

# Comparison of Advanced Cooling Technologies Efficiency Depending on Outside Temperature

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Haihua Zhao  
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September 2009



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**September 2009**

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**<http://www.inl.gov>**

**Prepared for the  
U.S. Department of Energy  
Office of Nuclear Energy  
Under DOE Idaho Operations Office  
Contract DE-AC07-05ID14517**

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## Comparison of advanced cooling technologies efficiency depending on outside temperature

INL/EXT- 09-17392

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# **Comparison of advanced cooling technologies efficiency depending on outside temperature**

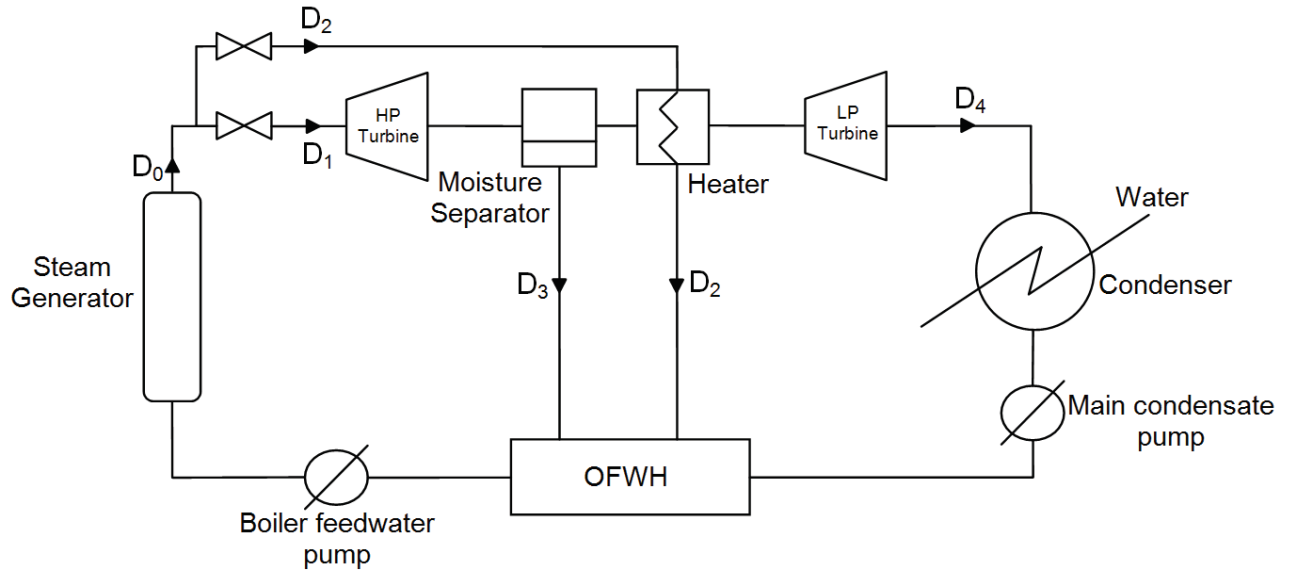
## **1. Introduction**

In some areas, water availability is a serious problem during the summer and could disrupt the normal operation of thermal power plants which needs large amount of water to operate. Moreover, when water quantities are sufficient, there can still be problem created by the waste heat rejected into the water which is regulated in order to limit the impact of thermal pollution on the environment. All these factors can lead to a decrease of electricity production during the summer and during peak hours, when electricity is the most needed. In order to deal with these problems, advanced cooling technologies have been developed and implemented to reduce water consumption and withdrawals but with an effect in the plant efficiency. This report aims at analyzing the efficiency of several cooling technologies with a fixed power plant design and so to produce a reference to be able to compare them.

## **2. The thermodynamic cycle**

### **2.1 Description of the cycle**

The cycle used to calculate the efficiency and to compare the cooling technologies is based on the cycle from a 900 MWe nuclear power plant described in *Abrégé de thermodynamique (Summary of thermodynamic)* (Fargue, 2004).



**Figure 1: The PWR Rankine cycle used in all calculations**

The steam generator produces 2970 MW of heat which is then used to produce the electricity. In the cycle, the different flow rates are:

- D0 = 1200 kg/s
- D1 = 1100.4 kg/s
- D2 = 99.6 kg/s
- D3 = 118.8 kg/s
- D4 = 980.4 kg/s

For normal operation, with a condensing temperature of 39 °C which corresponds to a pressure in the turbine of 0.070 bar, the data in the cycle can be found in Table 1. In nominal operation, turbines are considered to have an efficiency of  $\eta = 0.816$  and pumps of  $\eta = 0.9$ .

**Table 1 : Points data for nominal operation**

Point	P (bar)	T (°C)	x	h (kJ/s)	s (kJ/K.s)
A	56	271	1	2789	5.92
B	50	264	0.997	2789	5.96
Cis	11	184	0.866	2512	5.96
C	11	184	0.891	2563	6.08
J	11	184	0	781	2.18
D	11	184	1	2780	6.55
E	11	253	1	2946	6.89
H	0.070	39	0	163	0.559
Gis	0.070	39	0.820	2139	6.89
G	0.070	39	0.882	2288	7.36
Iis	15	39	0	164	0.559
I	15	39	0	165	0.559
F	50	264	0.997	2789	5.96
K	50	264	0	1154	2.92

L	11	73	0	309	0.997
Mis	70	74	0	314	0.997
M	70	74	0	315	0.999

Figure 2 : The T-s diagram of this Rankine cycle

## 2.2 Hypothesis

In order to be able to complete all the analysis, some hypotheses have been made and are true for all following calculations:

- a. The pressure in A is considered steady and equal to its nominal value for all calculations. This leads to determine the state of the points A, B, C, D, E, F, J, K independently from the cooling technology.
- b. The power provided by the steam generator is considered steady.

## 3. Once-through cooling

Once through cooling directly withdraws water from a river, a lake or ocean and rejects it “instantly” at a higher temperature. This cooling technology is efficient and very simple but it causes thermal pollution by heating the cooling water so the downstream temperatures are now strictly regulated and during summer time, a plant would have to close if these regulations can not be respected.

Figure 3 : Scheme of once-through cooling operation

### 3.1 Model

First is considered a once-through cooling system using water from a river as a cooling fluid. The model will study the thermal efficiency and the net power output of the power plant depending on the water inlet temperature while respecting an arbitrary limit in water outlet temperature of 35°C.

The important parameters for this case are:

- $T_s$ : the temperature in the condenser corresponding with the turbine backpressure.
- $T_i$ : the water inlet temperature.
- $T_o$ : the water outlet temperature.

In the condenser, we will use the equations:

$$\dot{Q} = A \alpha_0 LMTD, \quad (3.1)$$

$$\dot{Q} = c_p (T_o - T_i) \dot{m}, \quad (3.2)$$

$$\dot{Q} = \dot{Q}_r (1 - \eta_{th}), \quad (3.3)$$

with  $\dot{Q}$  the heat exchanged in the condenser,  $A$  the area of heat exchanged ( $A = 51816 \text{ m}^2$ ),  $\alpha_0$  a characteristic of the condenser, which is  $3000 \text{ W/m}^2\text{-K}$  here, and  $LMTD$  the Log Mean Temperature Difference :

$$LMTD = \frac{(T_s - T_o) - (T_s - T_i)}{\ln\left(\frac{T_s - T_o}{T_s - T_i}\right)}, \quad (3.4)$$

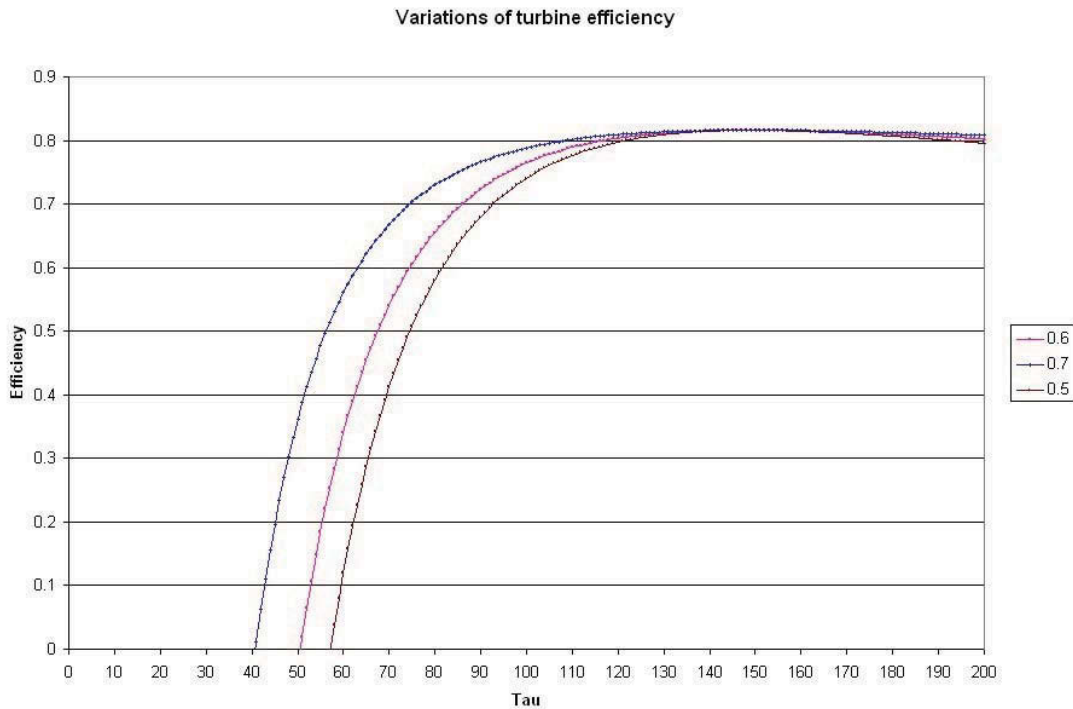
$c_p$  is the heat capacity of water, considered constant,  $\dot{m}$  the flow rate of the cooling water,  $\dot{Q}_r$  the heat from the steam generator and  $\eta_{th}$  the thermal efficiency.

However, due to the large range of condensing temperature reached depending on the water temperature, the variation of the efficiency of the turbine has to be taken into account.

These variations can be expressed as a function of the compression ratio  $\tau = \frac{P_{in}}{P_{out}}$  using the relation (3.5) given in Systèmes énergétiques (Energy Systems) (Gicquel, 2009).

$$\eta_{turbine} = \eta_{lim} + (\eta_{max} - \eta_{lim}) \left[ 2 \frac{\tau_{max}}{\tau} - \left( \frac{\tau_{max}}{\tau} \right)^2 \right], \quad (3.5)$$

where  $\eta_{lim}$  is the asymptotic efficiency limit of the turbine, i.e. the efficiency when the compression ratio is infinite,  $\eta_{max}$  the maximum efficiency of the turbine which is reached for nominal operation and  $\tau_{max}$  the pressure ratio corresponding to this value.



**Figure 4 : Variation of turbine efficiency depending on the variations of pressure ratio for several asymptotic efficiency limit values**

This relation allows calculating the thermal efficiency  $\eta_{th}$  of the cycle for different values of the temperature in the condenser  $T_s$ . For these calculations, an average value of  $\eta_{lim} = 0.6$  has been chosen and  $\eta_{max}$  corresponds to the efficiency for nominal operation for this design which is 0.816. In nominal operations, the pressure ratio is  $\tau_{max} = 149$ . Then, a regression of the relation between  $\eta_{th}$  and  $T_s$  is obtained and used for the calculations in different configurations (Figure 5).

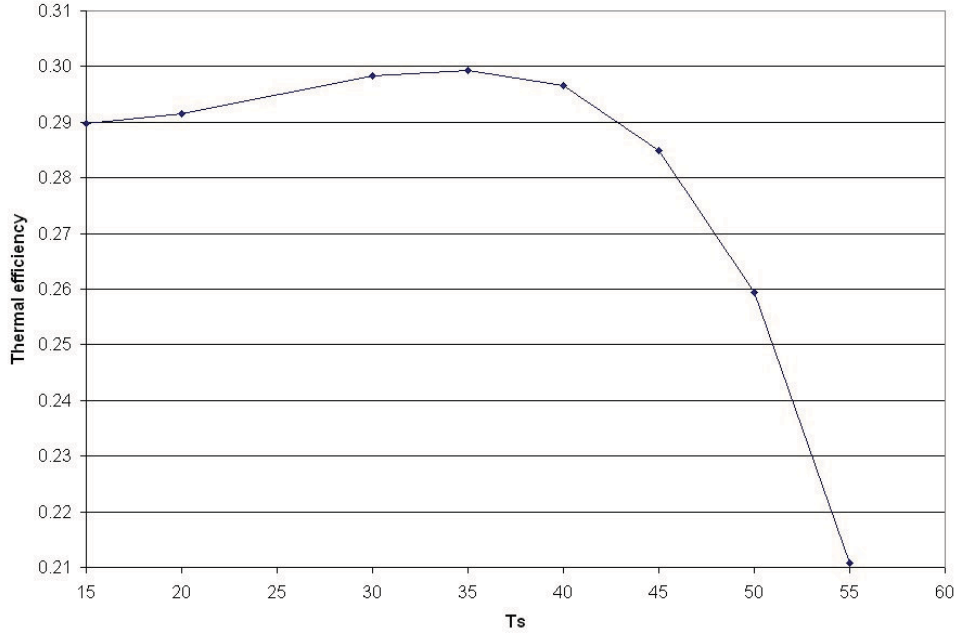


Figure 5: Curve of the thermal efficiency  $\eta_{th}$  depending on  $T_s$

The regression is

$$\eta_{th} = aTs^4 + bTs^3 + cTs^2 + dTs + e = f(Ts), \quad (3.6)$$

with  $a, b, c, d, e$  coefficients determined by Excel.

From these equations, (3.2) and (3.3) can be put together and become

$$\dot{Q}_r = \frac{c_p (T_o - T_i) \dot{m}}{1 - f(Ts)}, \quad (3.7)$$

when  $\eta_{th}$  is replaced by  $f(Ts)$

And so,

$$T_o = \frac{\dot{Q}_r}{\dot{m}c_p} (1 - f(Ts)) + T_i \quad (3.8)$$

Using Eqs (3.1), (3.2), and (3.4), we have

$$\frac{T_s - T_o}{T_s - T_i} = e^{-\frac{A\alpha_0}{\dot{m}c_p}} \quad (3.9)$$

$T_o$  is replaced in (3.9) by its expression in (3.8) and (3.10) is then solved in  $T_s$

$$T_s - \frac{\dot{Q}_r}{c_p \dot{m}} (1 - f(Ts)) - T_i = (T_s - T_i) e^{-K} \quad (3.10)$$



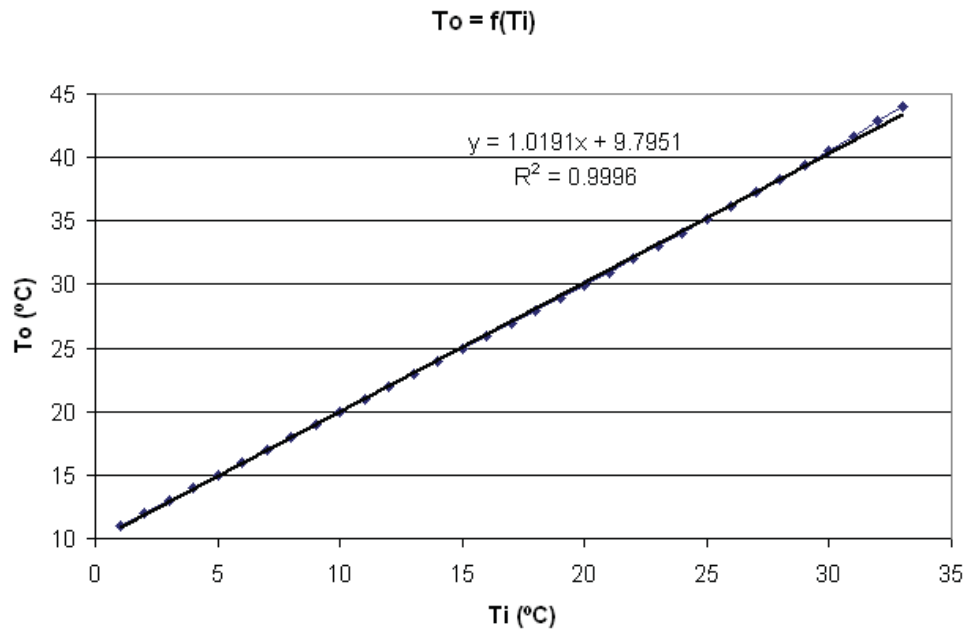
with  $K = \frac{A\alpha_0}{c_p\dot{m}}$ . Table 2 summarizes main parameters and their reference values.

**Table 2 : Summary of main parameters used for the calculations**

$\dot{Q}_r$ (kW)	2970000
$\alpha_0$ (kW/m <sup>2</sup> -K)	3
$A$ (m <sup>2</sup> )	51816
$\dot{m}$ (kg/s)	50000
$c_p$ (kJ.K <sup>-1</sup> .kg <sup>-1</sup> )	4186

### 3.2 Results

With the equations presented above we can find  $T_s$ , then  $T_o$  and  $\eta_{th}$  using (3.6) and (3.8) depending on the inlet water temperature  $T_i$ .



**Figure 6 :  $T_o = f(T_i)$**

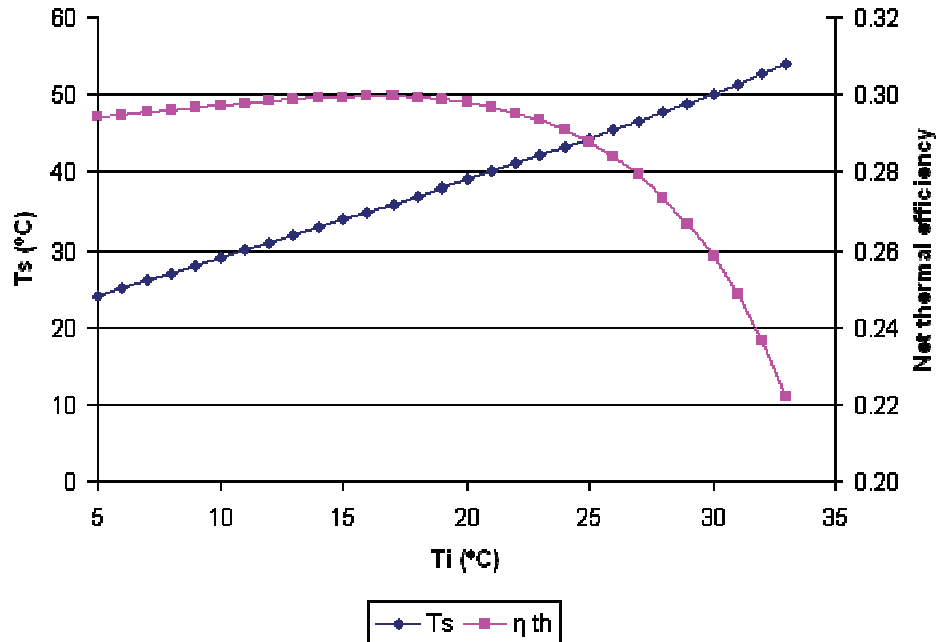
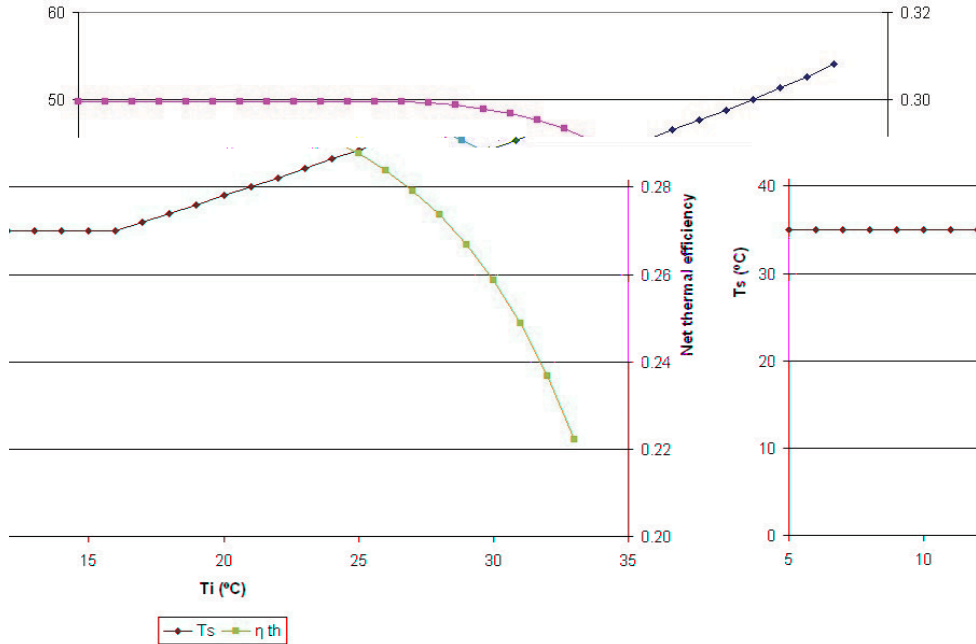


Figure 7:  $T_s$  and  $\eta_{th}$  depending on  $T_i$

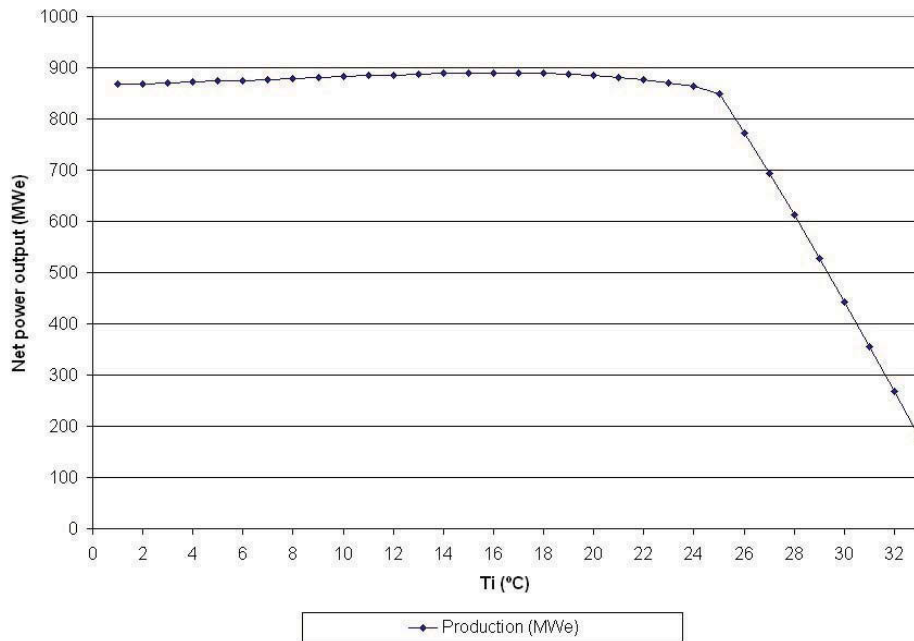
The relations between  $T_o$  and  $T_i$  as well as  $T_s$  and  $T_i$  are linear, an increase in the inlet temperature creates an increase in the outlet temperature and so in the condensing temperature. However, the thermal efficiency is greatly affected by the variations of the turbine efficiency, especially at high inlet temperatures. These temperatures correspond to a high condensing temperature which means less work retrieved and so less utilitarian work but they also correspond to a lower pressure ratio which is where the turbine efficiency drops.

From 5°C to 20°C the net thermal efficiency decreases because the turbine efficiency is leaving its nominal operation range. However, in real operations, it is very easy to play with the cooling water flow rate in order to stay at a steady condensing temperature. Thus the real shape of the curve of efficiency and condensing temperature would more look like the Figure 8.



**Figure 8 : Shape of the curves of condensing temperature and efficiency in real operation**

However, the temperature limit of water withdrawal is here not taken into account. The Figure 9 represents the net power output depending on water inlet temperature. Here, if the outlet temperature is more than the regulatory limit, the power provided by the core is calculated to have a water outlet temperature to respect the regulation, here 35 °C. Thus, the Figure 9 shows that in this case, the power production drops very quickly. Moreover, physical limits in reducing the power provided by the core are not taken into account here, and the plant may have to be shut down to respect these limits.



**Figure 9 : Evolution of net power output and thermal efficiency depending on  $T_i$  when a regulatory limit of 35°C is respected for  $T_o$**

## 4. Hybrid cooling

The hybrid cooling system considered here is a dry/wet cooling system. This system uses an air cooled condenser, to evacuate a part of the heat while saving water, and a wet cooling tower to evacuate the rest of the heat, in order to keep a good efficiency. First, the dry cooling system will be described then the wet cooling tower and finally the results from the hybrid system will be presented. The parameters used here will be:

- $T_s$ : the temperature in the condenser.
- $T_i$ : the inlet water temperature.
- $T_o$ : the outlet water temperature.
- $T_1$ : the inlet air temperature.
- $T_2$ : the outlet air temperature.
- $\dot{m}_a$  : mass air flow rate of air
- $L$ : water mass flow rate for wet cooling

### 4.1 The dry cooling part

#### 4.1.1 Generalities

The dry cooling considered for the model is a direct air cooled condenser, the most frequent in the USA, operated with a mechanical draft.

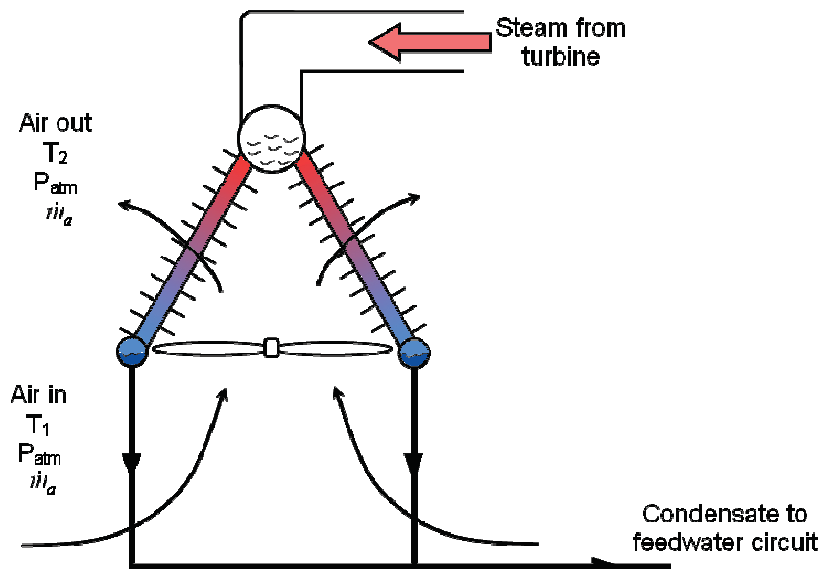


Figure 10 : Scheme of the direct air cooled condenser mechanism

With dry cooling systems, the characteristic quantity that will qualify the cost and the efficiency of the system is the Initial Temperature Difference  $ITD = T_s - T_1$ . A low  $ITD$  represents a larger, higher-capital-cost unit that is able to maintain a low condensing temperature on hot days while a high  $ITD$  represents a smaller and less costly tower that results in higher condensing

temperatures, higher turbine back pressures, and lower plant efficiency during hot periods. Usually, the *ITD* ranges from 10°C to 30°C and the temperature increase in the exchanger is from 10°C to 18°C (Comparison of alternate cooling technologies for U.S. power plants, EPRI, 2004).

### 4.1.2 Operating mode

In this report, one value of *ITD* is considered and with this value the variation of the efficiency is studied.

The value used will be 11°C which is expensive in capital but allows the best efficiency. It represents so the upper limit of dry cooling.

Doing the hypothesis the air heat capacity is constant in our range of temperature; the ACC follows the same equations the condenser does in once-through cooling.

$$\dot{Q} = A \alpha_0 LMTD, \quad (4.1)$$

$$\dot{Q} = c_p (T_2 - T_1) \dot{m}_a, \quad (4.2)$$

$$\dot{Q} = \dot{Q}_r (1 - \eta_{th}), \quad (4.3)$$

The only difference is that  $\alpha_0$  is calculated as a function of the air flow rate. Here,  $\alpha_0$  is approximated to the heat transfer coefficient  $h$  given by the formula:

$$h = \frac{k_w}{D_H} Nu$$

With  $k_w$  the thermal conductivity,  $D_H$  the hydraulic diameter,  $Nu$  the Nusselt number which is here equal to  $Nu = 0.023 Re^{4/5} Pr^{0.4}$  using Dittus-Boelter correlation.

In dry cooling, the turbine needs to be able to reach a wider range of backpressures and new turbines have been developed for these uses: conventional modified turbines and high-backpressure turbines.

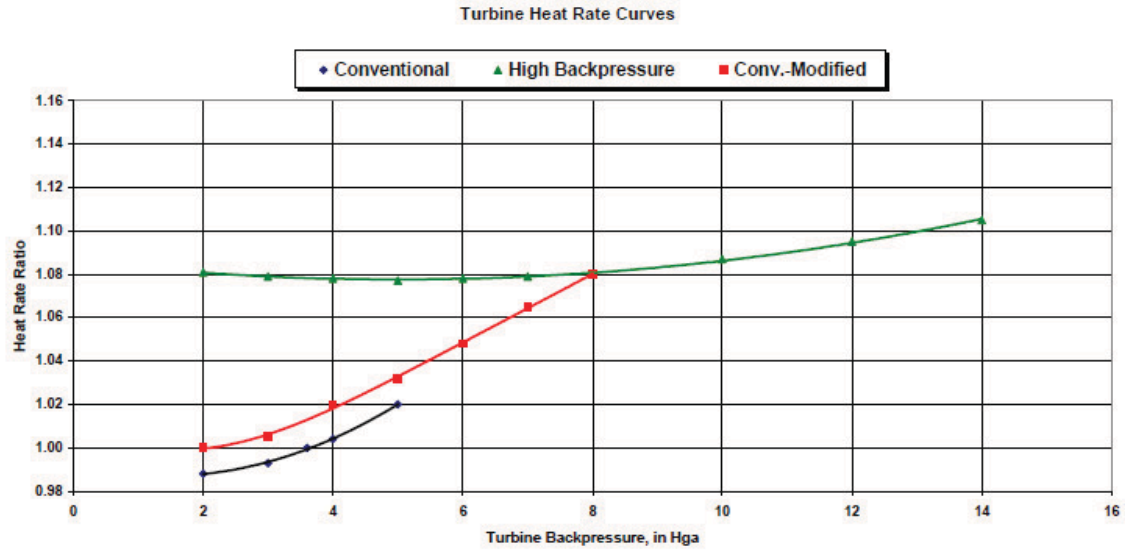


Figure 11: Comparison of heat rate characteristics for different turbines (Comparison of alternate cooling technologies for U.S. power plants, EPRI, 2004)

In order to model such a turbine, the characteristic of the efficiency of the turbine has been modified:  $\eta_{lim}$  has been increased in order to have a wider range of operation before the efficiency drops and the  $\eta_{max}$  has been decreased.

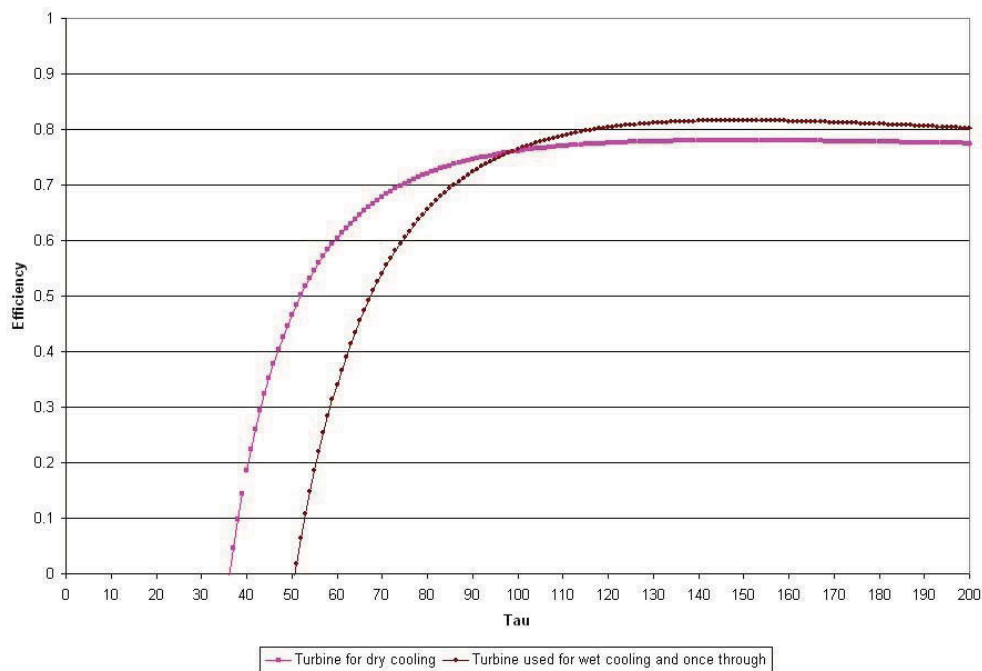


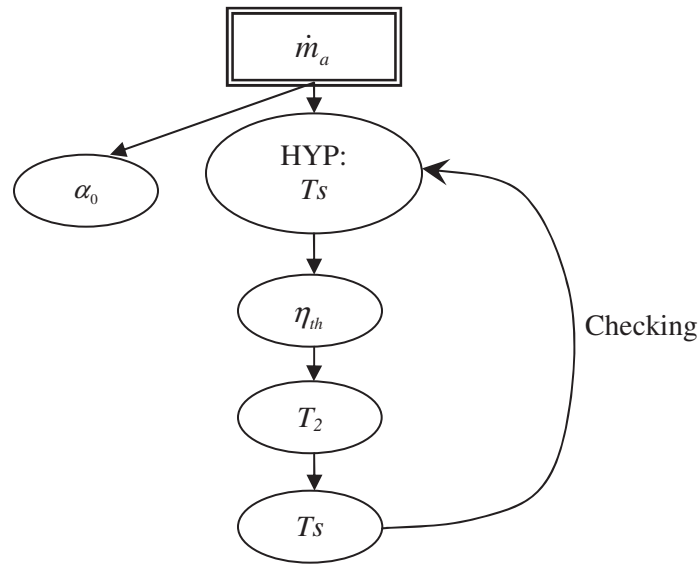
Figure 12: The variations of turbine efficiency for the two types of turbines

and the relation (3.6):  $\eta_{th} = aTs^4 + bTs^3 + cTs^2 + dTs + e = f(Ts)$  is now

$$\eta_{th} = a'Ts^4 + b'Ts^3 + c'Ts^2 + d'Ts + e' = g(Ts), \quad (4.4)$$

Moreover, the system is designed that the maximum efficiency corresponds with the wet cooling tower efficiency, in the limit of reasonable parameters, like condenser area, in order to be able to do relative comparison. The design value of the ITD is the minimum possible value but is not fixed so the mass air flow rate  $\dot{m}_a$  has to be fixed as a control parameter and the other parameters are then calculated in function of it.

First, knowing  $\dot{m}_a$  allows calculating  $\alpha_0$ , and the other parameters using the algorithm on Figure 13.



**Figure 13 : Algorithm for parameter calculation**

This algorithm gives back the values of  $T_s$ ,  $T_2$  and  $\eta_{th}$ . Knowing the air temperature and the specific volume of air, the volume air flow rate of air needed can be found. Then the power consumed by the fans is calculated in several steps.

The power consumed by the fans is  $P_{fan} = Da\Delta P$ , with  $Da$  the volume air flow rate. The pressure differential  $\Delta P$  created by the fans can be calculated:

$$\Delta P = K \cdot \frac{1}{2} \rho v^2 \quad (4.5)$$

With  $K$  is the total form loss,  $\rho$  the specific mass of air and  $v$  the speed of the air in the fan.

The number of fans and their diameter is needed to calculate the air speed. Since this information is not easily accessible, a comparison has been made with the new 550MW CCPP power plant of Astoria in New York City which uses 24 dry cooling modules (Dry cooling for power plants, an innovative modularization concept, Nagel & Wurtz, 2006). Knowing the heat that needs to be evacuated in this case and in the design used here, it is possible to make an approximation of the number of modules in the case studied here in order to be able to evacuate all the heat. The plant

needs here 135 modules which make an air entering cross-section area of 10079 m<sup>2</sup>. Since the volume air flow rate is known, the speed of this air can then be calculated.

Finally, the power consumed by the fans is determined by:

$$P_{fan} = \frac{K \cdot D_a^3}{2S_{total}^2 v}, \quad (4.6)$$

with  $v$  the specific volume and  $S_{total}$  the total air entrance area.

Then since  $\eta_{th} = \frac{\dot{W}_u}{\dot{Q}_r}$ , the net thermal efficiency, which takes into account the energy consumed by the fans, is equal to

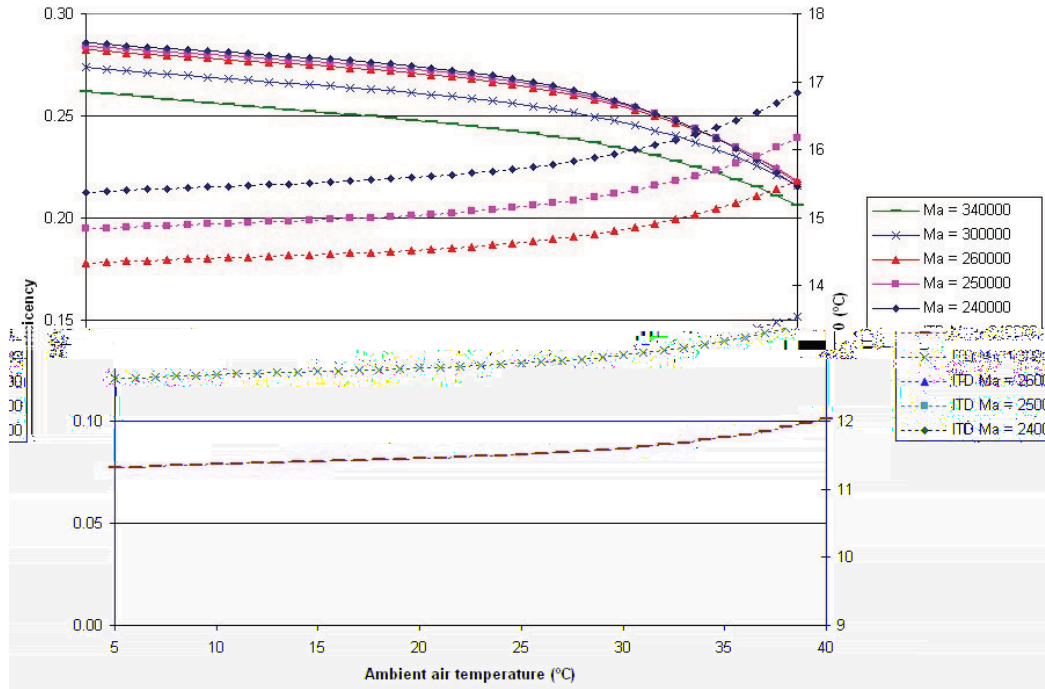
$$\eta_{net} = \eta_{th} - \frac{P_{fan}}{\dot{Q}_r} \quad (4.7)$$

### 4.1.3 Results

With this approach of the problem, the heat transfer area is fixed by the constraints and the mass air flow rate  $\dot{m}_a$  is a control parameter. The variation of the efficiency is studied for several different values of the air flow rate and then a best thermal efficiency is given with the air flow rate corresponding.

What limits the choice of values of  $\dot{m}_a$  is the possibility to evacuate all the heat in the condenser. In this model, this capacity is represented by the fact that it is possible to find a solution to the algorithm, with an air flow rate above its lower limit and respecting the minimal design value of *ITD* (11 °C here).

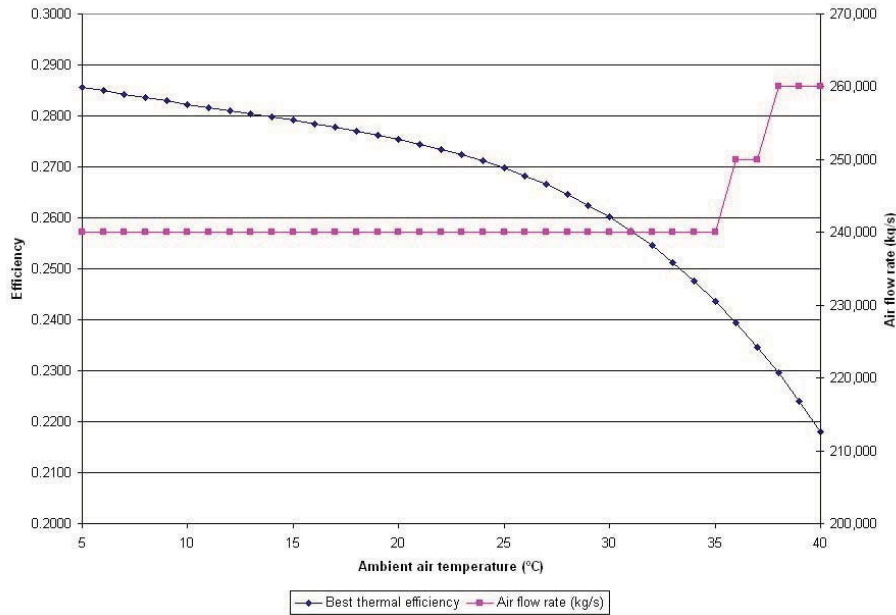




**Figure 14 : Variation of efficiency and ITD depending on the air flow rate for dry cooling**

Figure 14 shows the efficiency and ITD dependency on ambient air temperature. Here, the efficiency always decreases but from 25°C, the efficiency decreases much more. At low temperatures, the turbine efficiency is almost steady and the efficiency decreases because of the increase in the condensing temperature only. However, at high temperatures, both turbine efficiency and an increase in the condensing temperature are creating a decrease of efficiency which drops. It also appears that the curves the most efficient at low temperature are the ones with the lowest air flow rate because there are less power consumed in the fans and the condensation temperature is closer to optimal value, but at high temperature, the most efficient are the curves with a high air flow rate. The variation of turbine efficiency is still dominant and the surplus of power consumed in the fans is still less than the losses due to turbine efficiency variations when the condensing temperature increases. The optimal operation curve which has the best net efficiency should be the one that the air flow rate increases with ambient air temperature.

Concerning the *ITD*, the lowest value of *ITD* concerns the biggest flow rate because they allow having lower temperature difference. They should be more efficient but the losses in fan power make them less efficient at low temperature than curves with bigger *ITD*.



**Figure 15 : Best thermal efficiency with the corresponding air flow rate**

Figure 15 represents well the fact that more air is needed to cool the condenser as the temperature increases. It also represents well the non steady behavior of dry cooling efficiency since between the most efficient point and the least one, the efficiency drops of 24% in only 35°C. Moreover from nominal operation point to the point of maximum temperature, it drops of 21% in 20 °C.

Let's go back on the two different types of turbines. If we consider a conventional turbine on this configuration, we observe the curve on Figure 16. The two turbine characteristics have different shapes. The conventional one has of course the same shape as the one for once through and for the same reasons while the conventional modified doesn't follow the same variations. Since the value of  $\eta_{lim}$  has been increased, the efficiency reaches for a lower compression ratio  $\tau = \frac{P_{in}}{P_{out}}$  its

asymptotic value and there is very small decrease of efficiency between  $\tau_{max}$  and  $\tau_{\infty}$ . Thus the dominant effect is the variation of condensing temperature which makes that the efficiency is always decreasing. Moreover, we can also see that the fact that the maximum value of the turbine efficiency is lower with a conventional-modified turbine influence the whole cycle. Then, since the range of steady efficiency is wider with a conventional-modified turbine, the efficiency is also better at high temperature.

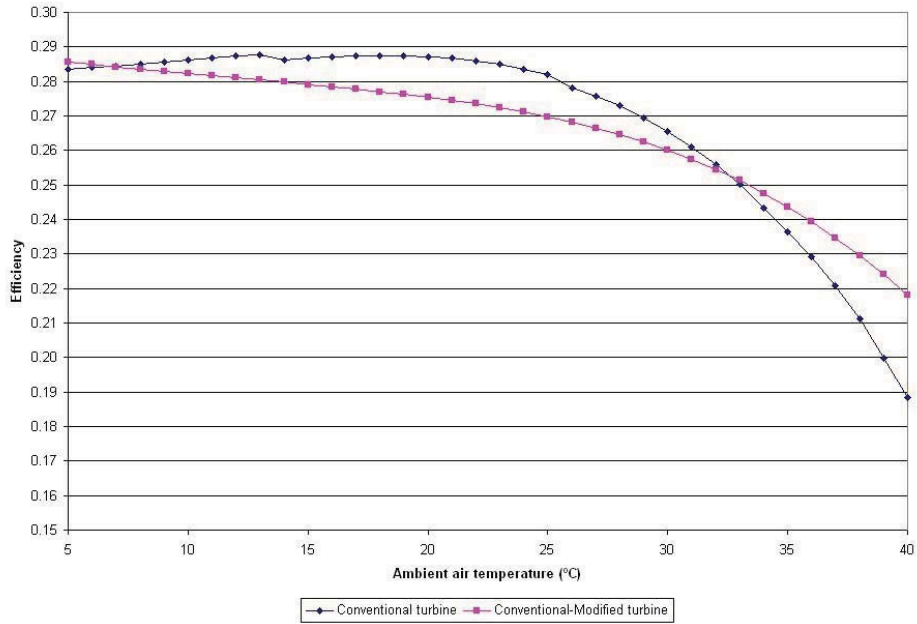


Figure 16: Net thermal efficiency depending on the temperature for a conventional and a conventional-modified turbine

## 4.2 The Wet cooling part

A wet cooling tower uses evaporation to remove waste heat. The main drawback of this technology is that a large quantity of water is consumed. Here a mechanical draft wet cooling tower as shown in Fig. 17 is considered and the limiting parameter will be water and/or energy consumption since these systems allow having quite stable efficiency.

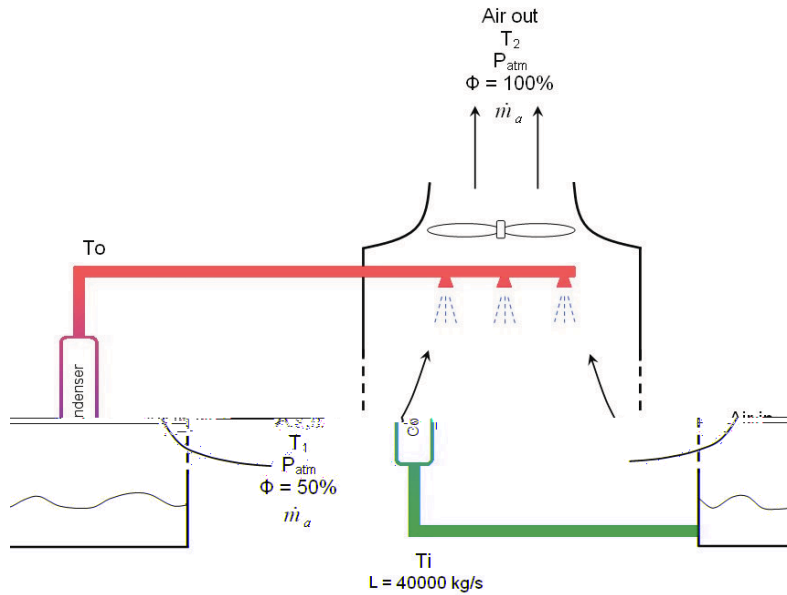


Figure 17 : A wet cooling tower with mechanical draft

The wet cooling tower is considered to have a constant  $Range = (T_o - T_i)$  so that the efficiency remains constant. This is possible by adjusting the water consumption and the air flow rate.

First, the condenser is designed so that the condensing temperature  $T_s$  is constant until a 1% wet bulb temperature which is only exceeded less than 1% of the time during the year. So, the condensing temperature is fixed and so are the cold and hot water temperature  $T_i$  and  $T_o$ . It is important to deal with the wet bulb temperature since the cooling water can't be cooled at a lesser temperature than the wet bulb temperature of the air entering the tower.

Then, according to the ambient temperature and relative humidity of 50%, the wet bulb temperature is determined from psychrometric charts. The outlet air temperature is fixed by design and so the air flow rate  $\dot{m}_a$  needed for condensing the steam can be determined using:

$$\dot{m}_a = \frac{L(h_{hotwater} - h_{coldwater})}{(h_2 - h_1) - (\omega_2 - \omega_1)h_{coldwater}}, \quad (4.8)$$

where  $h$  is enthalpy and  $\omega$  the humidity ratio. The subscript 1 corresponds to the entering air and 2 to the exiting air (Thermodynamics, an engineering approach, Cengel & Boles, 1989).

Then the power needed for the fans can be determined the same way it was done for dry cooling. It appears that this consumption is around a hundred of kW, and will be negligible.

The water consumption is approximated using (Wet cooling Tower : 'Rule of thumb', Leeper, 1981). This relation takes into account the water which is evaporated, the drift and the blow down.

$$\dot{m}_{make-up} = (0.0039R + 0.002)L, \quad (4.9)$$

with  $R$  the range in °F and  $L$  the mass water flow rate in kg/s.

With these parameters, Table 3 sums up the characteristics of the reactor which are steady while under the 1% wet-bulb temperature.

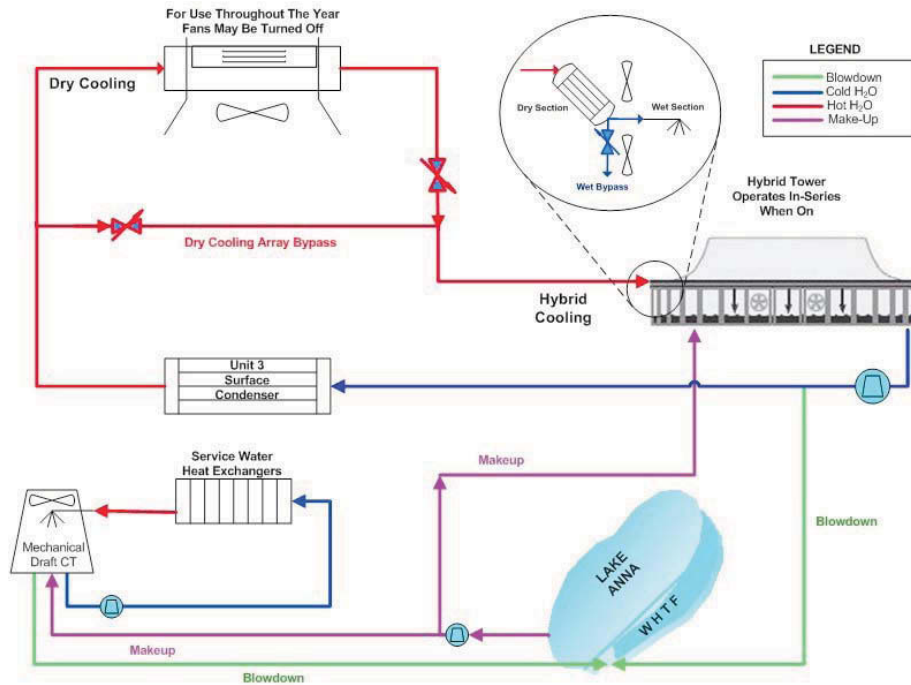
**Table 3 : Summary of wet cooling main parameters**

$A$ (m <sup>2</sup> )	95938
Range (°C)	10.03
$L$ (kg/s)	50000
$T_s$ (°C)	42.1
Efficiency	0.293
Water consumption (kg/s)	2049

## 4.3 The whole hybrid cooling system

### 4.3.1 Model

The whole hybrid system is based on the hybrid cooling system which is planned to be installed in North Anna Unit 3. The system which is planned to be installed there is a hybrid 1/3 Dry-2/3 Wet cooling system installed in series (Application of hybrid cooling technology for North Anna Unit 3, Waddil & Kemp, 2008).



**Figure 18 : Hybrid cooling system process flow diagram of North Anna Unit 3 (Application of hybrid cooling technology for North Anna Unit 3, Waddil & Kemp, 2008)**

In this system, since all biphasic fluids are considered to be on equilibrium, the condensing temperature, and so the thermal efficiency of the plant is fixed by the air cooled condenser for both wet and dry part.

That way, it is possible to calculate all parameters in the air cooled condenser the same way as it was done in the part about dry cooling, with only  $\dot{Q}_r$  reduced to one third of its value. However, it is not possible anymore to fix the same parameters it was done in the part about wet cooling. Since the condensing temperature is imposed, the temperature range has to be calculated as well as the hot and cold water temperatures. This is easy using the equations for once through, which are accurate here, and knowing the heat that has to be evacuated.

### 4.3.2 Results

Depending on the number of fans and their types, the results are different and the efficiency varies. However, in all cases because of the great influence of the turbine efficiency variations, the efficiency drops for high temperature and the functions of efficiency depending on the temperature for hybrid cooling and dry cooling have the same shape. Moreover, since the dry part is piloting the thermal efficiency, the same results as the ones for dry cooling are accurate here: efficiency decreases with the air temperature and the air flow rate needed for cooling increases with the temperature (shown in Fig. 19). Figure 20 compares wet cooling and two different hybrid systems. The types of hybrid cooling depend on the size of the fans which for a given flow rate will influence the speed of air and so it will impact the cooling a little bit.

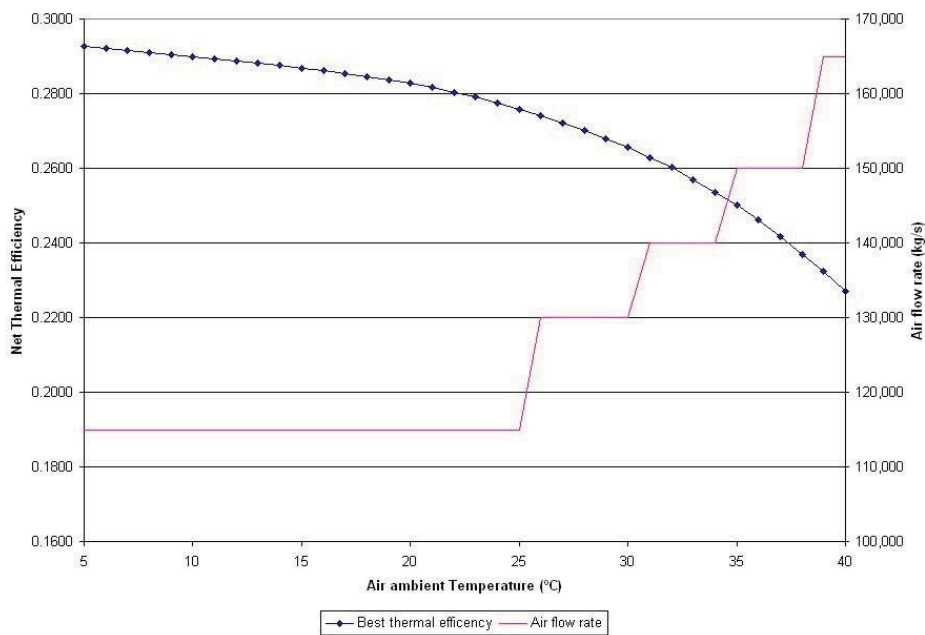
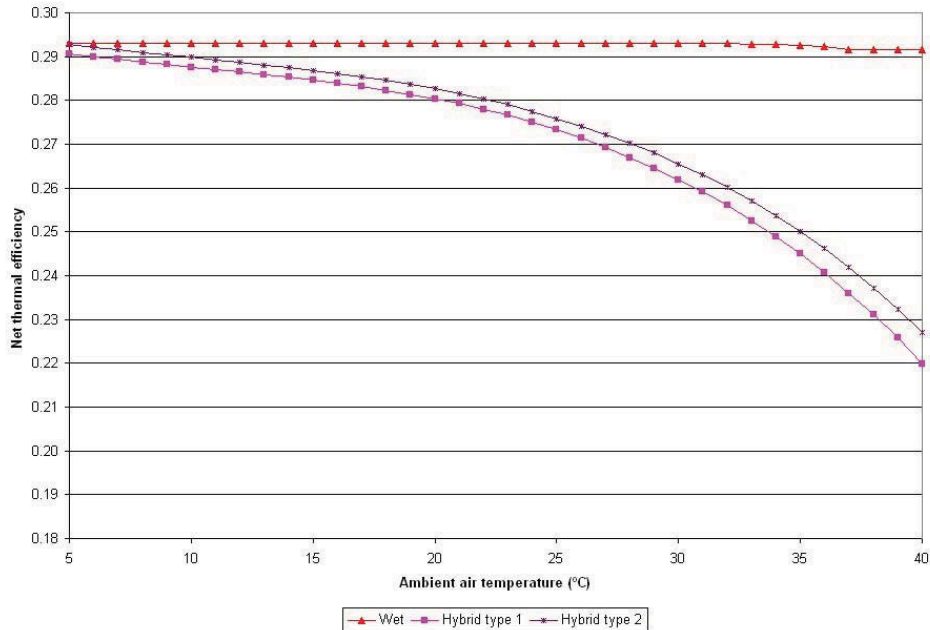


Figure 19: Net thermal efficiency of the plant with hybrid cooling



**Figure 20: Comparison of two types of hybrid cooling with wet cooling**

Whatever the choice of the characteristics for hybrid cooling, the water consumption is almost the same. Unfortunately, the period of time the savings are minimum is for high temperature, when water availability could be a problem. However, these results don't take into account the facts that the Wet-Dry ratio could be changed in order to have a better efficiency with larger water consumption or to reduce the water consumption even with the loss of net power output it creates.

Then the last point to take into account is the land footprint that is needed for the cooling system. Still comparing with the footprint of the air cooled condenser of the Astoria power plant in New-York, 10000m<sup>2</sup> to 12000m<sup>2</sup> will be needed for the dry part of a hybrid cooling system and 4000m<sup>2</sup>-5000m<sup>2</sup> more for the wet cooling part. In total it makes a footprint from 14000m<sup>2</sup> to 17000m<sup>2</sup>, only for constructed areas.

## 5. Ice thermal storage system (ITS)

The ice thermal storage system would use ice as a cooling media to a limited period of time and that way allows improving the instantaneous efficiency of the cycle during peak periods when electricity is needed and the most expensive. This system may be profitable because in the US peak hours correspond to the hottest hours of the year. The ice is made during off-peak periods on a daily, monthly or even yearly basis depending on the frequency of use and then stored in a tank. This system is so a supplemental cooling system aimed to be used only for a short period of time and added to a full cooling system.

The parameters used here will be:

- $T_s$ : the temperature in the condenser.
- $T_i$ : the inlet water temperature.
- $T_o$ : the outlet water temperature.
- $T_1$ : the inlet air temperature.
- $T_2$ : the outlet air temperature.
- $\dot{m}_{air}$  : mass air flow rate of air

## 5.1 Dry cooling with an ice thermal storage system

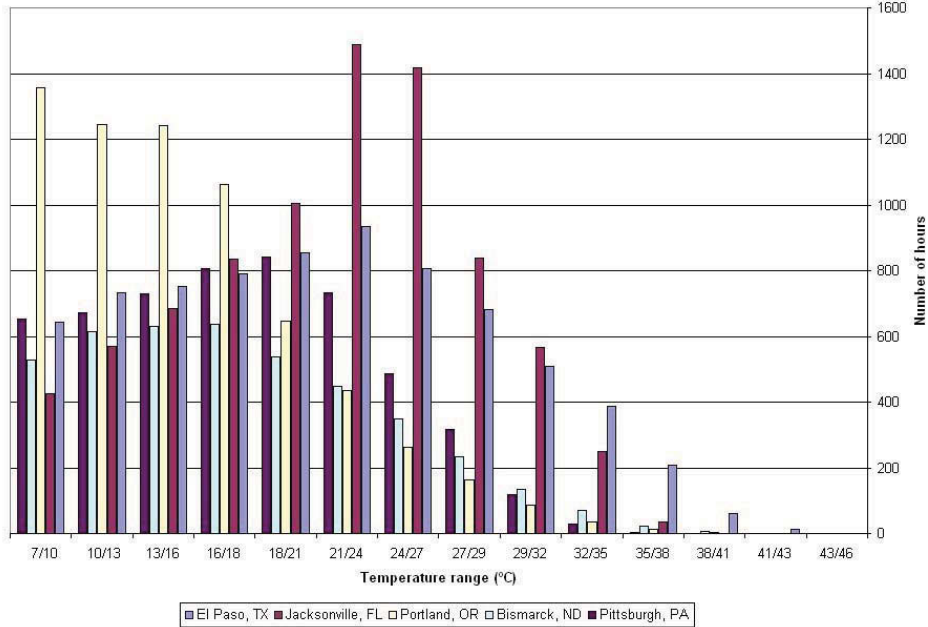
### 5.1.1 Model

The ice thermal storage system is added to the dry cooling system already studied previously in order to see what the gains in efficiency and in economics are. In order to compare these gains, the model is tested on the 5 different locations from the 2004 EPRI report (Comparison of alternate cooling technologies for U.S. power plants, EPRI, 2004) in order to have the gains with real temperature profiles. The system is only used when a temperature over 30°C is reached and calculations are made on a full year.

These five locations are:

- El Paso, TX
- Jacksonville, FL
- Bismarck, ND
- Portland, OR
- Pittsburgh, PA





**Figure 21: Yearly temperature profiles for the five locations studied (Comparison of alternate cooling technologies for U.S. power plants, EPRI, 2004)**

All these locations may not be eligible for either dry cooling or ice thermal storage system due to great quantities of water but the temperature profiles are used to see how they influence the energy increase during electricity peak periods.

The idea of ice thermal storage system is just being investigated therefore a specific technology for heat exchanger or a detailed cycle hasn't been defined and these calculations are just approximations based on the amount of ice needed to evacuate the heat from the plant by melting this ice.

Moreover, the results here concern an ice thermal storage system added to the dry cooling system already studied in part 4.1 in order to compensate the loss of efficiency during very hot periods. In this case, the ice cooling system has to be used from a certain temperature which must be optimized and the ice flow rate is controlled. Here the calculations are made with a maximum volume ice flow rate of  $2\text{m}^3/\text{s}$  to see what are the gains in energy during peak period.

$$\dot{Q}_{total} = \dot{Q}_{Dry} + \dot{Q}_{ice}$$

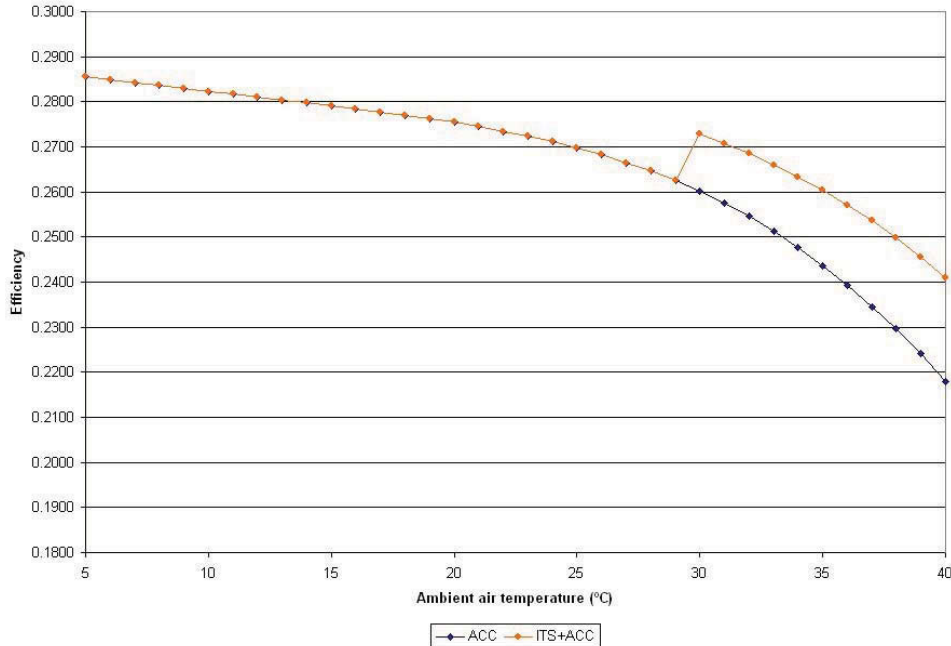
With  $\dot{Q}_{ice} = \dot{V}_{ice} h_{fusion} \rho$ .  $\dot{Q}_{ice}$  is the heat exchanged with the ice,  $\dot{V}_{ice}$  is the volume ice flow rate which is equal to  $2\text{m}^3/\text{s}$ ,  $h_{fusion} = 333.6\text{kJ}/\text{kg}$  is the fusion enthalpy of ice and  $\rho = 917\text{kg}/\text{m}^3$  the density of ice. Thus  $\dot{Q}_{ice}$  is fixed and the efficiency is determined by the value of  $\dot{Q}_{Dry}$  by the relation and using the relation between  $\eta_{th}$  and  $T_s$ .

$$Ts = \frac{\frac{\dot{Q}_{Dry}}{\dot{m}_{air} Cp}}{1 - e^{\frac{-A\alpha_0}{\dot{m}_{air} Cp}}} + Ti \quad (5.1)$$

To make a total balance on the yearly operation of the plant we also need to calculate the energy needed to make the ice:  $E_{ice} = se_{ice} \dot{V}_{ice} Time$  with  $\dot{V}_{ice} = \frac{\dot{Q}_{ice}}{h_{fusion} \rho}$  and  $se_{ice} = 26 \frac{kW.hr}{m^3}$  the energy needed to produce one cubic meter of ice.

### 5.1.2 Results

The ice thermal storage system allows an energy increase during the peak but in total, ice making needs more energy than the energy earned during the peaks. However, this energy is produced when it is the most needed and can be sold at a better price. Using an electricity baseload price of  $0.019 \frac{USD}{kW.hr}$  and an electricity peak price of  $0.108 \frac{USD}{kW.hr}$ , we can assess the benefits of this technology.



**Figure 22: Improvement of thermal efficiency (without the energy needed for ice making) by the addition of an ice cooling system on an air cooled condenser (ACC)**

**Figure 23: Energy increase during hot hours with an ice cooling system**

The figures show well that the best increase can be made in the place where there are high temperatures for the most hours. However, the dry cooling system used here is not optimized and with a system optimized to evacuate less heat knowing that an ice cooling system will be used in parallel, the energy increase could be greater in Bismarck where very high temperature can be reached but only for a few hours.

**Figure 24: Additional profits from electricity sales with ice thermal storage system**

These revenue increases doesn't take into account the cost to build the ice thermal storage which strongly depends on the location by the frequency of high temperature, the capacity of harvesting ice, the space available. Indeed, the price of the system will vary depending if the storage is made to keep enough ice for a month's use, a week or a day and depending on the choice of the technology, dynamic ice making or static ice making.

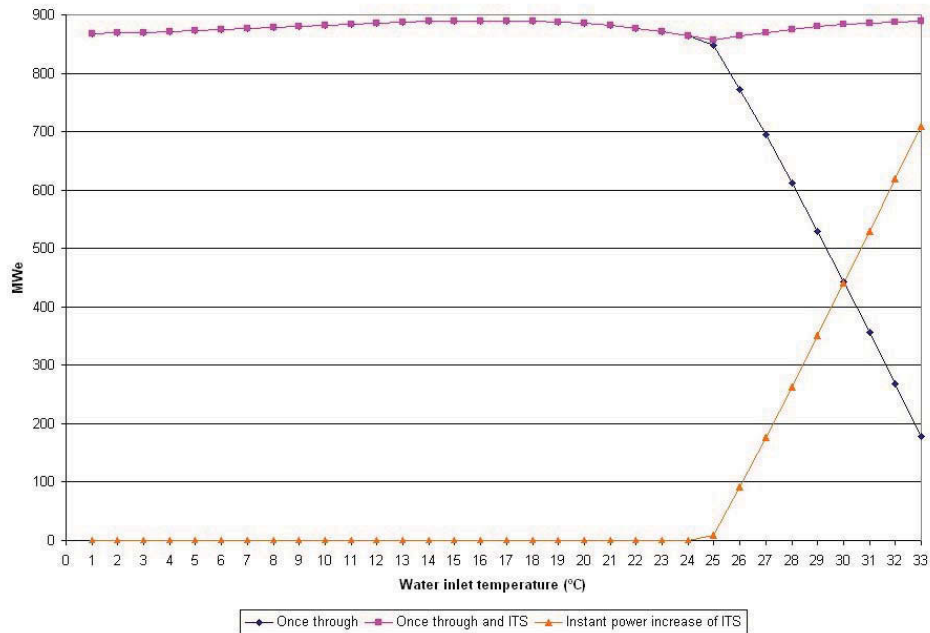
## 5.2 Once through and ice thermal storage system

The ice cooling systems allows, when the maximum temperature of discharge is reached, to use ice to compensate the heat that can't be evacuated into the river in order to keep a steady production.

We have

$$\dot{Q}_{total} = \dot{Q}_{once-through} + \dot{Q}_{ice}$$

when  $T_o$  is too high with  $\dot{Q}_{once-through}$  calculated like in part 1. Then, with  $\dot{Q}_{once-through}$ ,  $T_o$  and  $T_i$ ,  $T_s$  is known and so is  $\eta_{th}$  and the net power output. We can finally compare the gains in energy increase when the temperature is hot between the once through system with and without ice thermal storage.



The graph shows that a quasi steady output can be maintained and avoids reducing the power of the reactor. Moreover in case of very hot temperatures, it can be able to avoid the plant to close. Since the loss of power is very big and not just caused by an efficiency decrease, the gains by adding the ice cooling system are really significant.

Since  $\dot{Q}_{ice}$  is known, the volume of ice it corresponds to can be calculated in order to do an energy and economics balance.

Depending on the benefits in peak production, we can calculate the shifting efficiency. We have:

$$V_{ice} = \dot{V}_{ice} Time$$

With  $\dot{V}_{ice}$  equal the volume ice flow rate which is equal to  $\dot{V}_{ice} = \frac{\dot{Q}_{ice}}{h_{fusion}\rho}$  when

$h_{fusion} = 333.6kJ/kg$  is the fusion enthalpy of ice and  $\rho = 917kg/m^3$  the density of ice. Then, since  $E_{ice} = se_{ice}\dot{V}_{ice}Time$ , the energy needed can be calculated and then the shifting efficiency which is the gain in energy during the peak divided by the energy needed to make the ice, for one hour.

Here the shifting efficiency is always above 1 which translate the gains of ITS and once-through cooling.

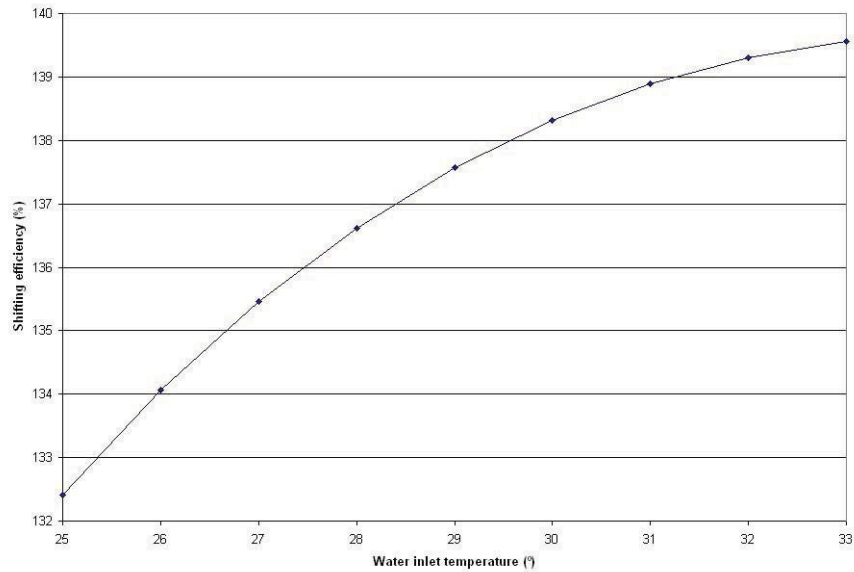


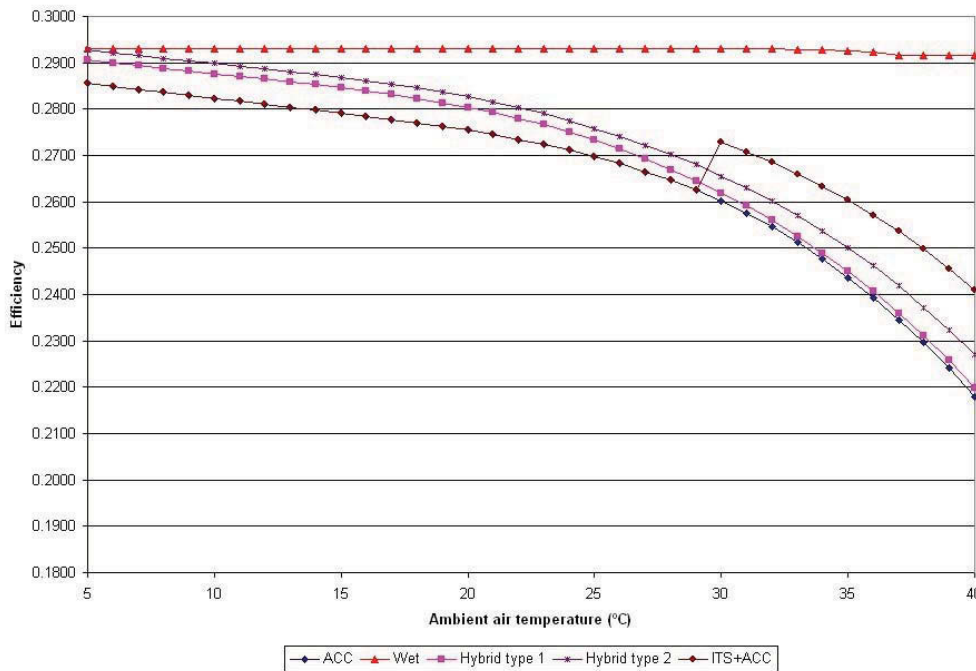
Figure 25: Variation of shifting efficiency

## 6. Technologies comparison

The comparison of these different technologies will be done on the basis of several factors:

- The *efficiency*, very important since it is directly related to the amount of electricity that can be sold
- The *water consumption*, which can be a limiting factor in some areas during the summer
- The *footprint of the condenser* which is limiting mostly in urban areas

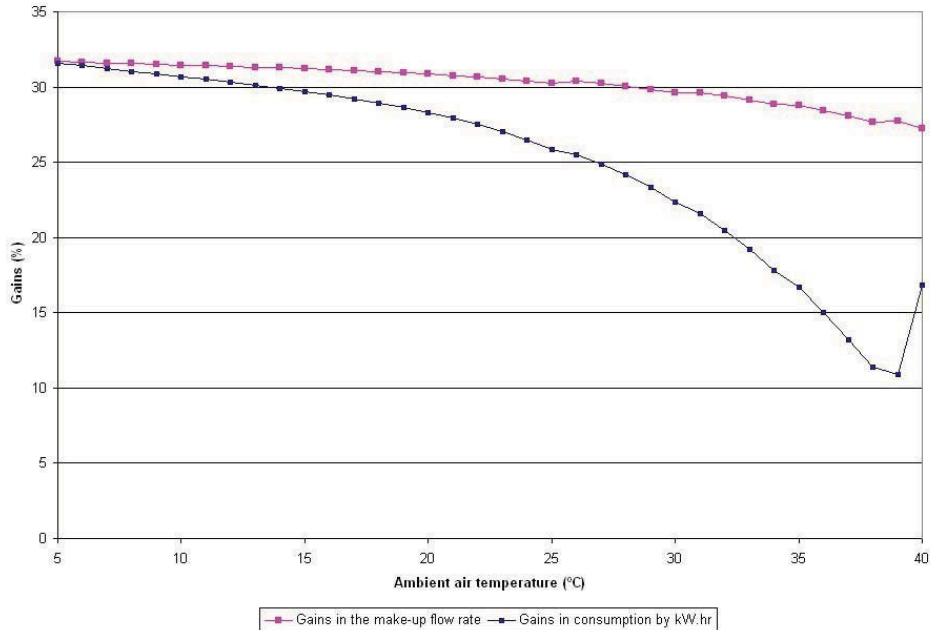
First, the efficiency is better with wet cooling towers and very stable, while ACC and hybrid can have efficiency close to wet cooling at low air temperature but their efficiency is also dropping when it's hot. However, the hybrid cooling system has a better efficiency than a dry cooling system, especially under 25°C even if there are still non negligible gains at higher temperatures. Unfortunately, at high temperature, the variations of turbine efficiency are dominating the variations and hybrid cooling and dry cooling presents the same variations of efficiency. Finally, adding an ice cooling system to an air cooled condenser allows good gains in efficiency and offer better results than hybrid cooling. Indeed, equation (5.1) shows well that reducing the quantity of heat exchanged in the Dry condenser will reduce the condensing temperature which has the effect to increase the thermal efficiency.



**Figure 26: Comparison of efficiency for several cooling technologies. Hybrid is represented two times but with different characteristics**

Then wet cooling towers also have a steady water consumption but greater than with the others technology. A dry cooling won't consume any water for cooling while a hybrid system 1/3 Dry-2/3 Wet will reduce its water consumption from 27% to 32% without using a water conservative mode (more fans power so less power output but less water consumption). However, due to the loss of efficiency at high temperature, the gain in water consumed by kWh varies much more: from 10% for the highest working temperature to 32%. This means very little since, either one has plenty of water and will use a wet cooling tower which is the most efficient or water availability is a problem and having a hybrid system will allow continuing to operate the plant even if the power output is limited.

For the ice cooling system, the water is used in closed cycle but may need blowdown like with other closed cycle cooling technologies. At this point the frequency of blowdown is not known and will have to be investigated to see how much water is consumed with this technology. However, the frequency of blowdown may be rather low since no impurities are accumulated.

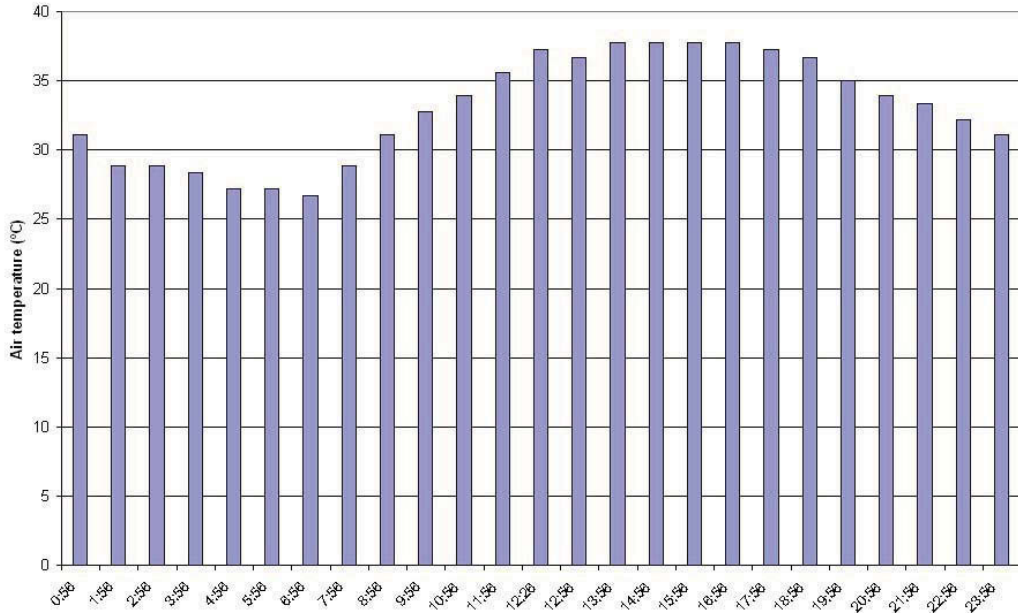


**Figure 27: Gains in water consumption with a 1/3 Dry-2/3 Wet hybrid cooling system compared to wet cooling**

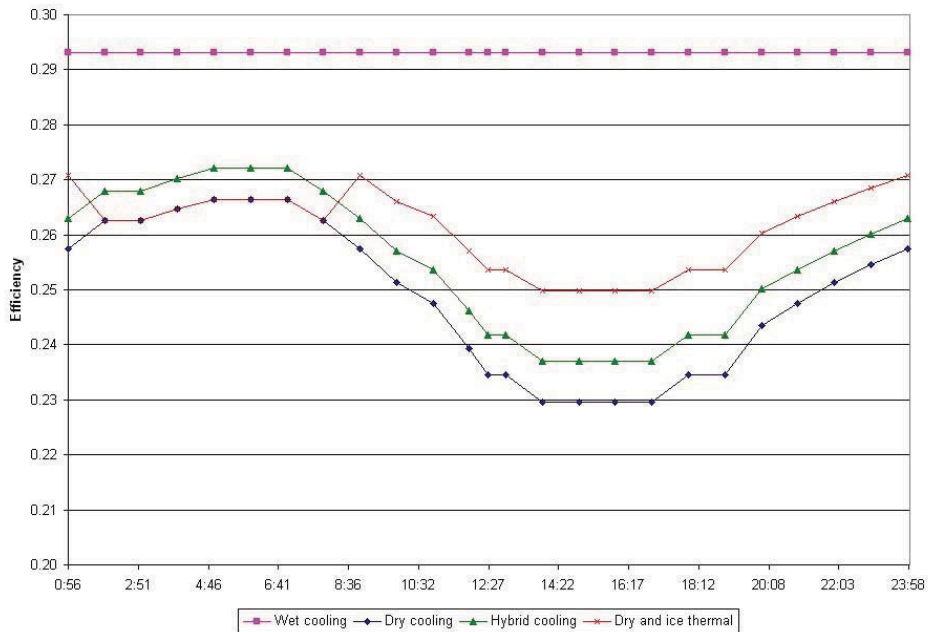
Finally, the footprint concerns plants which are located in urbanized areas and should not really concern nuclear power plants which usually have space around it. A dry cooling system as the one considered here may use around 25000m<sup>2</sup> while a hybrid system may only need around 15000 m<sup>2</sup> and a wet cooling tower 9000 m<sup>2</sup>.

## 7. Conclusion: comparison of the technologies on one day

To conclude, the efficiency of these technologies is compared during one day based on actual meteorological data from National Weather Service. The temperatures of the day chosen are specifically hot in order to study the efficiency when it makes sense to use alternative cooling technologies.



**Figure 28: Profile of temperature during the day studied**

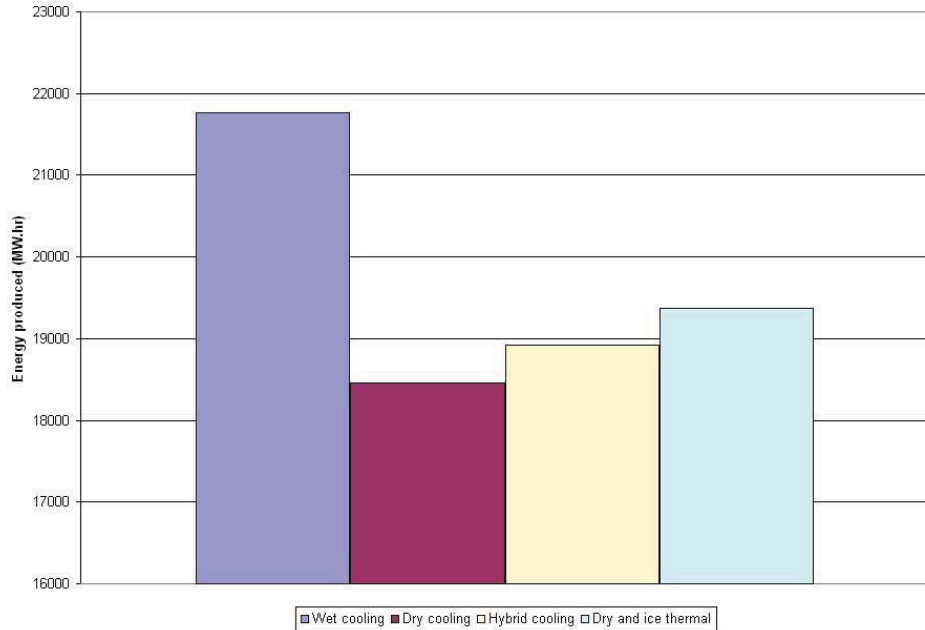


**Figure 29: Evolution of efficiency of different cooling technologies during the day**

This day fits into the sizing of wet cooling so its efficiency remains steady. On the opposite, the other efficiencies vary with the variation of air temperature. Since we only consider high temperature here, hybrid cooling is more efficient than dry cooling but the difference of efficiency remains the same the whole day. The ice cooling system, which is only used for a temperature above 30°C, shows better performance when ice is used than the two others technologies. Finally, wet cooling is still the most efficient but this situation is exactly a situation where water availability would be a problem and would limit the operation of the plant.

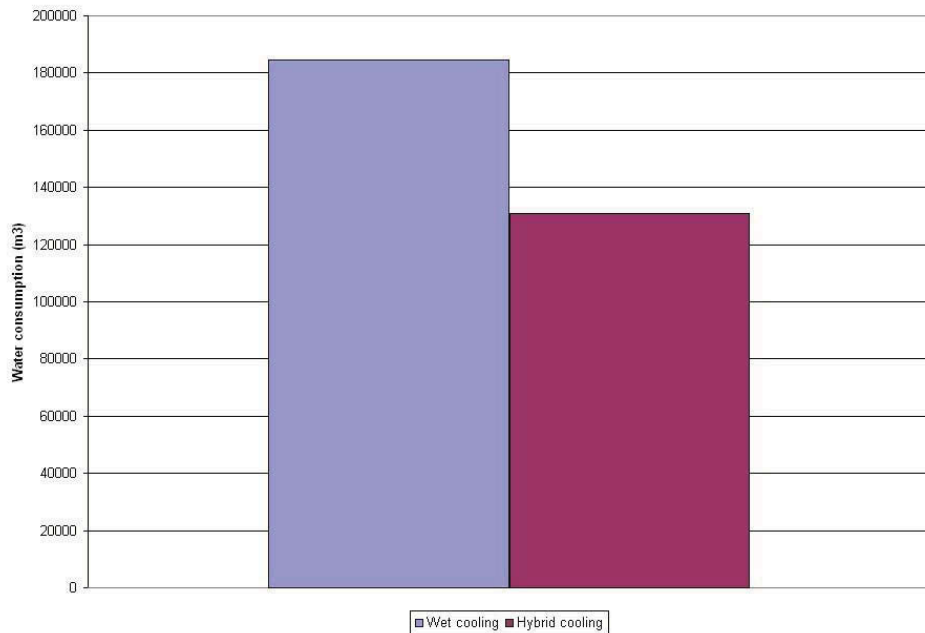


In total on the day, we can also compare the total production of electricity. Compared to the production with wet cooling, dry cooling produces 85%, Hybrid 87% and ice cooling 89% (without electricity consumption for ice making). The differences between the alternative technologies are very small and the loss compared to wet cooling is also rather limited.



**Figure 30: Total production of electricity for the day**

Moreover, regarding water consumption, using hybrid cooling instead of dry cooling allows saving 29% of water on average during that day.



**Figure 31: Comparison of total water consumption during the day**

Finally, compared with a wet cooling tower, the other cooling technologies are less efficient. However, in case wet cooling tower can't be implemented due to water availability problems, they offer a good alternative in efficiency and a wide range of solutions depending on the weather conditions of the plant. For example, hybrid is a good solution if water is available but limited while dry cooling is able to reduce almost to null the water consumption. Ice cooling system allows the best efficiency but its water consumption is yet to be determined but may be less than hybrid cooling. In this case, the problem may be more that a lot of energy is needed for making ice and so a cheap electricity production is needed during off-peak periods and that the capital cost of the storage may be very important.

## 8. References

- Cengel, Y. & Boles, M.(1989). *Thermodynamics, an engineering approach*.
- EPRI. (2004). *Comparison of alternate cooling technologies for U.S. power plants*. Palo Alto.
- Fargue, D. (2004). *Abrégé de thermodynamique (Summary of thermodynamics)*. Paris: Presses des Mines.
- Gicquel, R. (2009). *Systèmes énergétiques (Energy systems)* (Vol. 3). Paris: Presses des Mines.
- Leeper, S (1981). *Wet cooling tower: "Rule of thumb"*. Idaho Falls: INEL.
- Nagel, P., & Wurtz, W. (2006). *Dry cooling for power plants, an innovative modularization concept*. Cologne.
- Waddil, J., & Kemp, D. (2008). *Application of hybrid technology for north Anna unit 3*. ICAPP. Anaheim, CA.