Guidance page for practical work 3: solar pond

1) Objectives of the practical work

The project objective is to study the operation of power plants using the inverse temperature gradient existing in solar ponds (Figure 3.1) and show how to model them realistically with Thermoptim.

Inverse gradient solar ponds are collectors of a particular type. They are reservoirs of salt water (NaCl or $MgCl_2$) with an area of several thousand m2, whose bottom is covered with an absorbent surface. To avoid the internal convection and heat loss that result, the pond is divided vertically into three zones:

- a thin surface region at low salt concentration (1 g/cm3), at almost homogeneous temperature, stirred by wind and convection;
- a zone of 1 m–1.5 m, increasing salt concentration with depth, the salt gradient preventing the onset of convection;
- a 2 to 4 m high lower zone of homogeneous and very high salt concentration (>1.3 g/cm3, and sometimes saturated) which stores solar heat.

Such a configuration allows a surface temperature close to that of the atmosphere (25 °C), while the bottom of the pond reaches 90 °C or more. Heat losses are reduced.

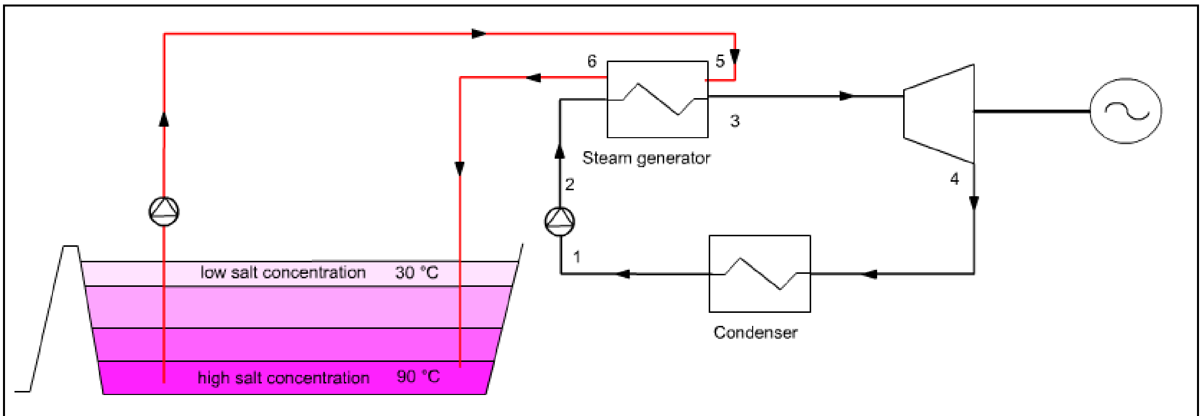

Figure 3.1: Solar pond

Inverse gradient solar pond applications are numerous: space heating, absorption refrigeration, power generation through a thermodynamic closed vapor cycle.

In the latter case, hot water is pumped and used to evaporate a working fluid, which follows a motor cycle and is condensed by exchange with ambient air, for example in an air-coil.

Although technically valid, the concept of an inverse gradient solar pond is not economically viable, which explains why prototypes built in Israel (a solar pond of 20 ha producing 5 MW worked in Israel until 1989) and USA (the University of Texas at El Paso pond has an area of 3000 m2 of 70 kWe capacity) are no longer exploited.

This relatively simple practical work is designed for students who have studied steam plants. If this is not the case, they should do it firstly.

2) References

Le site http://www.solarpond.utep.edu of the El Paso solar pond (Figure 3.1) in the United States has a lot of information directly relevant to this practical work.

Many references on solar ponds are also available in journals such as Solar Energy, in particular under the name H. Tabor, one of the world's leading experts of the subject. However, most of these papers deal with the physics of ponds, and not with associated thermodynamic cycles.

3) Main practical work

The thermodynamic cycle is aHirn or Rankine cycle, whose modeling in Thermoptim poses no particular problem.

The objective of the work is to model such a cycle and to calculate its efficiency, then to build its exergy balance. Among the available thermodynamic fluids availble in Thermoptim (ammonia, butane, propane and R134a), the one which leads to the best performance should be sought.

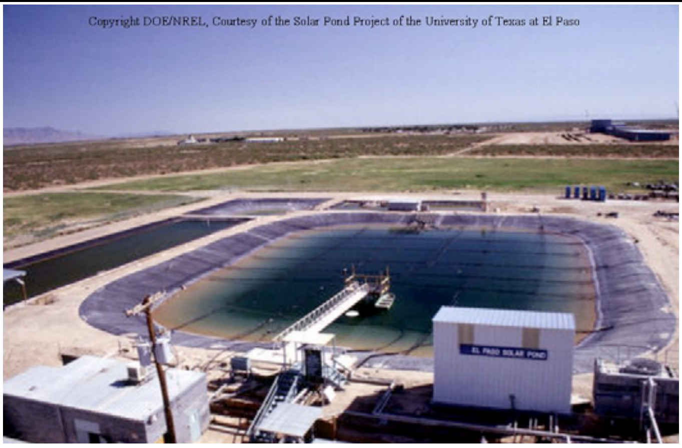

Figure 3.1: El Paso solar pond under construction

The sizing of heat exchangers is obviously critical given the low temperature difference between hot and cold sources. The values of pinches should be as low as possible while remaining realistic. Obviously, when attempting to compare the performance of cycles using different fluids, the values of pinches should be about the same.

It will be seen that although the energy performance of solar pond is very low (<3%), the exergy efficiency may be correct.

The flow of hot water is 12 kg/s, its temperature is 90 °C, and the temperature of the cold water is equal to 25 °C.

All other values must be determined, justifying the assumptions made. Estimations of system size magnitudes (exchange surfaces, free flow areas...) should be done, not forgetting to take into account the pumping power, a priori not negligible.

3.1 Resolution approach

Modeling in Thermoptim of a solar pond leads to a diagram like the one in Figure 3.2. The efficiency is very low (8.4%) due to the small temperature difference between the two sources.

For students who have modeled a simple steam cycle (if it is not the case, they must begin by working on the sessions S25En and S26En), the only difficulty is the construction and configuration of the three part heat exchanger.

How to solve this problem is explained in detail at the end of session S18En on the thermodynamics of heat exchangers in an exercise where are set exchangers representing a steam cycle boiler.

The setting of the cycle is less classic. We assume initially that the selected working fluid is ammonia.

The pump can be considered isentropic, but not the turbine, whose isentropic efficiency can by example be taken equal to 0.9.

To design heat exchangers, it is necessary to set the values of minimum temperature differences between the fluids, that is to say, pinches. The lower they are, the better the thermodynamic cycle, but also the larger will be the exchange surfaces.

For temperature differences between the cycle and hot and cold sources, we choose pinches as low as possible: 10 °C for superheating (gas-liquid exchange), 11°C at the economizer outlet (liquid-liquid exchange), 8 °C at the condenser outlet (liquid/two-phase mixture exchange).

The need to have a sufficient temperature difference between the hot source and the working fluid makes us select a superheating temperature of 80 °C and a vaporization temperature of 72.5 °C, i.e. a high pressure of 35 bar.

For similar reasons, the condensation temperature of ammonia is 36 °C, which corresponds to a pressure of 14 bar. The hot water flow-rate is given but not that of the working fluid. We must start with an estimate value, which is then modified so that the pinch at the economizer outlet is equal to 11°C as chosen. Once this value obtained, the flow of cooling water can be directly determined by exchanger “condenser” if we set its outlet temperature.

3.2 Auxiliary consumption

We have so far not quantified the consumption of auxiliary circulating pumps including hot and cold brine. Try to estimate them.

3.3 Sensitivity studies

Based on these results, it is possible to conduct several sensitivity studies, first by changing the working fluid, as suggested in the setting out, and possibly also by changing the cycle to reduce irreversibilities.

One can wonder in particular what is the point of superheating, which leads to increase the temperature difference between the evaporator and the heat source (the irreversibilities in the evaporator and condenser are each 35% of total).

Representation in thermodynamic charts

Once the cycle points determined, it is easy to plot them in a diagram such as the thermodynamic entropy chart using the features offered by Thermoptim (figure 3.3).

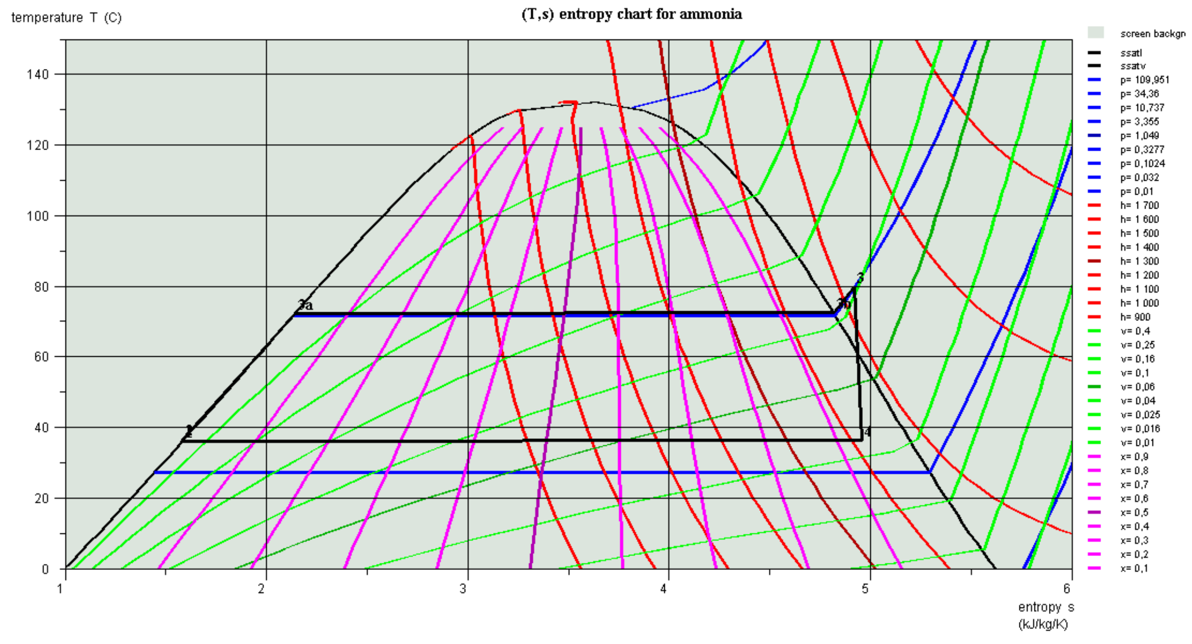

Figure 3.3: Cycle entropy chart

4) Variant: cycle exergy balance

We only propose here one variant, but many alternatives can be imagined, depending on the time available, the level of students, their number, and educational objectives pursued.

It is now possible to ask students to build up the cycle exergy balance, if they have enough time. The great interest of this work is to show that cycles with very low efficiency (due to the low thermal gradient available) may still have a pretty good exergy efficiency.

Diapason session S06En will provide necessary explanations on how to proceed, the S28En session being devoted to the simple steam exergy balance.

The exergy balance of the ammonia cycle can be established without any particular difficulty, considering that the surroundings temperature is 25 °C, and the hot source temperature is 90 ° C.

What is most remarkable is that the exergy efficiency is very high (47%), while the efficiency is very low: this means that, although the difference in temperatures of sources is low, the plant performance is very good.