

# Guidance page for practical work 6: Cooling towers

## 1) Objectives of the practical work

The project objective is to study the operation of cooling towers, to show how they can be realistically modeled with Thermoptim, and to use such a model to demonstrate their interest by comparing the energy performance of two industrial refrigeration cycles, one using an air condenser, and the other a direct contact cooling tower.

The practical work allows students to build models of direct and indirect contact cooling towers. They are thus led to reflect on the properties of moist gases.

This relatively complicated practical work is designed for students who have already studied refrigeration machines and heat exchangers. If this is not the case, there should first do it.

## 2) References

ASHRAE Handbook 2001, "Cooling Towers", chapter 36

A. BRICARD, L. TADRIST, Echangeurs de chaleur à contact direct, BE 9 565, Techniques de l'Ingénieur

F. KREITH, "The CRC Handbook of Thermal Engineering", CRC Press Springer, 3 "Cooling Towers" by A.F. Mills, Chapter 4.9, 1999

D.R. BAKER, H.A. SHRYOCK. 1961. A comprehensive approach to the analysis of cooling tower performance. *ASME Transactions, Journal of Heat Transfer* (August):339.

J.E. BRAUN, S.A. KLEIN, J.W. MITCHELL, "Effectiveness Models for Cooling Towers and Cooling Coils", *ASHRAE Transactions*, Vol. 92, Part 2, p. 164-174, 1989

## 3) Main practical work

### 3.1 Question

Consider a R134a refrigeration machine producing 200 kW of cooling at  $-12^{\circ}\text{C}$ , the condenser being cooled with air at  $25^{\circ}\text{C}$ .

The objective is to compare the performance of the machine depending on whether one uses an air exchanger with a pinch of 16K or a cooling tower, the minimum pinch between water and the refrigerant being below 12 K.

Students have to model the machine with Thermoptim, using external components "DirectCoolingTower" and "DirectCoolingTowerInlet" representing the cooling tower.

Initially, students will reflect on the operation of a cooling tower and its modeling. According to the educational objectives pursued, they can either go into the details of the model implemented as an external class, or just use the external class provided

### 3.2 Cooling tower models

A cooling tower is a heat exchanger of a particular type that discharges heat in the surrounding air in the form of both sensible heat and latent heat due to the increase of its moisture. By working this way, it is possible to cool a fluid at a temperature a few degrees above the ambient air wet bulb temperature (and possibly below its dry bulb temperature), at the cost of a water consumption of about 5% of that which water cooling would require. Both economically and environmentally, cooling towers are very interesting systems, in particular in hot and dry climates.

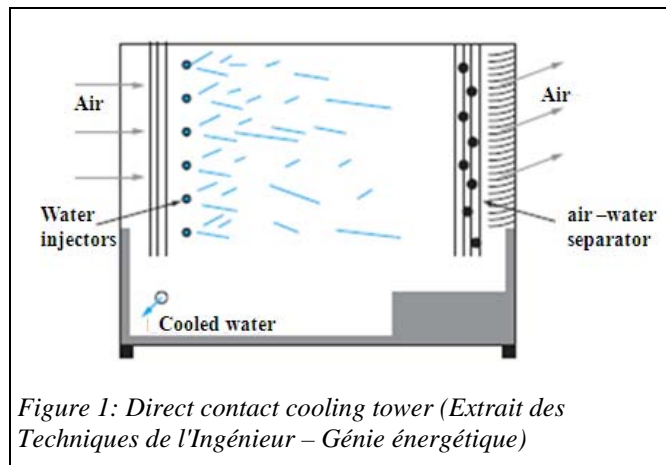


Figure 1: Direct contact cooling tower (Extrait des *Techniques de l'Ingénieur – Génie énergétique*)

There are two main types of cooling towers, called direct or open-cycle, and indirect contact or closed-cycle.

In a direct contact tower (Figure 1), hot water is cooled in contact with ambient air either by spraying fine droplets, or by runoff along flow surfaces. Both fluids being in contact, heat is exchanged by convection, and part of the water vaporizes, thereby increasing the humidity. If it is not saturated, it starts to cool in an almost adiabatic process, before warming up along the saturation curve. Therefore water may come out at a temperature lower than ambient air.

An indirect contact tower involves two circuits known as external and internal. In the latter, the cooling fluid, which can be arbitrary, remains confined in a tube bundle around which the external cooling circuit water runs. It warms on contact, then is cooled by exchange with ambient air by the same mechanism as in a direct contact tower.

Due to the evaporation of a portion of the water, the concentration of dissolved salts and impurities, including biological, is expected to rise. This leads to conduct blowdown similar to those presented for boilers in the textbook.

### 3.3 Theory

The theoretical modeling of cooling towers is quite complex given the multiplicity of transfers that take place. The 2000 ASHRAE Handbook (Chapter 36) presents the theory proposed in 1961 by Baker and Shyrock, which considers (Figure 2) three media exchanging mass and energy:

- the flow of water that cools;
- the air flow which is heated and whose moisture increases;
- an interstitial film.

The assumptions are:

- air leaving the tower is nearly saturated;
- the interstitial film is saturated with moisture air, at the water temperature;
- the Lewis number is equal to 1;

- thermal resistance on the liquid side is negligible compared to that on the air side (for indirect contact towers, it is assumed that the air-liquid film temperature gradient is equal to that of air-water cooling).

The water balance of a small tower element of surface  $\Delta A$ , writes with our usual notations:

$$\Delta Q_{\text{water}} = \dot{m} C_{p\text{water}} \Delta T_{\text{water}}$$

This energy is used firstly to heat air ( $\Delta Q_s$ ), and secondly to evaporate water absorbed by air ( $\Delta Q_L$ ).

$$\Delta Q_s = k_s \Delta A (t'' - t_a)$$

$k_s$  being the heat exchange coefficient in the presence of mass transfer,  $t''$  being the film temperature (assumed equal to that of water) and  $t_a$  that of air.

$$\Delta Q_L = k' \Delta A L_{\text{water}} (w''_{\text{sat}} - w_{\text{asat}})$$

$k'$  being the conductance referred to mass transfer,  $w''_{\text{sat}}$  the film saturation moisture and  $w_{\text{asat}}$  that of air.

We have thus:

$$\Delta Q_{\text{water}} = \dot{m} C_{p\text{water}} \Delta T_{\text{water}}$$

$$\Delta Q_{\text{water}} = k_s \Delta A (t'' - t_a) + k' \Delta A L_{\text{water}} (w''_{\text{sat}} - w_{\text{asat}})$$

In turbulent regime,  $Sc$  and  $Pr$  being the Schmidt and Prandtl numbers and  $Le$  the Lewis number, it can be shown that  $k$  and  $k'$  are related by the following equation:

$$\frac{k_s}{k' C_{p\text{air}}} = \left[ \frac{Sc}{Pr} \right]^{2/3} = Le^{2/3} \quad (5.8.1)$$

It turns out that for water-air mixture,  $Le \approx 1$  as long as  $t_{\text{water}} < 50$  °C, as shown by Merkel in 1921 when he proposed to consider that for cooling towers  $Le = 1$ .

With this assumption, the above equation is simplified, showing the specific enthalpy  $q'$ :

$$\Delta Q_{\text{water}} = \dot{m} C_{p\text{water}} \Delta T_{\text{water}} = k' \Delta A [C_{p\text{air}} (t'' - t_a) + L_{\text{water}} (w''_{\text{sat}} - w_{\text{asat}})]$$

$$\Delta Q_{\text{water}} = k' \Delta A [q'_{\text{sat}}(t'') - q'_{\text{sat}}(t_a)] \quad (5.8.2)$$

This equation can be modified and allows us to write, per unit of water mass flow:

$$\frac{k' \Delta A}{\dot{m}_{\text{water}}} = \frac{C_{p\text{water}} \Delta T_{\text{water}}}{q'_{\text{sat}}(t'') - q'_{\text{sat}}(t_a)} \quad (5.8.3)$$

This equation can be numerically integrated along the cooling tower, leading to results in Figure 3, which shows the evolution of temperatures in the tower as a function of reduced distance between its input and output.

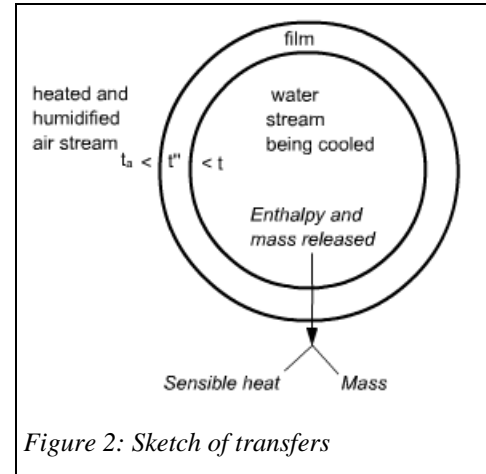


Figure 2: Sketch of transfers

This gives a performance indicator of the cooling tower, which appears in good agreement with experimental data.

Let us now recall a characteristic of heat exchangers, the number of transfer units NTU introduced Chapter 5 of Part 2.

Equation 5.2.1 of Part 2 gives the value of NTU:

$$NTU = \frac{UA}{(\dot{m} \cdot c_p)_{\min}} = \frac{\Delta T_{\max}}{\Delta T_{\text{ml}}} \frac{UA}{(\dot{m} \cdot c_p)_{\min}} \frac{\Delta T_{\text{ml}}}{\Delta T_{\max}}$$

As the heat fluxes exchanged are equal,  $UA \Delta T_{\text{ml}} = (\dot{m} \cdot c_p)_{\min} \Delta T_{\max}$ , and:

$$NTU = \frac{\Delta T_{\max}}{\Delta T_{\text{ml}}}$$

NTU can be physically interpreted as the ratio of the maximum variation in temperature in one of the fluids to the log-mean temperature difference between them. This is indeed an indicator of the quality of the device: the more efficient the heat exchanger, the more it allows for heating or cooling a fluid with a low temperature difference with the other.

The concept of NTU can be generalized by considering not the ratio of the temperature variations, but that of the enthalpy differences. One may in particular put in the denominator the average logarithmic difference of specific enthalpy of saturated air taken at the temperatures of the two fluids, and in the numerator the variation of enthalpy  $\Delta h$  of one of the two fluids, the two being equal.

$$NTU = \frac{\Delta h}{\Delta h_{\text{ml}}} \quad \text{or} \quad NTU = \frac{\Delta h}{\Delta q'_{\text{sat ml}}}$$

In these conditions, equation (5.8.3) of the tower elementary heat balance that we integrated is like an elementary NTU and integration provides the NTU of the tower. To differentiate it from normal NTU in exchangers, we will note it with index m (Merkel).

$$\Delta NTU_m = \frac{k' \Delta A}{\dot{m}_{\text{water}}} = \frac{C_{p\text{water}} \Delta T_{\text{water}}}{q'_{\text{sat}}(t'') - q'_{\text{sat}}(t_a)} \quad \quad NTU_m = \int \frac{C_{p\text{water}} \Delta T_{\text{water}}}{(q'_{\text{sat}}(t'') - q'_{\text{sat}}(t_a))}$$

It is indeed an indicator of how difficult it is to realize the heat transfer in the cooling tower. Incorporating complex coupled phenomena of energy and mass exchange and dependent on technology implementation, it is not possible to give a simple expression.

We call the "approach" the pinch of the tower, i.e. the difference between the outlet temperature of the water and the wet bulb temperature of ambient air (4 K in the example in Figure 3) and the "range" the value of  $\Delta T_{\text{water}}$  (8 K in this example).

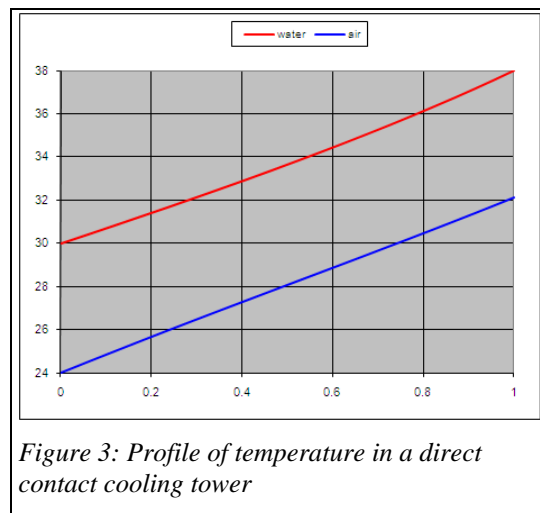


Figure 3: Profile of temperature in a direct contact cooling tower

$NTU_m$  is usually given as a function of four variables:  $\frac{\dot{m}_{water}}{\dot{m}_{air}}$  and  $T_{ai}$ , the wet bulb air inlet temperature, as well

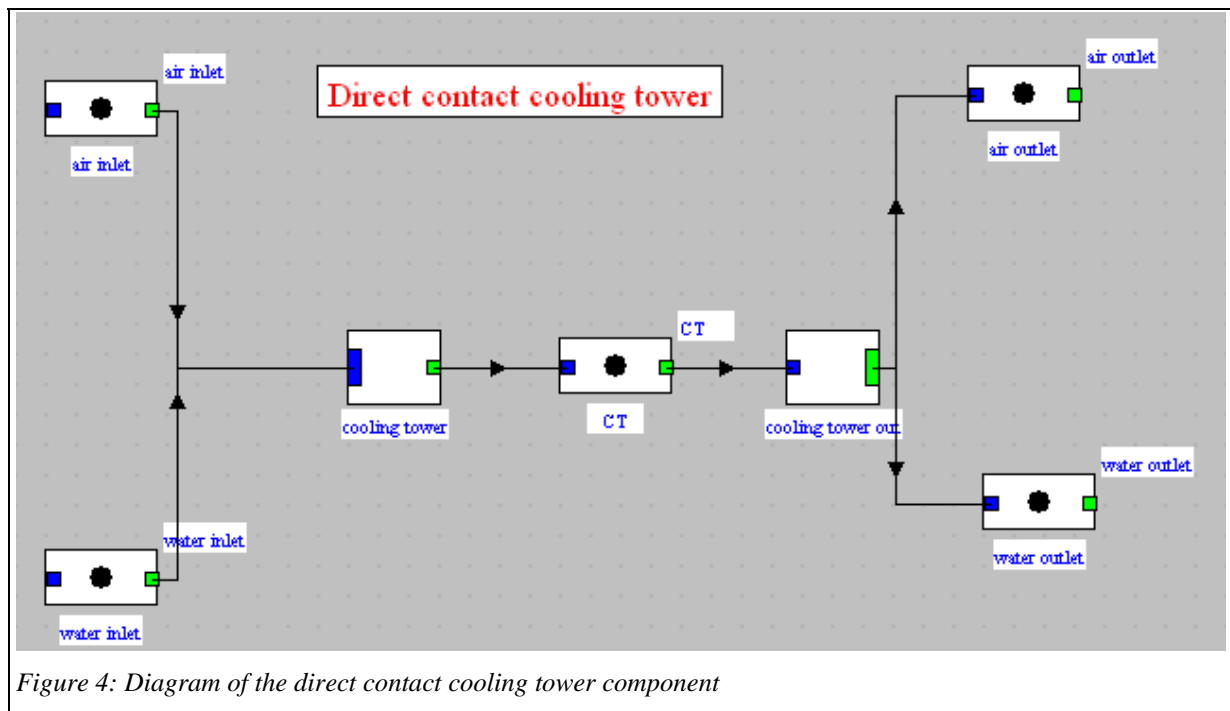
as either the two water temperatures  $T_{wi}$  (inlet) and  $T_{wo}$  (outlet), or the values of range ( $T_{wi} - T_{wo}$ ) and approach ( $T_{wo} - T_{ai}$ ).

If they operate in the same environment with the same water inlet temperature and fluid mass flow of same ratio, two different cooling towers will cool more or less water depending on the quality of their design: the better resulting in the lowest value of  $T_{wo}$ , and thus having a greater  $NTU_m$ .

### 3.4 Modeling of a direct contact cooling tower in ThermoOptim

A direct contact cooling tower has the distinction of being crossed by two separate streams: air and water that exchange matter and energy through an interface. It behaves like a quadrupole receiving two input fluid, and of out which come two others (Figure 4).

For the model to be consistent, it synchronizes the calculations made by the two nodes. More specifically, the outlet divider takes control of the mixer, whose role is to perform an update of the coupling variables associated with the inlet streams.



Based on the model presented above, a reasoning can be made by knowing the total inlet and outlet enthalpy conditions air side entrance and those on the water side, and the model results are consistent with the experimental values and those provided by the manufacturers.

Flow rates of both inlet streams are set by conditions upstream of the component and not recalculated. If water flow is insufficient for its cooling (up to moist bulb temperature of incoming air) to provide air with the enthalpy required, a message warns the user.

Recall that, as usual in calculations of moist functions, values are referred to dry gas, whose composition is invariant, while other calculations in ThermoOptim are relative to the actual gas composition, i.e. are referred to the moist gas. It is therefore necessary to make the corresponding conversions.

The functions for calculating properties of ThermoOptim moist gases and points are available from external classes. We advise you to refer to the note: "Calculations of moist gas from external classes<sup>1</sup>" for a detailed presentation of the available methods

Another point to mention is that when you save a point properties, moisture is not taken into account because it is derived from the gas composition. If we desire to save a change in the humidity of the air inlet, we must also change the gas composition from the moist screen calculations of the inlet point.

The model that we can choose, when we know the exit air temperature, is the following:

- 1) we begin by calculating inlet moist air properties, and determining the dry gas mass flow from that of moist gas; inlet relative humidity  $\varepsilon$  is displayed on the screen;
- 2) the outlet relative humidity is set equal to 1, and the outlet moist air properties are calculated, which gives the specific and total enthalpy to be brought to air;
- 3) the flow of water carried by air is determined and the outlet moist air composition is changed;
- 4) the water enthalpy balance provides its outlet temperature, which must exceed the moist bulb temperature of air entering;
- 5) values downstream the node are updated;
- 6) NTU is calculated by integrating the Merkel equation (NTU integrated) established above and on the basis of the mean logarithmic enthalpies; effectiveness can be deduced from the classical relationship for counter-flow heat exchangers.

The tower is as we said represented by an external mixer connected to an external divider, calculations being made by the latter. Classes are called DirectCoolingTowerInlet and DirectCoolingTower.

node:  type:

main process:    iso-pressure

m global:  h global:  T global:

process name	m abs	m rel	T (°C)	H
air outlet	10.1996	1.002	27.87	3.09
water outlet	13.3008	0.3058	26.03	109.21

**direct cooling tower**

NTU value:   Twater out known  NTU known

inlet rel. humidity: 0.700 epsilon: 0.575

$\Delta Q'$ : 283.662 UA: 47.419

water involved: 0.09958 R: 0.734

approach (°C): 5.071 LMTD: 4.081

range (°C): 5.066 NTU (integrated): 1.197

Figure 5: Cooling tower component screen

<sup>1</sup> <http://www.thermooptim.org/sections/base-methodologique/extensions-thermooptim/calculs-gaz-humides>

The cooling tower component screen is given in Figure 5. We wish here to cool 1kg/s of water from 30 °C to 25 °C. With a rate of 0.9 kg/s of air at 18 °C and relative humidity equal to 0.5, the exhaust air temperature is 20.1 °C and 7.5 g/s water are evaporated. The tower capacity is 20.9 kW.

On this screen we see two values of NTU. That which is in the editable field is calculated by the approximate formula based on the logarithmic mean differences of enthalpies of saturated air between the tower inlet and outlet, while the one displayed at the bottom is the result of the integration of the Merkel differential equation for a counter-flow tower. It is in principle more accurate, but we can see that their values are close.

### 3.5 Solving approach

Let us recall that the objective is to compare the performance of the R134a refrigeration machine ensuring the production of 200 kW of cooling at -12 °C whose condenser is cooled by air at 25 °C, depending on whether one uses an air exchanger with a pinch of 16K or a cooling tower, the minimum pinch between water and the refrigerant being below 12 K.

#### 3.5.1 Part I: installation with cooling tower

For students who have already modeled a simple refrigeration cycle (if not, they must begin by studying the S30En and S31En<sup>2</sup> sessions), the only difficulty is selecting components "DirectCoolingTowerInlet" and "DirectCoolingTower". Refer for this to the note « Using external classes » or operate as shown in the first nine steps of the Diapason session S07En\_ext<sup>3</sup>. The class labels are "cooling tower inlet" and "direct cooling tower".

A global model for the installation is given in Figure 6. In the lower part, the refrigeration cycle is entirely conventional. In the top is the coupling with the cooling tower, which is via two heat exchangers that form the condenser (desuperheater and condenser).

If students are not familiar with heat exchangers, it is recommended that they study sessions S18En and S19En<sup>4</sup> on the thermodynamics and technology of these devices.

The tower settings can in turn (Figure 6) be done as follows: we set air inlet T and  $\varepsilon$  (here 25 °C and 70% RH) and inlet water T (here 31.1 °C) and flow (13.4 kg/s). The compressor of isentropic efficiency is equal to 0.8. Balancing the condenser and desuperheater gives the power to extract  $\Delta H$  and the water output temperature (26.03 °C). We choose option "Twater out known" and the calculation of the tower provides  $T_{air} = 27.89$  °C.

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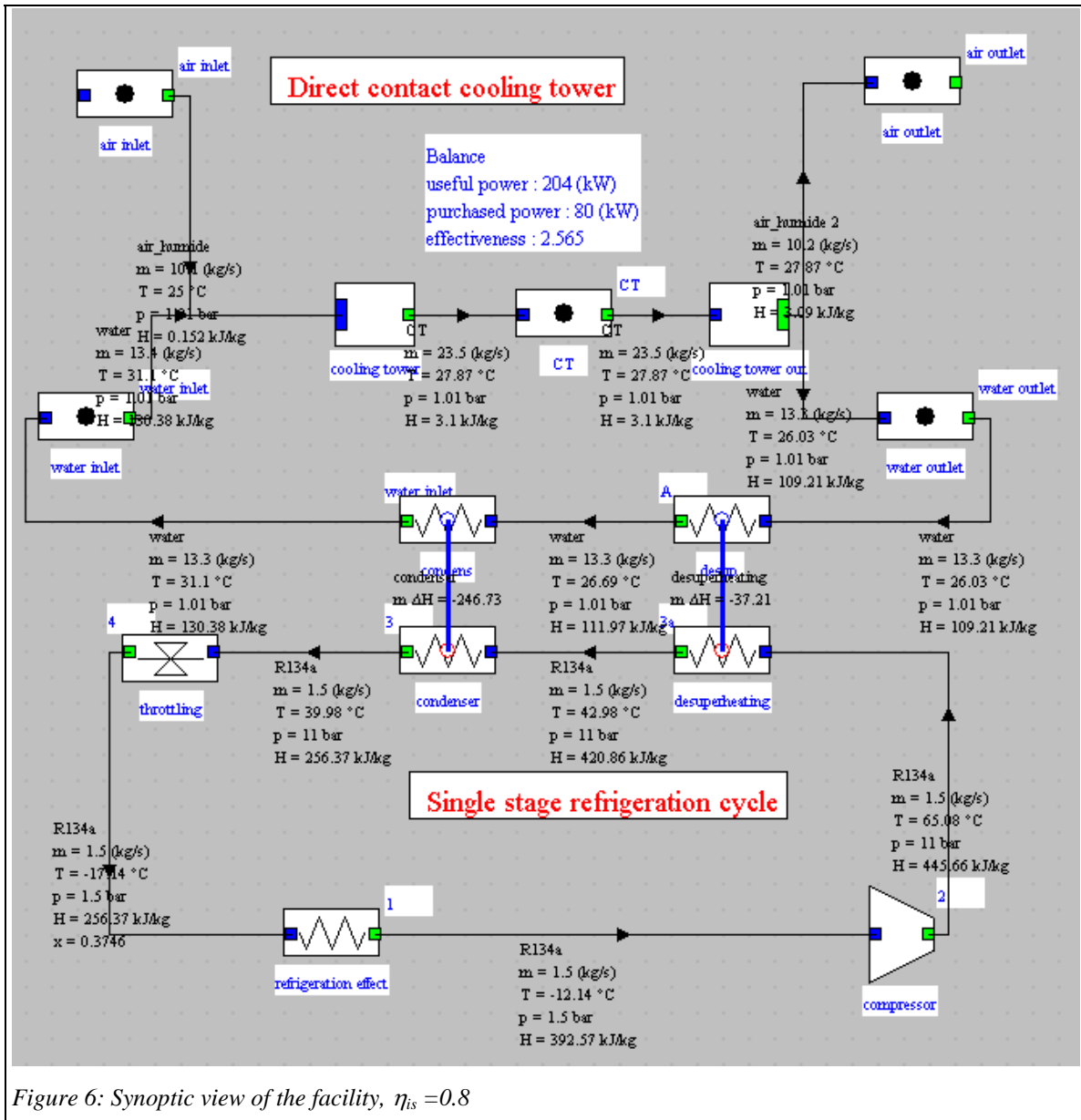
<sup>2</sup> Séance S30 <http://www.thermoptim.org/SE/seances/S30/seance.html>

Séance S31En <http://www.thermoptim.org/sections/enseignement/cours-en-ligne/seances-diapason/session-s31en-water>

<sup>3</sup> Séance S07\_ext : [http://www.thermoptim.org/SE/seances/S07\\_ext/seance.html](http://www.thermoptim.org/SE/seances/S07_ext/seance.html)

<sup>4</sup> Séance S18En <http://www.thermoptim.org/sections/enseignement/cours-en-ligne/seances-diapason/session-s18en>

Séance S19 <http://www.thermoptim.org/SE/seances/S19/seance.html>



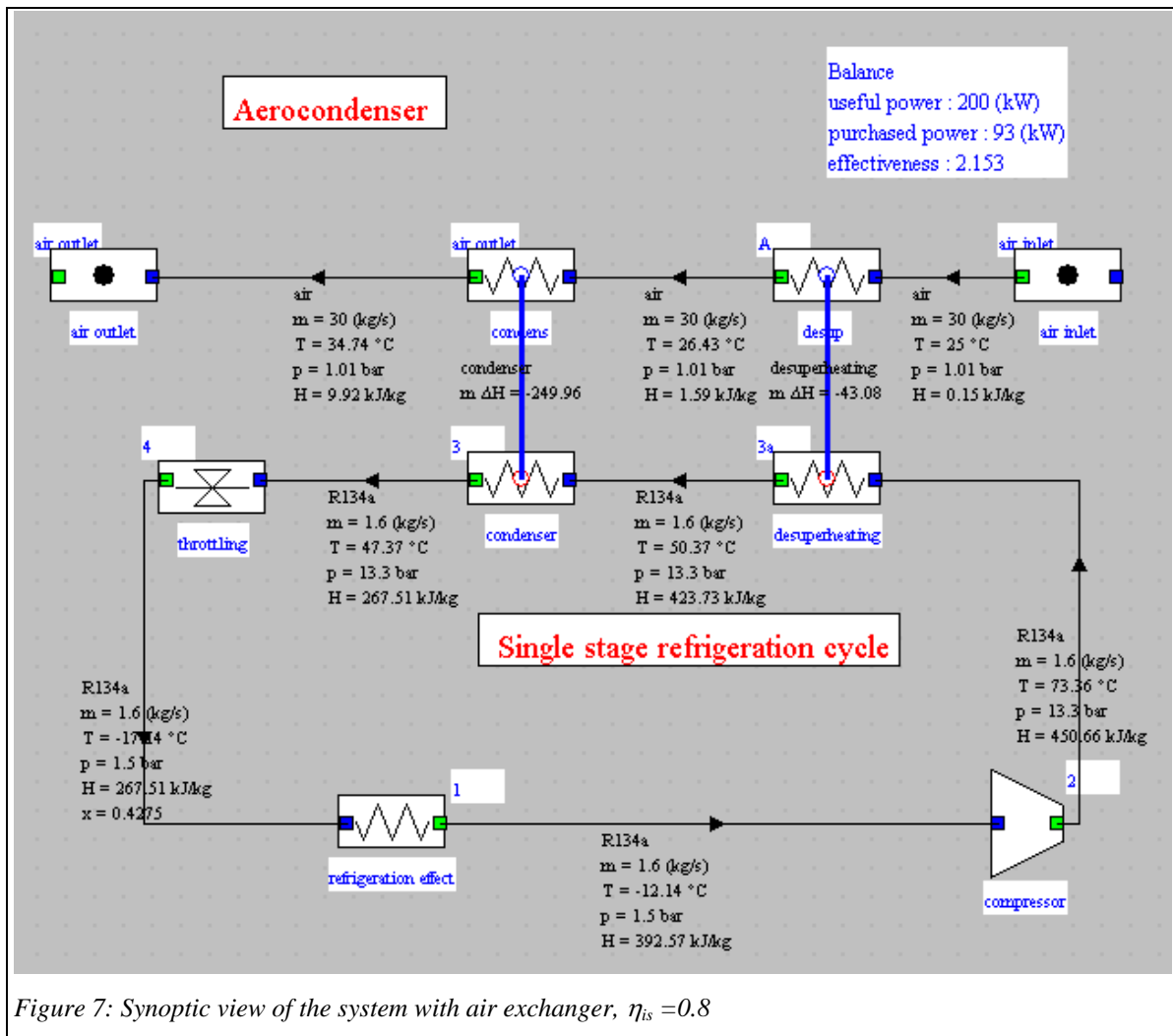
### 3.5.2 Part II: installation with air condenser

Without cooling tower, the air heats up much more. Retaining a maximum flow-rate of 30 kg / s and a pinch of 16 K, the simulation leads to an air temperature at the exchanger outlet equal to 34.7 ° C and thus a condensation temperature of the R134a equal to 50, 4 ° C, i.e. a condensing pressure of 13.3 bar, instead of 11 in the previous case (Figure 7).

COP without the tower is equal to 2.15 instead of 2.57, representing a decrease of 16%. And yet we have neglected the fan power, air flow passing from 10.1 kg/s to 30 kg/s, or the lower isentropic efficiency, compression ratio passing from 7.33 to 8.87.

In addition, the refrigerant flow must be increased from 1.5 to 1.6 kg/s, due to the increased quality as the end of expansion, which implies a larger displacement, volumetric efficiency dropping too.



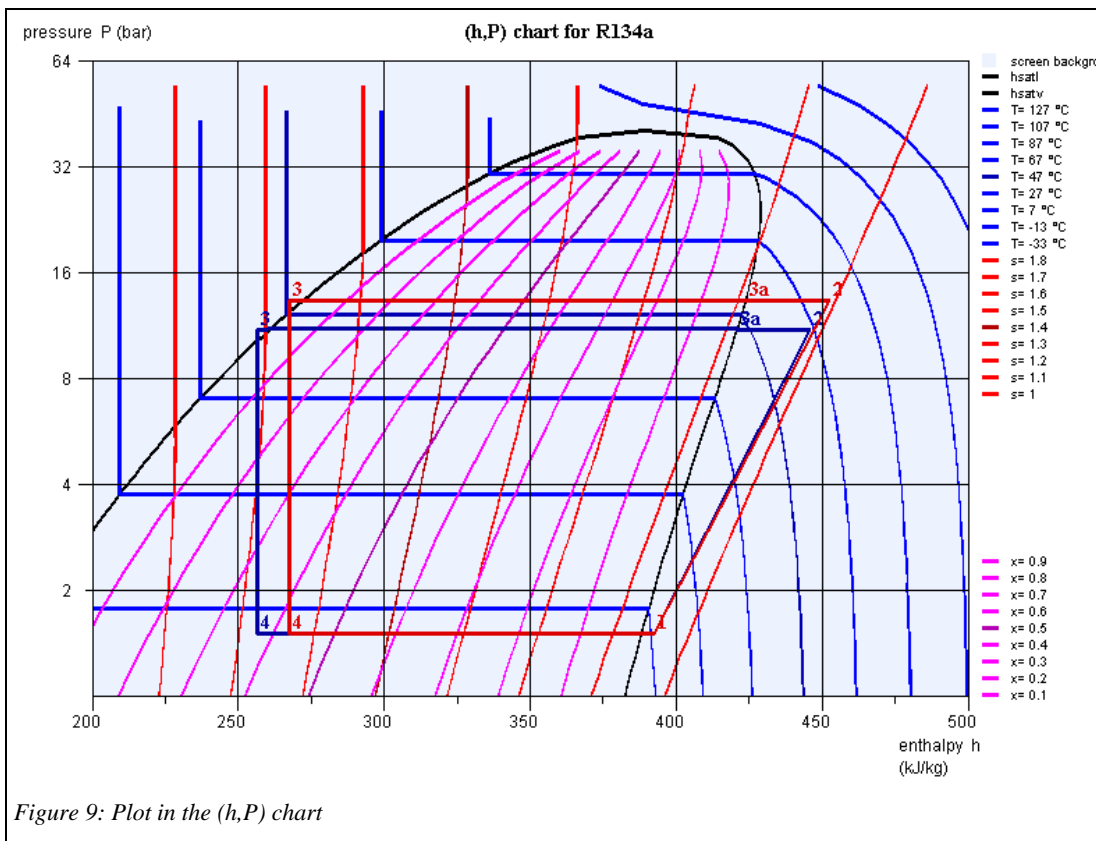


### 3.6 Plot in the thermodynamic charts

Once the points of both cycles determined, it is possible to superimpose them in a thermodynamic chart such as the entropy chart or the (h, P) chart using the features offered by Thermoptim.

Figure 8 shows the result, the air-cooled cycle being drawn in black and the other in blue. A comparison with a Carnot cycle can then be made.

Figure 9 shows two plots in the (h, P) chart. You can directly read on the x-axis the increase in mechanical work consumed by the compressor and the reduction of the cooling effect capacity, which both explain the decrease in COP.



### 3.7 Technological design

A possible activity if students have sufficient time is to ask them to calculate orders of magnitude of system sizing (sections of passage ...), to estimate the pressure drops in heat exchangers, and refine the model accordingly.

They can also calculate the displacement of the compressor, which is pretty easy, and search, which is more difficult, the design of equipment from a manufacturer's range for both the compressor for the cooling tower.

4) variants

#### 4) Variants

We propose a few variants, but many others can be imagined, depending on the time available, the level of the students, their number, and educational objectives pursued.

#### 4.1 Taking into account the variation of isentropic efficiency

Let us consider the compressor isentropic efficiency law:

$$\eta_{is} = 0.893 - 0.004 (\tau - 5)^2 - 0.5 / (\tau - 0.3)$$

Its value goes from 0.8 to 0.775, bringing the COP decline to 18.7%.

#### 4.2 State of the art of cooling towers technology

If the goal is for students to deepen their knowledge of cooling towers, you can ask them to make a brief state of the art of technology used in these facilities and the main constraints that exist at this level, including health issues (legionellosis).

#### 4.3 Cycle exergy balance

Finally, it is possible to ask students to build up the cycle exergy balance, if they have enough time.

Diapason session S06En<sup>5</sup> will provide necessary explanations on how to proceed.

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<sup>5</sup> Séance S06En : <http://www.thermooptim.org/sections/enseignement/cours-en-ligne/seances-diapason/s06en-exergy-balances>

## 5) Work files, recommendations

### 5.1 Work files

The following work files are attached in the archive CoolingTower.zip:

- programming note of external class "DirectCoolingTower"
- extUser.zip file containing external classes
- Project and diagram files of the ThermoOptim model of the refrigeration machine with and without cooling tower
- plot files of cycles in the thermodynamic charts (frigoR134aAir.txt and frigoR134aTour.txt)

### 5.2 Recommendations

It is necessary that the teacher checks that students have at their disposal in their work environment ThermoOptim external classes they need.

The easiest way to this is to run ThermoOptim, then open the external class viewer (menu Special of the simulator). Classes will be grouped by type, class and DirectCoolingTower DirectCoolingTowerInlet should appear respectively among the dividers and mixers, as shown in Figure 10.

If a class is missing, replace the extUser.zip file by that provided in the archive CoolingTower.zip. If present, it is unnecessary to make this change.

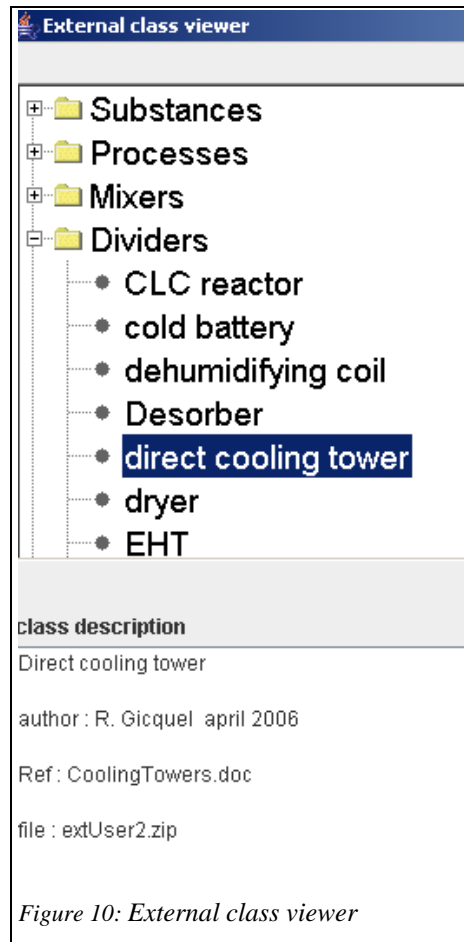


Figure 10: External class viewer