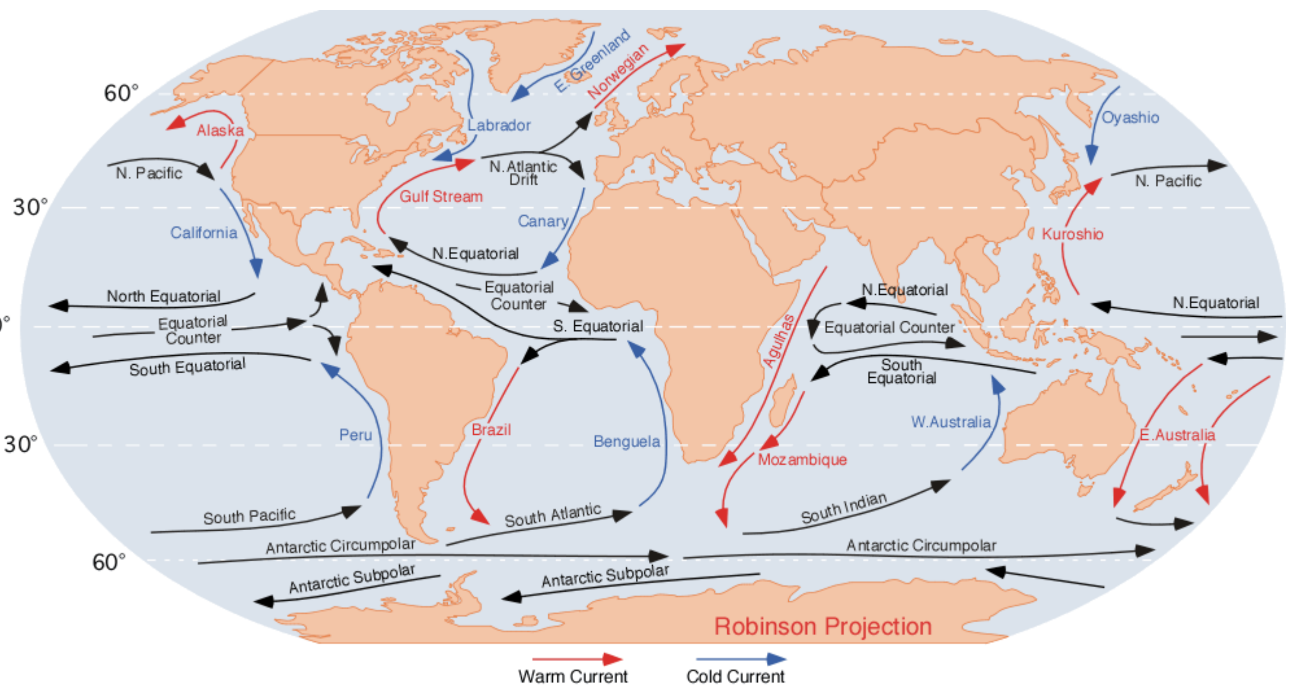

Figure 2: ocean surface currents
From Wikipedia. Author Dr. Michael Pidwirny. Public domain. U.S. government publication.

4. Types of heat exchanger used for the proposed machine

4.1 Generalities about the addressed heat exchangers

The fluids considered here are:

- ambient atmospheric air and dry compressed air for the refrigerant,
- water (from sea, ocean or big river).

In the following, it will be considered plate-type heat exchangers (air-water and air-air), inspired from the reference [9] page 683. Such exchanger can be configured as:

- a cross-flow exchanger (as on the figure 3). It is much easier to manage, relatively to the inlet/outlet pipes, than a counter-flow one, so it is, a priori, the type of exchanger considered further. There is no need for a symmetry, i.e. the length travelled by the flow of refrigerant ("D" in figure 3) can be very different from the exterior fluid one ("W" in figure 3), so leaving many possibilities to size the exchanger. The air refrigerant fluid flow will be oriented horizontally, whereas the exterior fluid flow can be oriented either horizontally (preferably) or vertically.
  Note that a cross-flow exchanger is less efficient than a counter-flow exchanger. However, the air-water exchanger will be considered, for calculations, as a counter-flow heat exchanger without reduction of the efficiency. This because the difference of temperature between outlet and inlet of the exterior fluid (water) will be very low compared with the difference of temperature of the refrigerant fluid (dry compressed air). Now for the air-air exchanger, a reduction of efficiency will have to be taken into account (cf. §6.4.1).

- a counter-flow exchanger (as on the figure 4). The inlet/outlet pipes are more difficult to manage compared to a cross-flow exchanger.

Note: this type of exchanger (plate-type) would also applied to the machines proposed in §5.

4.2 Cross-flow plate-type heat exchanger (principle)

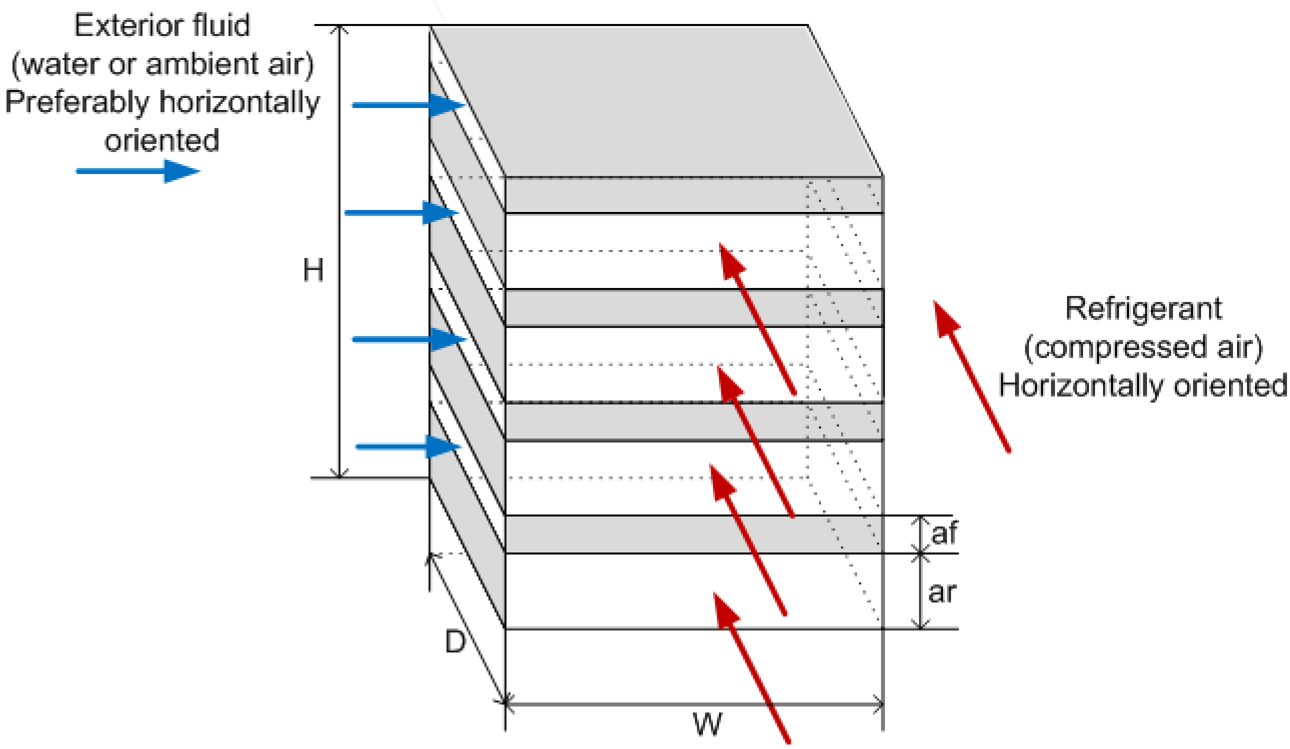

Figure 3

The canal height for the refrigerant (ar) and for the exterior fluid (af) can be of different values, the goal being to improve the global heat transfer, keeping a reasonable pressure drop.

An exchanger comprises N canals (8 in the figure 3): N/2 for the refrigerant and N/2 for the exterior fluid.

The heat exchanger can be (non-exhaustive list) in stainless steel, in cupro-nickel 90/10 or in plastic.
The refrigerant (dry compressed air) pressure is supposed equal to 10 bar (absolute pressure). Of course, the air pressure could be higher than 10 bar so as to reduce the exchanger sizes and the power necessary to make circulate the refrigerant (compressed air).

> Note 1: the refrigerant could also be not compressed at all, at the atmospheric pressure, but this would increase the size of exchangers, pipes and turbine.
> Note 2: further, once the air compressed, it is not supposed air leak. However, in a real project, it would be necessary to assess the probability of leak (not nil) and the probable mass of air lost by unity of time, this to finally determine the expected mean loss of electric power due to air leak.

With 10 bar, it is taken the following hypothesis about the thickness of plates (t_p): t_p is equal to 0.6 mm for stainless steel, 1.2 mm for cupro-nickel and 3 mm for plastic.
Now, let's call:

- Hr the refrigerant intake height with Hr=N/2 x ar
- Hf the exterior fluid intake height with Hf=N/2 x af
- Hrf the total height for the refrigerant and exterior fluid intake: Hrf= Hr + Hf, so H=Hrf+(N+1) x t_p
- Wrf the width for the refrigerant intake and the canal depth for the exterior fluid: Wrf = W - 2 x t_p
- Sr the refrigerant intake surface Sr=Hr x Wrf
- Drf the canal depth for the refrigerant and the width for the exterior fluid intake: Drf = D - 2 x t_p.
- Sf the exterior fluid intake surface Sf=Hf x Drf

It will be supposed cupro-nickel for the calculations, because it is one of the best material: good thermal conductivity, good elastic limit and good resistance against corrosion from sea water.
For these exchangers, due to the weak ambient air specific heat and to the weak ambient air mass density, the global transfer heat per unit area for ambient air will be very weak and almost determined by the sole air convection heat transfer coefficient. Consequently, it can be expected that the air-air heat exchanger volume will be very important, so as to reconcile a weak pressure drop with the necessary heat transfer. This would lead to a plant cost almost summed up by the air-air exchanger cost. So, even if not ideal, exchangers in plastic, a lot less expensive, could be envisaged.

It can also be noted that making circulate 1 kg/s of air through a pipe is much more costly, in term of power than making circulate 1 kg/s of water through the same pipe, for the same pressure drop coefficient. Indeed, the ratio of power Wair / Wwater is equal to $(\rho water / \rho air)^2$.

4.3 Counter-flow plate-type heat exchanger (principle)

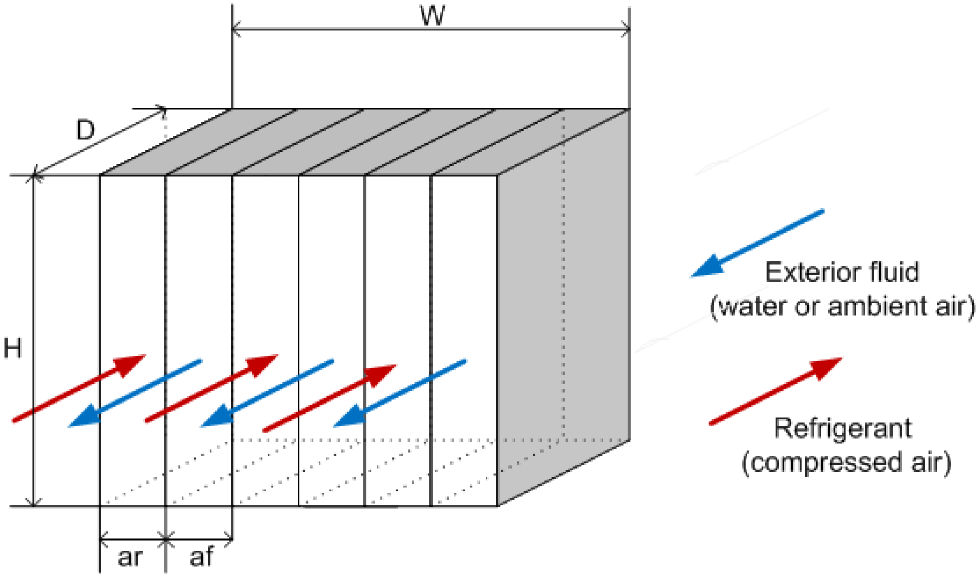

Figure 4

This type of exchanger is not used in this paper, but cross-flow exchangers are calculated as counter-flow exchangers, with a reduction of efficiency for the air-air exchanger.

5. Abandoned possibilities for a cold air / warm water configuration

Two possibilities of cycle were envisaged and then abandoned. The first one is the classical Rankine closed cycle. The other one is a not classical open Brayton cycle.

5.1 Rankine closed cycle

The diagram of a machine based on the well known Rankine closed cycle (used on OTEC plants) is proposed below, in figure 5. The refrigerant is usually ammoniac (R717). As said previously in §4.2, the volume of the ambient air – refrigerant exchanger will be very important due to the very low ambient air convection heat transfer coefficient. So the necessary quantity of refrigerant will be enormous (at best, several tons per electric kW). The potential hazards and the cost of such installation would be extremely high. So such solution is abandoned.

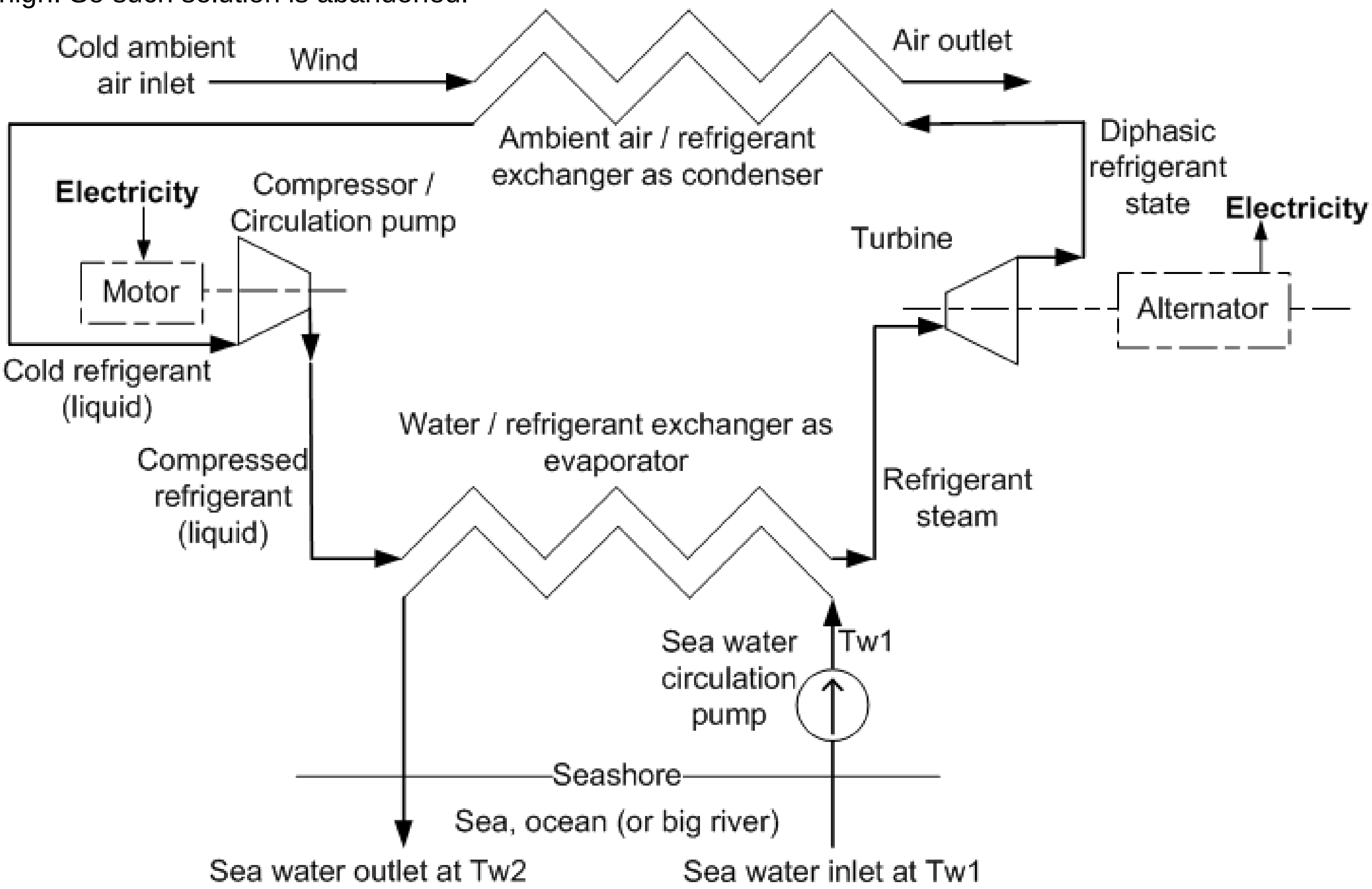

Figure 5

5.2 Brayton open cycle

The Brayton open cycle uses the ambient cold air as refrigerant, the hot source being the sea water. Below, in figure 6, a diagram of the necessary equipment and a description of the theoretical cycle are displayed.

Note: "P" is worth for "Pressure", "v" for "specific volume", "T" for "Temperature" and "s" for "specific entropy".

In an ideal case, with perfect efficiency of the compressor and the turbine, this cycle, would work. Unfortunately, due to the limited efficiency of the motor compressor (i.e. effective energy of air compression / electric energy consumed) about 0.75, the electric power generated by the alternator would be inferior to the electric power consumed by the motor compressor. So such solution is abandoned.

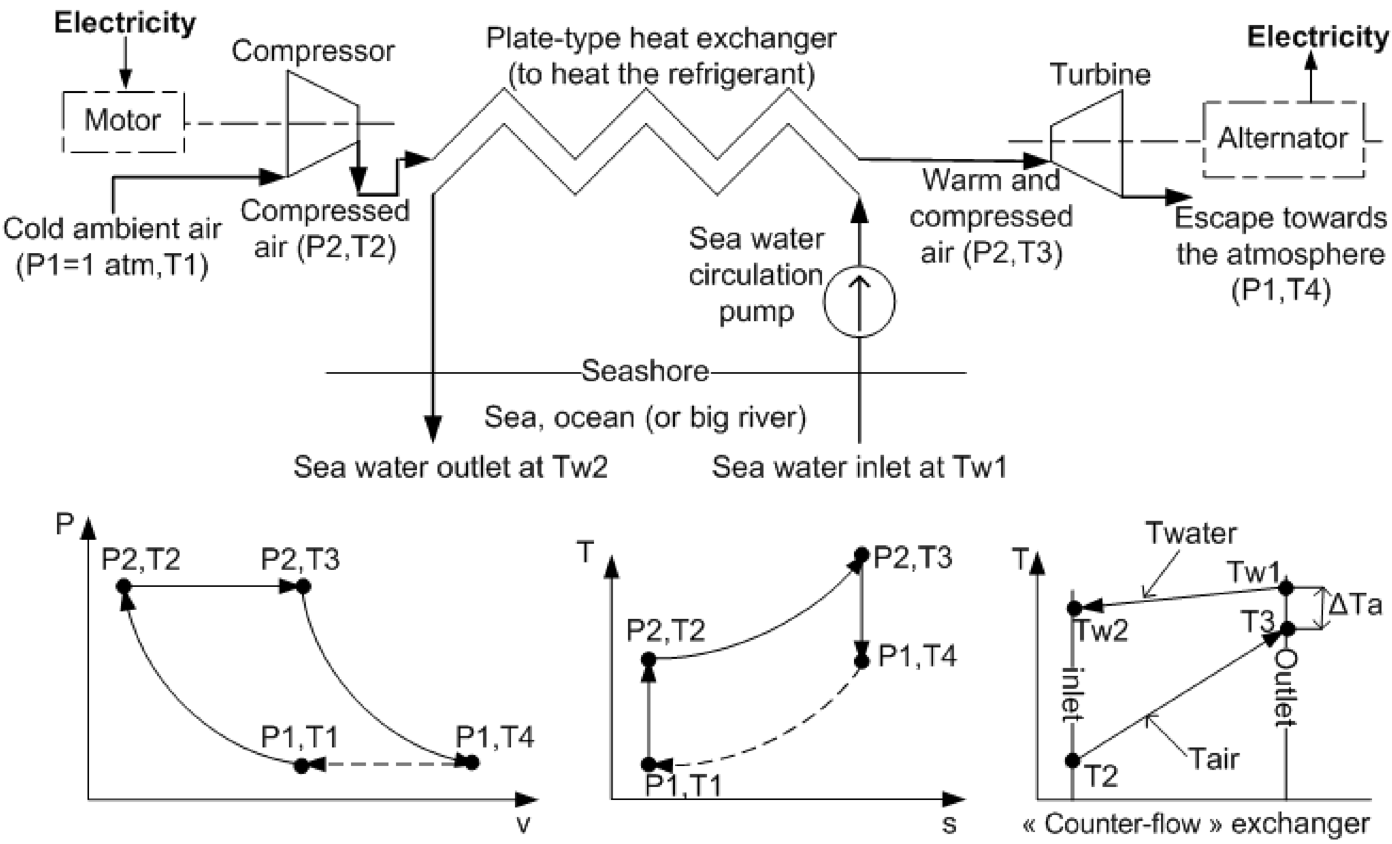

Figure 6

6. Proposal of an aeraulic thermosiphon machine in a cold air / warm water configuration

6.1 Principle of the aeraulic thermosiphon closed cycle

6.1.1 General description

This (not classical) closed cycle uses compressed air as refrigerant, the hot source being the sea water and the cold source being the ambient air.

Further, in figure 7, a diagram of the necessary equipment and a description of the theoretical cycle are displayed.

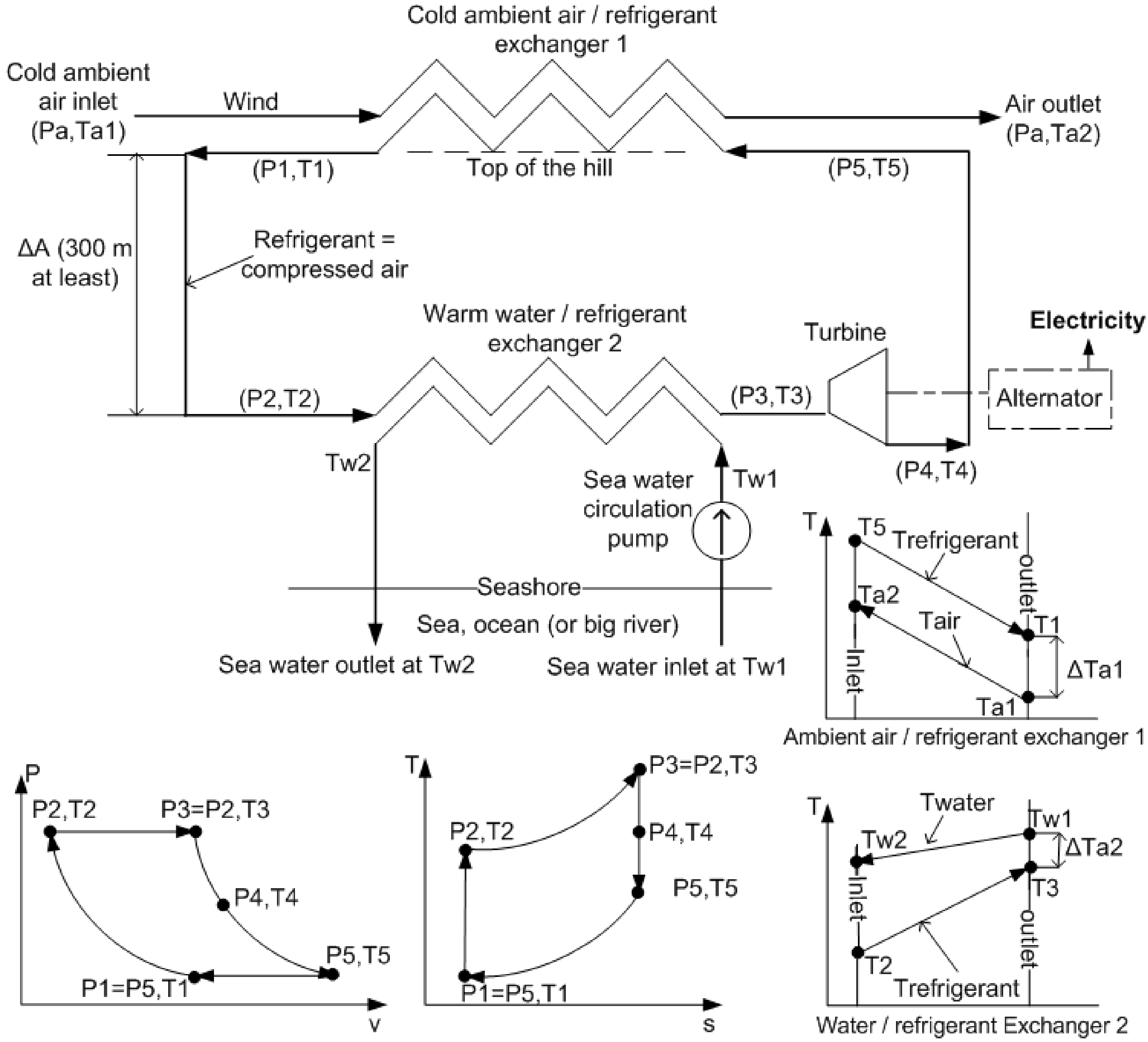

Figure 7

6.1.2 About the sea water and the exchanger 2

The sea water suction will be located near the sea surface (where the water is the warmest), so close to the coast. Let's say between 15 and 30 m depth. The sea water exhaust will be located at the same depth than the suction but far away from the suction so that the pumped water remains at the same temperature, without interference with the water exhaust. In a cold sea (in Greenland, for example) where the water is close to freeze, the difference of temperature Tw1-Tw2 will be sufficiently small to avoid a freezing of the water at the outlet (let's say Tw1-Tw2 ≤ 0.5 °C), taking into account the necessary marine current which evacuates and mixes the exhausted water in the ocean. Moreover, this condition will permit to consider the cross-flow exchanger 2 as a counter-flow exchanger, without reduction factor.

The water speed Vf through the heat exchanger will be supposed equal to 0.1 m/s.
The necessary electric power Wf to make circulate the sea water across the exchanger 2 will be limited to 3% of the generated electric power. As an hypothesis, let's say that the total length Lt of sea water pipe is equal to 1000 m, with a square cross section equal to Sf (cf. §4.2). So the pressure drop ΔPf through the pipe and the exchanger 2 is equal to $\Delta \mathrm{Pf} = \lambda \mathrm{f} \times \left(\frac{\mathrm{Lt}}{\sqrt{\mathrm{Sf}}} + \frac{\mathrm{W}}{\mathrm{Dhf}}\right) \times \rho_{water} \times \frac{Vf^2}{2}$
The friction factor λf can be taken equal to 64/Reynolds for a laminar flow (cf. [9] page 490).

ρ_{water} is equal to 1000 kg/m^3. W is defined in §4.2, figure 3. Dhf, the hydraulic diameter through the exchanger 2 is equal to 2 x af (cf. §4.2).
The electric power Wf consumed by the sea water circulation pump, necessary to compensate this pressure drop is equal to $Wf = Qv_f \times \Delta Pf / \eta_{pump}$
With $Qv_f = Vf \times Sf$ and the efficiency of the pump η_{pump} taken equal to 0.75 (as a first hypothesis).

Possibly, the sea water speed could be reduced (respecting Tw1-Tw2≤0.5°C), because the water convection heat transfer coefficient is much greater than the air one.

> Note 1: it would be possible, as an option, to fix the exchanger 2 directly into the ocean, at a depth between 15 and 30 m, if marine currents are sufficiently fast (let's say ≥ 0.1 m/s) and permanent. This would permit to economize the sea water circulation pump.
>
> Note 2: in any cases, a minimum marine current must exist to evacuate the refreshed water into the ocean.

6.1.3 About the ambient air, the refrigerant air and the exchanger 1

About the ambient air

The air intake will be arranged to take profit of the wind. It will be supposed that a wind at the mean speed Vaa equal to 2 m/s crosses the exchanger 1. The relationship between the initial wind speed Vaa0 (i.e. before the air inlet) and Vaa depends on the pressure drop through the exchanger and the way the air is delivered to the exchanger, because it is slightly accelerated due to the reduction of cross section, so an entrance profile could be used. This complex calculation is outside of the scope of this document. An approximation would be Vaa=Vaa0/2, so Vaa0=4 m/s. So it is implicitly supposed a mean wind speed of 4 m/s (14.4 km/h). Below 4 m/s the global working will be inferior to the nominal and reversely.

> Note: a mean speed Vaa through the exchanger 1 equal to 2 m/s is probably a pessimistic hypothesis for locations as Greenland where the Katabatic wind blows almost permanently, at about 15 m/s.
> A mean speed superior to 2 m/s would increase the air heat transfer and the efficiency of the machine, or it would permit to reduce the exchanger 1 size.

Moreover, to simplify the calculation it will be supposed that:

- the ambient air mass flow rate will be supposed equal to the refrigerant air mass flow rate (see §6.4.1 for details).
- the specific heat at constant pressure of ambient and refrigerant air (Cp_{air}) will be fix and equal to 1006 J/(kg.°K).

The air escape will be separated from the air intake so as to avoid a return to the air intake. It will be directed so that the prevailing wind moves away the air escaping from the plant.

About the refrigerant air

The (absolute) pressure of the compressed air (P3) can be of any reasonable value (let's say from 1 atm=1.013 bar to 100 bar). In the further calculation, it will be supposed a pressure of 10 bar.

The pressure drop through the exchangers 1 and 2 and the pipes will be limited to 10% of the expected difference of pressure across the turbine (see §6.1.6 for details).
It will be supposed 1500 m of refrigerant pipes, for ΔA=559 m.

6.1.4 Option about the use of the "warm" refrigerant air for air heat pumps

Possibly, a small part of the refrigerant air at exchanger 2 outlet (P3/T3) which is much less cold than the ambient air (at Ta) could be extracted to be used in air heat pumps installed on the plant and on the surrounding area, through a closed circuit (not described). This will economize on the necessary power consumed by these heat pumps.

6.1.5 About the theoretical aeraulic thermosiphon cycle
This cycle (supposed perfect in the figure 7) is composed of:

- An adiabatic compression due to the cold refrigerant (compressed air) weight thanks to the difference of altitude between the exchangers 1 and 2, from P1,T1 to P2,T2.
- An isobaric heating in the water/refrigerant exchanger 2, from P2,T2 to P2,T3.
  Note: the temperature approach (ΔTa2) might be, ideally, close to 0° C.
- An adiabatic expansion in the turbine, from P2,T3 to P4,T4.
- An adiabatic expansion due to the warm refrigerant weight, due to the difference of altitude between the exchangers 1 and 2, from P4,T4 to P5,T5.
- And finally an isobaric cooling in the air/refrigerant exchanger 1, from P5,T5 to P5,T1.
  Note: the temperature approach (ΔTa1) might be, ideally, close to 0° C.

The cycle will be calculated below. Notations are the ones used in §1 and [8].

For about both columns of refrigerant (compressed air)
The first principle of thermodynamics can be written: dq = di - vdp (cf. [8] p 46).
For an adiabatic evolution, dq=0. Now, from the Bernoulli equation, dP= - ρ x g x dZ (ρ is the air density).
Moreover, di = Cp_{air} x dT and v=1/ ρ.
So, it comes: Cp_{air} x dT = (1/ ρ) x (-ρ x g x dZ) and dT/dZ = -g/Cp_{air} = - 9.75 E-3 °C / m for dry air.
Note that dT/dZ does not depend of ρ. In other words, it means that an increase of 100 m in altitude makes decrease the dry compressed air temperature by about 1 °C, and reversely.
But dP/dZ = - ρ x g depends on ρ.

Determination of the temperature and the pressure at the turbine outlet (T4)
It is necessary to determine T4 to be able, further, to determine the specific (i.e. for a flow rate of 1 kg/s) power generated by the turbine: $P_{turbine}$= Cp_{air} x (T3 – T4), the evolution being supposed adiabatic (and reversible).

Moreover, for any adiabatic evolution between a point 2 and a point 1, we have $\frac{P2}{P1} = \left(\frac{T2}{T1}\right)^{\frac{k}{k-1}}$ (cf. [8] p 263).
The temperature at point 1 (T1, i.e. the cold source inside the cycle) and the conditions at point 3 (P3,T3, i.e. the warm source inside the cycle) are supposed known. T1 and T3 depend on the exterior cold and warm source : T1 = Ta1 + ΔTa1 ≈ Ta + ΔTa1 and T3 = Tw1 – Δta2 (see figure 7).
P3 is a user choice concerning the pressure of the compressed air (10 bar in the example).
Note that ΔA (difference of altitude between exchangers 1 and 2) is not known at this level.

In any cases, it can be written P3/P4 = (P3/P1) x (P1/P4)
From the figure 7, it can be seen that P1=P5 and P2=P3 (no pressure drop supposed in a perfect cycle).
It comes P3/P4 = (P2/P1) x (P5/P4) so $\frac{P3}{P4} = \left(\frac{T3}{T4}\right)^{\frac{k}{k-1}} = \left(\frac{T2}{T1}\right)^{\frac{k}{k-1}} \times \left(\frac{T5}{T4}\right)^{\frac{k}{k-1}}$
Or T3/T4 = (T2/T1) x (T5/T4) so T4/T3 = (T1/T2) x (T4/T5) and finally T4 = T3 x (T1/T2) x (T4/T5)
From dT/dZ = -g/$Cp_{air,}$ T2 = T1 + g x ΔA / Cp_{air} and T5 = T4 - g x ΔA / Cp_{air}
To simplify the writing, let's call ΔTA = g x ΔA / Cp_{air} so T2 = T1 + ΔTA and T5 = T4 - ΔTA
So T4 = T3 x (T1/ (T1 + ΔTA)) x (T4/(T4 - ΔTA)) or T4 - ΔTA = T3 x (T1/ (T1 + ΔTA))
So T4 = ΔTA + (T3 x T1) / (T1 + ΔTA)
After several developments, this formula can also be written T4 = T3 – ΔTA x ((T3/(T1+ ΔTA))-1)
The pressures P4 and P5 can be deduced from T4, T5, T3 and P3:

$$P4 = P3 \times \left(\frac{T4}{T3}\right)^{\frac{k}{k-1}} \quad \text{and} \quad P5 = P3 \times \left(\frac{T5}{T3}\right)^{\frac{k}{k-1}}$$

Determination of the best efficiency η_{thermo} and comparison with the Carnot efficiency
The aeraulic thermosiphon cycle efficiency η_{thermo} can be calculated as η_{thermo} =(Qw-Qc)/Qw with
Qw = Cp_{air} (T3 – T2), the specific heat (in W/(kg/s)) transferred to the system from the warm source and
Qc = Cp_{air} x (T5 - T1), the specific heat (in W/(kg/s)) transferred from the system to the cold source.
So η_{thermo} =((T3 - T2) – (T5 –T1)) / (T3 - T2) (with η="Eta")
η_{thermo} =((T3 – T1 - ΔTA) – (T4 – ΔTA –T1)) / (T3 - T2)

So η_{thermo} = (T3 – T4) / (T3 - T2), which is logical as Cp_{air} x (T3 – T4) corresponds to the power generated by the turbine ($P_{turbine}$), i.e. η_{thermo} = $P_{turbine}$ / Cp_{air} x (T3 - T2)
Now η_{thermo} can be developed, knowing T4: η_{thermo} = (T3 – T3 + ΔTA x ((T3/(T1+ ΔTA))-1)) / (T3 - T2)
So η_{thermo} = ΔTA x ((T3/T2)-1) / (T3 - T2) = ΔTA x ((T3-T2) / T2) / (T3 - T2) = ΔTA / T2 = (T2 – T1) / T2
Or η_{thermo} = (T2 – T1) x (T3/T2) / T3 = (T3 –T1 x (T3/T2)) / T3

The Carnot efficiency for this cycle between T3 and T1 is equal to η_{Carnot} = (T3 – T1) / T3, which can be compared with η_{thermo} = (T3 –T1 x (T3/T2)) / T3
Because T3 is always superior to T2, η_{thermo} is always inferior to η_{Carnot}. η_{Carnot} is the limit for η_{thermo} when Qc tends to 0 (which has no interest).

What is more interesting is to define the "best efficiency" for which $P_{turbine}$ is the maximum possible.
$P_{turbine}$ = Cp_{air} x (T3 – T4) but also $P_{turbine}$ = η_{thermo} x Cp_{air} x (T3 - T2) with T2 = T1 + ΔTA and so:
$P_{turbine}$ = ΔTA / T2 x Cp_{air} x (T3 - T2) = Cp_{air} x (T3 – T1 - ΔTA) x ΔTA / (T1 + ΔTA)
$P_{turbine}$ = (Cp_{air} x (T3 – T1) - Cp_{air} x $ΔTA^2$) / (T1 + ΔTA)
The maximum $P_{turbine}$ is found for d($P_{turbine}$)/d(ΔTA)=0
After several developments, it comes only one real solution $ΔTA_{best}$ equal to:
$\Delta\mathrm{TAbest} = \sqrt{\mathrm{T3} \times \mathrm{T1}} - \mathrm{T1}$, . It can be approximated as $ΔTA_{best}$ ≈ (T3-T1) / 2
The best $P_{turbine}$ corresponding to this $ΔTA_{best}$ can be approximated as $\mathrm{Pturbine} \approx \frac{\mathrm{Cp} \times (\mathrm{T3}-\mathrm{T1})^2}{2 \times (\mathrm{T3}+\mathrm{T1})}$
The "best efficiency" corresponding to this $ΔTA_{best}$ can be approximated as $\eta\mathrm{thermo} \approx \frac{\mathrm{T3}-\mathrm{T1}}{\mathrm{T3}+\mathrm{T1}}$
Of course, the real efficiency will be inferior to this value.
It can be deduced that the best η_{thermo} ≈ η_{Carnot} /2
It can also be deduced, from ΔTA = g x ΔA / Cp_{air}, that $ΔA_{best}$ = $ΔTA_{best}$ x Cp_{air} / g

For example, suppose that T3=-2°C, P3=1E6 Pa and T1=-12°C, it can be determined:

- $ΔTA_{best}$ = 4.95 °C, T2 = - 7.05°C, T4= - 2.0939 °C, P4=9.988 E5 Pa, T5=-7.05°C, P5=9.364 E5 Pa
- $P_{turbine}$ = 94.5 W/(kg/s), with ΔPTurbine = P3-P4 =1212 Pa
- η_{thermo} = 1.86 % to compare with η_{Carnot} = 3.69%
- $ΔA_{best}$ = 508 m

Note that, in this configuration, the exchanger 1 must be located 508 m above the exchanger 2.

Advantages and disadvantages of this cycle
The theoretical efficiency is less than the Rankine efficiency, which is close to the Carnot efficiency. It is close to the Brayton efficiency (about half of the Carnot efficiency).

The big advantage of this cycle is that it does not need a compressor as with the Brayton cycle, as the compression is done by the difference of densities in the pipes located between the exchangers 1 and 2. These pipes don't need to be vertical, but their long length leads to a stronger pressure drop, which must be limited by using a very large duct diameter (or by increasing the air pressure).

> Note: from a rest configuration (no heating, no cooling, and no air movement, after the air filling for example), heating and cooling in both exchangers will not make circulate the air towards the turbine. The air direction is, a priori, undetermined. To force the air direction towards the turbine, an asymmetry is necessary:
> - either a small specific fan must be started at the beginning and then stopped,
> - or the air contained in the right pipe between the points (P4,T4) and (P5,T5) must be provisionally heated.
>
> This subject is outside the scope of this paper.

6.1.6 About the real aeraulic thermosiphon cycle
Now there are 3 mains sources of irreversibility:

- The turbine-generator group has a global efficiency η_{Tur} (electrical power generated by the alternator / aeraulic power supplied to the turbine) around 0.75.

- The temperature approaches of the exchangers 1 and 2 ($\Delta Ta1$ and $\Delta Ta2$, cf. figure 7) are not nil. Moreover, they can't be too much weak, because the exchangers would be very large. There is a compromise to do so as to have the smallest exchanger possible, for an electric power target. To simplify the calculation, it will be supposed that $\Delta Ta1 = \Delta Ta2 = 2°C$.

- The mechanical power lost due to air pressure drop ΔPr (Pa) through the exchangers and the pipes is transformed in heat. This heat is almost completely lost for the cycle. A reasonable condition is to have a maximum global pressure drop ΔPr_{max} equal to 10% of the expected difference of pressure across the turbine (P3-P4) called $\Delta PTurbine_{best}$. So to keep $\Delta PTurbine_{best}$ (= P3-P4), the expected difference of altitude ΔA_{best} will be increased by a factor 1.1 to obtain $\Delta A_{real} = \Delta A_{best}$ x 1.1.

  To simplify the calculation, the maximum total pressure drop ΔPr_{max} will be split into:
    - the pipes: $\Delta Pr_{p\_max} = 0.5 \times \Delta P_r$ (ΔPr_p calculated in §6.5)
    - the exchanger 2: $\Delta Pr_{2\_max} = 0.1 \times \Delta P_r$ (ΔPr_2 calculated in §6.3)
    - the exchanger 1: $\Delta Pr_{1\_max} = 0.4 \times \Delta P_r$ (ΔPr_1 calculated in §6.4)

Now, it will be considered that:
- 3% of the electric power generated will be used to make circulate the sea water across the heat exchanger at 0.1 m/s (cf. §6.1.2).
- 4% of the electric power generated will be used by the plant itself for different uses (light and heating, control, automatic cleaning of the exchangers, etc).

The real cycle efficiency η_{Real} calculated between T1 and T3 (so taking into account the temperature approach), is equal to $\eta_{Real} = \eta_{thermo} \times \psi$, with ψ (psi) equal to: $\psi = \eta_{Tur} \times (1-0.03) \times (1-0.04) = 0.70$.
For example, for η_{thermo} = 1.85 %, η_{Real}=1.29 % (about 1/3 of η_{Carnot}).

Now the theoretical mechanical power delivered by the turbine is equal to:
$W_{turbine} = Qm_r \times P_{turbine} = Qm_r \times Cp_{air} \times (T3 - T4)$ with Qm_r the refrigerant air mass flow rate.
The heating power (Wh) is equal to $Wh = Qm_r \times Cp_{air} \times (T3 - T2)$
So the net real generated electrical power (We) is equal to $We = \eta_{real} \times Wh$ (= $\psi \times W_{turbine}$),
Consequently $Wh = We / \eta_{real}$

6.1.7 About the turbine and the control

The turbine will have to work with very low difference of pressure (between about 100 to 10000 Pa depending on the compressed air pressure). It will be similar to the turbines developed for OTEC open cycle plants, i.e. with a very large diameter. For more information, see the §4 of [13].

The turbine alternator group is considered as technically classical (even if it is very complex), contrary to the exchangers.

As atmospheric and ocean conditions (temperatures and wind speed) will slowly change all the time, a control of the air circulation pump and its bypass, the wind turbine and its bypass, the sea circulation pump and the turbine-generator group will have to take place.

6.1.8 About the plate-type heat exchangers

There are two main types of plate-type heat exchangers:
- With smooth plates.
- With corrugated plates, i.e. the plates have a relief in form of chevrons. This relief improves the heat transfer but also increases the pressure drop through the exchanger.

Further, it will only be calculated exchangers using smooth plates, because exchangers with chevrons could only be calculated by the manufacturers of these exchangers.

6.1.9 Comparison with a wind turbine
It is interesting to compare this machine with a wind turbine, by unity of mass of air (1 kg) exploited.

The specific energy Ewt generated by 1 kg of ambient air crossing a wind turbine is equal to
$Ewt=V^2/2 \times \eta wt \times Cf$.
V is the nominal speed of the wind (in general about 15 m/s) for the maximum power. ηwt is the efficiency of the wind turbine, i.e. 0.47 for a modern one. Cf the capacity factor (mean real electric power generation / maximum electric power generation) is supposed favorable and equal to 40 %. So Ewt = 21.2 J/kg.

For about the specific energy Eat supplied by kg of compressed air (or by kg of ambient air, due to a previous hypothesis in §6.1.3) for the machine studied here (aeraulic thermosiphon), we have:
$Eat = Cp_{air} \times (T3 - T4) \times \psi \times Cf$
The capacity factor Cf is supposed equal to 0.75 (deduced from §3). See §6.1.6 for the other parameters.
So Eat=1006 x 0.0939 x 0.7 x 0.75= 49.6 J/kg

Eat ≈ 2.3 x Ewt, so the final extraction of energy by kg of air is only 2.3 times better for this big machine compared to a modern wind turbine. Note that this estimation is a bit pessimistic because the mean wind speed supposed for this machine is taken equal to 4 m/s (cf. §6.1.3) whereas the implicit mean wind speed supposed for the wind turbine is much higher (i.e. about 9 m/s).

Now, the cost of an aeraulic thermosiphon would be very high, mainly because it would be necessary to install very big ducts to transport the compressed air (1500 m in the example of §6.1.3) along a hill. The civil engineering work would be enormous. It is evident that this machine would be much more expensive than a wind turbine, for the same electric power generated. If this would prevent a commercial use, this will not prevent to study it.

> Note about the use of compressed helium gas instead of compressed air as refrigerant
> Helium has a heat capacity Cp equal to 5193 J/kg instead of 1006 J/kg for the air, i.e. about 5.2 times better. So the flowrate of helium would only need to be 1/5.2 (19%) of the air flowrate. Consequently, the exchangers and the ducts would be smaller and the equipment less expensive. But it is necessary to take into account the price of helium to fill the equipment and then to maintain the pressure. Finally, it would not be a cheaper solution.

6.2 Generalities about the calculation of the air-water exchanger 2

Hypothesis (a priori fixed):
- The exchanger is in cupro-nickel 90/10 (cf. §4.2).
- The thickness of plates (t_p) is equal to 1.2 mm (cf. §4.2).
- The cupro-nickel thermal conductivity K_{cn} is supposed equal to 40 W/(m.°K)
- The mean roughness ε of smooth plates is taken equal to 0.01 mm (10 microns). Note that a lower value of ε would permit to decrease the pressure drop.

- ar and af (cf. figure 3) are supposed respectively equal to 1.4 cm and 0.6 cm. These values could be refined.
- The equivalent canal "hydraulic" diameters (Dh) for compressed air and for water are respectively equal to $Dh_r = 2 \times ar$ and $Dh_f = 2 \times af$.
- To simplify, according to the figure 3, it is supposed that Wrf=Hrf (cf. §4.2).

- It will be supposed that the net power generated We is equal to 10 kW (10000 W).
  Note that there is no obvious technical superior limitation about the power to generate. 10 kW corresponds to a test machine. The other conditions supposed are:
  Ta1=-14°C, T1=-12°C, Tw1=0°C, T3=-2°C, so 10 °C between T3 and T1 and 14 °C between the warm and the cold sources, P3= 10 bar (1E6 Pa):

- The water speed Vf is equal to 0.1 m/s all along the exchanger 2 (cf. §6.1.2).

- The water thermal conductivity K_{water} is supposed equal to 0.6 W/(m.°K)
- The specific heat at constant pressure of water Cp_{water} is equal to 4190 J/(kg.°K).
- The water density ρ_{water} is equal to 1000 kg/m^3.

- The air thermal conductivity K_{air} is supposed equal to 0.024 W/(m.°K)
- The specific heat at constant pressure of air Cp_{air} is equal to 1006 J/(kg.°K).
- The air density ρ_{air} at 0°C and 10 bar is equal to 12.75 kg/m^3.

Parameters (expected to be modified):

- N (cf. §4.2) is the number of canals (even number) which will determine Hrf and Wrf. For example, for N=500 then Hrf =5 m and Wrf=Hrf=5 m .
- It must be done an hypothesis about the canal depth Drf (cf. §4.2), for example Drf=1 m.

Preliminary calculations

Several calculations will be done before entering in the calculation of the exchanger:

- Hr, Hf, Hrf, Wrf, Sr, Sf will be calculated according to §4.2.
- From T1, T3 and P3, it will be calculated $\Delta TA_{best,}$ T2, T4, T5, P4, P5, η_{thermo}, ΔA_{best} according to §6.1.5.
- $\Delta PTurbine_{best,}$ ΔA_{real}, ΔPr_{max}, ΔPrp_{max}, $\Delta Pr2\_{max}$, $\Delta Pr1\_{max}$, η_{Real} and Wh will be calculated according to §6.1.6.
- The air mass flow rate (kg/s) is equal to Qm_r = Wh / (Cp_{air} x (T3-T2)).
- From the mean air temperature Tmean=(T3+T2)/2 and P3, the air mass density ρ_{air} can be deduced.
- The air volume flow rate (m^3/s) is equal to $Qv_r = Qm_r / \rho_{air}$ and the air speed Vr = Qv_r / Sr (cf. §4.2)
- The dynamic air viscosity μr (Mu) can be deduced from Tmean.
- The Reynolds number of air (Rer) can be deduced from ρair, Vr, Dhr and μr.
- The Prandl number of air (Prr) can be deduced from μr, Cpair, Kair.
- The dynamic water viscosity μf can be deduced from Tw1.
  The Reynolds number of water (Ref) can be deduced from ρwater, Vf, Dhf and μf.
- The Prandl number of water (Prf) can be deduced from μf, Cpwater, Kwater.

6.3 Calculation of the air-water exchanger 2

6.3.1 Air pressure drop for smooth plates, for a given canal depth (Drf)

The friction factor for air λ_r (Lambda) is calculated in a recursive way, from the Colebrook and White formula: $\frac{1}{\sqrt{\lambda r}} = -2 \times \log_{10}\left(\frac{\varepsilon}{3.71 \times \mathrm{Dhr}} + \frac{2.51}{\mathrm{Rer} \times \sqrt{\lambda r}}\right)$. It will be done the hypothesis that the flow is always turbulent (i.e. not laminar) and inside the “Smooth pipes” zone in the Moody diagram (see reference [9] page 491), so to avoid any discontinuity in the calculation. Flow singularities are neglected.

The pressure loss ΔPr2 is equal to $\Delta\mathrm{Pr2} = \lambda\mathrm{r} \times \frac{\mathrm{Drf}}{\mathrm{Dhr}} \times \rho_{air} \times \frac{Vr^2}{2}$

Once Drf calculated, it will be checked that $\Delta Pr2 \leq \Delta Pr2\_{max}$ (cf. §6.1.6).

Note: the real required value of Drf will be obtained after calculation of the heat transfer. So several calculations will be necessary to converge towards stable values of Drf and ΔPr2.

6.3.2 Heat transfer for smooth plates

Classically, an exchanger will be defined by 3 equations:

1. Wh = Qm_r x Cp_{air} x (T3-T2) (air side) (see §6.1.6)

2. Wh = Qm_f x Cp_{water} x (Tw1-Tw2) (water side)
Qm_f (kg/s) = Vf x Sf x ρ_{water} (see §6.2 and §4.2)
Tw2 unknown can be deduced from the equations 1 and 2.
Note: Tw2 is close to Tw1 (by less than 0.5 °C).

3. Wh = U x Sh x ΔTlog

With U the overall heat transfer coefficient (W/(m^2 x °K), defined as:
1 / U = 1 / hair + fr + t_p/K_{cn} + ff +1 / hwater

- hair is the convection heat transfer of the refrigerant (air).
  hair = K_{air} x Nu_r / Dh_r where the Nusselt Nu_r is calculated with the formula for air issued from the reference [10] page 9.

$$\mathrm{Nur} = 0.0214 \times [\mathrm{Rer}^{0.8} - 100] \times Prr^{0.4}$$

- hwater is the convection heat transfer of the exterior fluid (sea water).
- hwater = K_{water} x Nu_f / Dh_f where the Nusselt Nu_f is calculated with the Colburn formula (cf. reference [10] page 9).

$$\mathrm{Nuf} = 0.023 \times \mathrm{Ref}^{0.8} \times Prf^{\frac{1}{3}}$$

- fr: the fouling factor on the air side is estimated to 3.5E-4 (m^2.°K)/W (cf. reference [11] page 31).

- ff: the fouling factor on the water side is estimated to 1E-4 (m^2.°K)/W (cf. reference [11] page 31).

- For tp and Kcn see §6.2.

For a counter-flow exchanger $\Delta\mathrm{Tlog} = \frac{\Delta T2 - \Delta T1}{\ln\frac{\Delta T2}{\Delta T1}}$ with ΔT1 = Tw1 - T3 and ΔT2 = Tw2 – T2
Sh (m^2) is the required surface between the two fluids (i.e. air and water) to transfer Wh.
Sh = Wh / (U x ΔTlog)
Now from Sh = $Drf_{required}$ x Wrf x (N-1), it can be deduced $Drf_{required}$ = Sh / (Wrf x (N-1))

Ideally $Drf_{required}$ must be equal to Drf (taken as an hypothesis). Several calculations will be necessary until $Drf_{required}$ converges towards Drf.

6.3.3 Results

- ΔA_{best} = 508 m and ΔA_{real} = 559 m
- $\Delta PTurbine_{best}$ = 1212 Pa, so $\Delta Pr_{\_max}$ = 121 Pa
- Qm_f = 151 kg/s and Vr = 0.79 m/s, with Sr=14.8 m^2
- Qm_f = 387 kg/s (and Vf = 0.1 m/s), with Sf=3.97 m^2
- N=460
- Drf=2.80 m
- Hrf = Wrf = 4.6 m
- ΔTlog=3.84 °C

The necessary plate surface is equal to 5920 m^2, which is important for only 10000 W of electric power. This is due to the small hair: 36.8 W/(m^2 x °K) to compare with hwater which is equal to 492 W/(m^2 x °K).
Note that:

- Tw1-Tw2=0.47 °C, which is, as required (cf. §6.1.2), ≤ 0.5 °C.
- ΔPr2 = 11.2 Pa , which is, as required (cf. §6.1.6) ≤ ΔPr_{2_max} = 12.1 Pa.
- Wf = 215 W which is, as required (cf. §6.1.2 and §6.1.6) ≤ 3% We (=300 W), the water square pipe section being equal to 3.87 m^2.

Note that at 559 m the ambient air is colder than at the surface, by about 3.6 °C. This is favorable for this machine, because the difference of temperature between the ocean and the ambient air is greater than expected and, consequently, the electricity generation would be higher.

6.4 Calculation of the air-air exchanger 1

6.4.1 Differences with the exchanger 2

It is an ambient air/compressed air exchanger. The ambient pressure (Pa) is supposed equal to 9.48E4 Pa at 559 m.

This exchanger will be calculated in the same way that the one described in §6.2 and §6.3:

- With the sign "r" for the refrigerant (compressed air) and "f" for the ambient air.
- With "water" which corresponds now to "ambient_air".

About the exchanger 1, the differences with the exchanger 2 are listed below:
- ar and af (cf. figure 3) are supposed respectively equal to 0.4 cm and 1.6 cm (instead 1.4 and 0.6 cm). These values could be refined.
- Wrf=Hrf / 1.36 instead Wrf=Hrf
- Vaa=2 m/s (cf. §6.1.3)
- (T3-T2) is replaced by (T5-T1) and (Tw1-Tw2) by (Ta2-Ta1)
- 1 / U = 1 / hair + fr + t_p/K_{cn} + ff +1 / hambient_air, with ff=fr
- To limit the size of the exchanger 1 and to simplify, it will be supposed that Qmf = Qmr.
  So T5-T1=Ta2-Ta1, ΔTlog=ΔTa1=T1-Ta1=2°C by hypothesis (cf. §6.1.6) and Ta2=Ta1+(T5-T1)
  However, because the exchanger is a cross-flow type, there is a F factor to take into account, this one reducing the efficiency of the exchanger (see reference [12]). It can be estimated that F=0.675 (from [12]). So Wh = U x Sh x F x ΔTlog = 1.35 x U x Sh.
  Consequently: Sh = Wh / (1.35 x U)
- For the ambient air, the characteristics (ρair etc) will be calculated at Pa and (Ta1+Ta2)/2.

6.4.2 Results
- Qm_f = 151 kg/s, Vr = 0.477 m/s, with Sr=25.6 m^2
- Qm_f = 151 kg/s (and Vf = 2 m/s), with Sf=59.4 m^2
- N=1320
- Drf=5.62 m
- Hrf = 13.2 m and Wrf=9.7 m

The necessary plate surface Sh is equal to 71970 m^2, which is very important. This is due to the small hambient_air: 11.0 W/(m^2 x °K) to compare with hair (for the refrigerant) which is equal to 26.5 W/(m^2 x °K). Note that ΔPr1 = 44.9 Pa, which is, as required (cf. §6.1.6) ≤ ΔPr_{1_max} = 48.5 Pa.

6.5 Calculation of the pressure drop through the compressed air pipes
With ΔA_{real} = 559 m, it will supposed a length Lp=1500 m of pipes. The pressure drop through these pipes ΔPr_p must be inferior or equal to ΔPr_{p_max} (cf. §6.1.6). These pipes are supposed having a circular section of diameter Dp.
All the air characteristics (ρair, μp …) will be calculated at the conditions $Pmean_p$= (P3+P1)/2 and $Tmean_p$= (T3+T1)/2.

As explained in §6.1.3, the friction factor for air λ_p is calculated in a recursive way, from the Colebrook and White formula: $\frac{1}{\sqrt{\lambda p}} = -2 \times \log_{10}\left(\frac{\varepsilon}{3.71\times \mathrm{Dp}} + \frac{2.51}{\mathrm{Rep}\times\sqrt{\lambda p}}\right)$. It will be done the hypothesis that the flow is always turbulent to avoid any discontinuity in the calculation.
The Reynolds number Rep can be deduced from ρair, Vp (air speed in the pipe), Dp and the dynamic water viscosity μp. The mean roughness ε is taken equal to 0.01 mm.

The pressure loss is equal to $\Delta \mathrm{Prp} = \lambda \mathrm{p} \times \frac{\mathrm{Lp}}{\mathrm{Dp}} \times \rho_{air} \times \frac{Vp^2}{2}$
Qm_f being known, $Qv_p = Qm_f / \rho_{air}$ and $Vp = Qv_p$ / Sp with Sp the pipe section equal to π x Dp^2 / 4
So an hypothesis on Dp will give Sp, Vp, Rep, λp and finally ΔPr_p.
From the previous example developed from §6.2 to §6.4, it can be shown than $\Delta Pr_p = \Delta Pr_{p\_max}$=60.6 Pa for Dp=3.23 m. In this case, Vp= 1.46 m/s.

7 <u>Possible improvements of the exchangers and the machine efficiency</u>

7.1 About the way to reduce the exchanger sizes
A problem of this machine is the exchanger size, and particularly the air-air exchanger for which the exchange surface is equal to about 72000 m^2. This is equivalent to 7.2 m^2 for 1 W in electric power, which is a lot. As explained above (§6.4.2), the problem is related to the weak overall heat transfer coefficient

(U=7.75 W/(m^2 x °K)), due to weak convection heat transfer coefficients: hambient_air (11.0 W/(m^2 x °K)) and hair (26.5 W/(m^2 x °K)).
To increase the "hair" coefficient of the refrigerant, one solution is to increase the compressed air pressure. This would lead for the same pressure drop "credit" (i.e. ΔPr_{1_max} in §6.1.6) to increase the speed Vr through the exchanger and consequently the Reynolds (Rer) and hair. Another solution for higher Vr and hair is to increase the pressure drop "credit" by increasing ΔA_{real} (cf. §6.1.6), which is equal to 559 m in the example. A last possible solution would be to add chevrons on the compressed air side to increase convection, taking into account the correlative increase in the pressure drop.

To increase the "hambient_air" coefficient of the ambient air, it will be necessary to know more precisely the mean speed Vaa through the air-air exchanger (§6.1.3). Indeed, it is supposed Vaa =2 m/s, but this hypothesis is probably pessimistic (see §6.1.3). Another possible solution is to add chevrons on the ambient air side to increase convection, taking into account the correlative increase in the pressure drop.

The parameters (ar, af, Hrf, Wrf…) taken for both exchangers 1 and 2 could certainly be improved.
Moreover, 2°C for the temperature approaches (ΔTa1 and ΔTa2) is not certainly the best choice and could be improved.
Note that an air-air exchanger 1 in plastic would reduce drastically the price of such exchanger.

The calculation of this machine as proposed in this paper is relatively simple, the goal here being to test the concept. A more realistic calculation of the exchangers would need a simulation based on a 1D or better a 2D finite-element method.

7.2 About the efficiency of the machine

In the example, it is taken 14 °C between the ambient air temperature (-14°C) and the see ocean temperature (0°C), considered at the surface. Because the air-air exchanger 1 is at 559 m in the example, the ambient air is 3.6 °C colder than at the surface (based on -0.65 °C per 100 m). So the real difference would be 14+3.6 °C=17.6 °C in fact, giving a better efficiency.
Note that it would be possible to search for even colder ambient air by raising both the exchangers 1 and 2.

8 Conclusion

It has been shown (in §3) that it exists in different zones of the world, relatively strong differences of temperatures between the ambient air and the sea, or ocean surface water, sufficient to extract electric energy, as shown in §2, this natural source being extremely important in term of potential energy.

To take profit of the difference of temperatures between the ambient air and the surface sea water, it has been first studied and abandoned (in §5) the possibilities of a Rankine cycle and a Brayton cycle.
Then in §6, it has been proposed an aeraulic thermosiphon system, installed between the seashore and a high relief (300 m at least).
By a simple calculation (§6), it has been shown that this concept could work in the cold air / warm water configuration. However, this machine, by symmetry, could also work in the warm air / cold water configuration.

One drawback is that the size of the exchangers (see §4, §6.3.3 and §6.4.2) and especially the air-air one which would be important.
It has been proposed solutions to reduce the exchangers size and to improve the machine efficiency (cf. §7).

Now, the cost of such machine is expected to be very high and, consequently, an aeraulic thermosiphon is not competitive compared to a wind turbine (cf. §6.1.9 and note below), for about generating electricity, which prevents any commercial use.

> Note: about wind farms to install in the future, in Greenland, these ones associated with a possible future international electricity network, look at the interesting Karataba project (cf. [14]).

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