TECHNICAL AND ECONOMICAL STUDY OF AN ORC DEDICATED TO THE PRODUCTION OF ELECTRICITY FROM A GEOTHERMAL SOURCE

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ABSTRACT

This paper presents an original method to design equipment of an organic Rankine cycle power plant. It is based on the entropy production equipartition theorem in the geothermal heat exchanger. The criterion of best pressure in favor of exergy at turbine inlet at given temperature is also used to initialize the problem at an ambient design temperature. According to these design criteria, the price of each main pieces of equipment is estimated. Then the equipment are simulated throughout a typical year, the net electrical production is deduced. The same method is repeated with different ambient temperatures chosen as design temperature. Then the product price per kWh is calculated and compared to the one obtained at the average design temperature. The methods is revealed not monotonous and so a filter is applied to results in order to keep the best cases regarding economical criteria. Three sites are studied. Regarding the ambient temperature climate distribution, a temperature where design appear to be the best, is found. Finally the developed simulation tool of ORC power plant allows to test the relevance of different design criteria.

1. INTRODUCTION

ORCs are proving to be an effective solution to produce mechanical or electrical power from low temperature heat source. These low temperature sources also make them sensitive to the sizing of their equipment (Schuller et al., 2014). Indeed, due to the evolution of the Carnot efficiency for ORCs influenced by ambient temperature (see figure 1 where equation (1) is plotted), the pinches involved in the evaporators or condensers have high consequences. This particular point must be carefully addressed. In these specific conditions, technical and economical trade-offs must be determined in order to make ORC competitive. This paper focuses on the technical-economic study of a supercritical ORC with propane, without regenerator, dedicated to the production of electricity from a geothermal source. The supercritical cycles are chosen for their ability to extract the heat from the hot source and the absence of the phase change plateau that avoids a large pinch inside the steam generator (Schuster et al., 2010). We firstly describe how the four main pieces of equipment (pump, geothermal heat exchanger, turbine and condenser) are designed by applying the entropy production equipartition theorem in the geothermal heat exchanger (Schuller et al., 2014). In the same time a specific criterion (Schuller et al., 2014) regarding working fluid in the turbine is used to constrain the problem and avoid liquid formation in the turbine. The costs of the four main items of equipment are estimated in relation to their sizing parameters (surface of exchanger, turbine power,...) designed at an arbitrarily chosen ambient temperature. The effect of this temperature selection on the design and the overall efficiency (energetic and economic) will be dis-

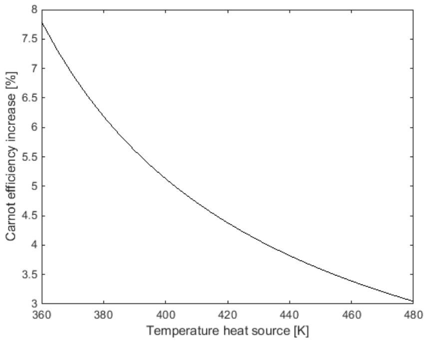

Figure 1: Carnot efficiency rise for a temperature drop from 283 **to** 277 **K**

$$f(T_{hot}) = \frac{(1 - \frac{277}{T_{hot}}) - (1 - \frac{283}{T_{hot}})}{1 - \frac{283}{T_{hot}}} * 100 \quad (1)$$

cussed. Secondly, this paper describes how equipment performances change according to the weather variation throughout the year. A typical ambient temperature distribution along the year is estimated with Meteonorm (Remund and Kunz, 1997)(see figure 2) and is used to calculate the yearly production. Thirdly, the same geothermal source is virtually placed in three sites in the world with different climates. By the previously described method three different ORC plants are designed. The comparison of the three plants is presented. This opens a discussion about the consequence of heat sink temperature variations. The consequence on the characteristics of equipment, designed at a fixed ambient temperature, but used at a variable ambient temperature is observed.

The three places chosen to illustrate the influence of the ambient air as heat sink are:

- Site 1, East of France
- Site 2, Turkey
- Site 3, Caribbean Island

The typical distribution of temperature given by Meteonorm (Remund and Kunz, 1997) for each site is presented in the figure 2 and the mean values are given in table 1

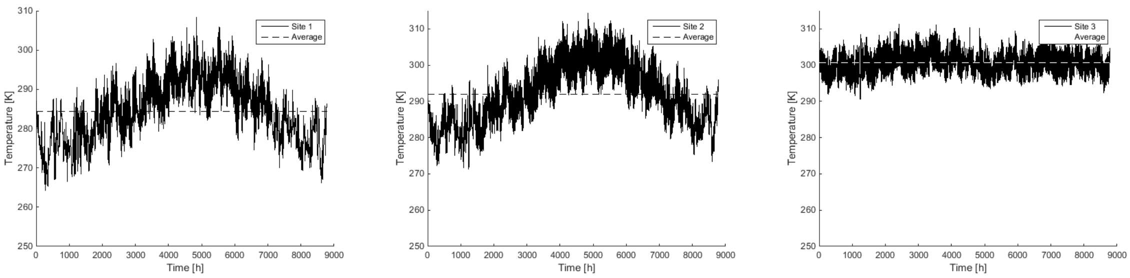

Figure 2: Hourly ambient temperature on typical year

Site	Average temperature (K)	Standard deviation (K)
1	284.5	8.0
2	292.0	8.6
3	300.6	3.5

Table 1: Average and standard deviation of ambient temperature distribution along a typical year

In the present work, the results are obtained by numerical simulations performed with a dedicated simulator developped with Matlab (Schuller et al., 2014).

2. THERMODYNAMICS MODEL

The thermo-physical fluids characteristics are calculated by Refprop (Lemon, 2013), directly linked to Matlab The main physical properties taken account are summarized in the table 2.

Physical properties	value	unit
Geothermal inlet temperature	435.15	K
Geothermal outlet temperature	273.15	K
Geothermal inlet pressure	2000	kPa
Geothermal mass flow	45	kg/s
Geothermal fluid	pure water	
Working fluid	propane	
Superheating margin (regarding cricon)	≥ 200	kPa
Hex Working fluid outlet entropy	≥ Cricondentrope entropy	
Pinch Hex	≥ 5	K
Expander inlet temperature	$> T_{crit} + 10$	K
Expander inlet pressure	$> P_{crit} + 50$	kPa
$\Delta T_{Condensation-Ambient\ air}$ (approach design)	18	K
Condenser working fluid pressure drop	20	kPa
Evaporator working fluid pressure drop	100	kPa
Evaporator geothermal fluid pressure drop	200	kPa

Table 2: Hypothesis summary

2.1 Details of equipment model

This supercritical ORC has four main items of equipment [1]:

- a pump transferring and pressurizing the working fluid in liquid phase.
- a geothermal heat exchanger transferring heat from the geothermal source, toward the working fluid.
- a turbine expanding the working fluid to remove the mechanical energy, which coupled to a generator via a gear box will produce electricity then injected on the electrical network.
- a condenser, which desuperheats and then condensates the expanded working fluid, in contact with the cold source, i.e. the ambient air.

Propane is chosen as a working fluid because adapted to the temperature of the hot source (Marcuccilli, 2010) (Sauret, 2011).

2.2 Condenser, cold source

The air cooled condenser is a classic model similar to API661 models that are found in ORC facilities for geothermal applications. The working fluid flows in finned tubes. Air is driven by fans. The power of the fans is calculated according to Robinson and Briggs correlation (Nir, 1991). Fan speed is assumed constant. Fan power depends only on ambient air temperature and its density (calculated directly by Meteonorm). Geometry is fixed, i.e tube length, diameter, fin width, height, fin density, as well as the number of tube rows. Only the number of tubes required by the condensation power is considered for the design calculation, whereas the condensation pressure and temperature will be considered for the off-design calculation.

2.3 Pump

The pump is of centrifugal type. Because of high fluid pressure (beyond its critical pressure), the variation in density between the input and output of the pump is significant. The model considers therefore, for the

[1]No regenerator is considered in this study

pump power consumption calculation, a polytropic compression with efficiency depending on its variable speed, head and flow (Troskolanski, 1977). The power and the speed are assumed to be sufficient to reach all the requirements for all off-design points.

2.4 Geothermal heat exchanger, hot source

The geothermal heat exchanger is of the shell and tube type. For cleaning considerations, geothermal water flows in the tubes. The exchanger is discretized in intervals of equally exchanged heat power . The diameter of tubes is considered constant, as well as pitch and baffles spacing and number. The number of parallel tubes depends only on water flow. The length of these tubes is calculated from the required exchange surface at design point.

2.5 Turbine

The turbine is of radial inflow type (Marcuccilli, 2007). The wheel diameter and the speed of rotation are determined to maximize isentropic efficiency (Balje, 1981). Pressure and temperature at turbine inlet are determined according to an optimization procedure (Schuller et al., 2014). The turbine is equipped with variable inlet guide vanes. The turbine inlet pressure is controlled by the position of these inlet guide vanes whereas the inlet temperature is governed by the working fluid mass flow rate (i.e. controlled by the pump).

3. ALGORITHM OF DESIGN AND OFF-DESIGN CALCULATION

The program contains a multitude of parameters. A lot of them are geometric, such as diameter of heat exchanger tube, fin height etc. In this study we focus on the following parameters:

- Ratio of mass flows, water over propane
- Pressure at turbine inlet
- Temperature at turbine inlet
- Design ambient temperature

Simulations are performed in two steps: design step and off-design step. In the design step, a specific ambient design temperature and condenser approach are chosen. Then the four main items of equipment previously presented are designed with respect to some optimization processes (Schuller et al., 2014) (Schuller, 2011). Optimal values of pressure and temperature at turbine inlet (see figure 3) and mass flows ratio are obtained at this stage. In figure 3, the thermal path of water is scaled to propane enthalpy as described in Augustine et al. (2009). In the second step, with off-design calculation, the performances of the designed plant are predicted keeping all the geometrical parameters constant. In this stage, the influence of the varying ambient temperature can be estimated. The flow of propane is calculated to reach the same turbine inlet temperature than at design. In real control scenario, it would be done by varying the speed of the pump. Finally, two different indicators are predicted, one with respect to the annual energy production, the other with respect to the economic efficiency of the plant, depending on the chosen business plan (see paragraph 5). The algorithm is detailed as follows (see figure 4): first, the temperature, at three quarter from supercritical to geothermal source temperature is chosen as the turbine inlet temperature. Thus the corresponding pressure on the curve EMTD[2] (see figure 3) is deduced for turbine inlet. The turbine outlet pressure is the condensation pressure corresponding to the ambient temperature plus the chosen approach shift from an arbitrary pressure drop. We consider no subcooling [3]. Then optimization is launched on the turbine inlet pressure and temperature and mass flow of propane, in order to minimize the variance of the reduced temperature difference to the average reduced temperature difference along the geothermal heat exchanger. In other words, the optimization makes the two thermal

[2]Maximal exergy at given temperature: for a given temperature, there exists a corresponding enthalpy or pressure where the exergy reaches a maximum (Tondeur, 2006)

[3]The NPSH, net pressure suction head, required by the pump is supplied by the height of the condenser

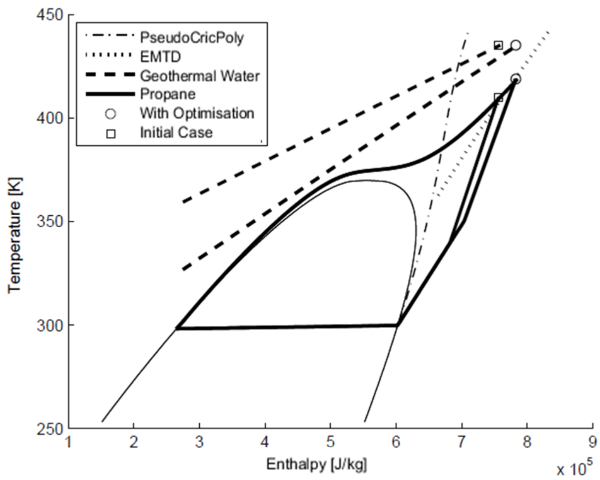

Figure 3: Thermal path of cycles before and after optimization to minimize the criterion of entropy production equipartition

paths (water and propane) as parallel as possible in a diagram of reduced temperature[4] versus enthalpy. The pinch point is not fixed as recommended byWalraven et al. (2014) for optimum issue, but it is a consequence of using the equipartition criterion. Furthermore, the turbine inlet point should be on the right hand side (in the H,T diagram) of the pseudocricondentropic curve (PseudoCricPoly in figure 3) in order to avoid liquid ingress during the expansion through the turbine, with respect of constraints and condition summarized in table 2. At this step, the design is fixed and equipment characteristics are defined. Off-design turbine inlet temperature and pressure are set to the design value. Temperature data from Meteonorm are implemented at condenser inlet, and at fixed characteristics, equipment performances are calculated:

- condenser approach
- condensation pressure
- fans power consumption
- pump speed
- pump efficiency and losses
- pump power consumption
- turbine nozzle opening
- turbine efficiency
- turbine, gearbox, generator losses
- generator power
- propane mass flow
- Hex pinch

This off-design is calculated hourly and production is assumed constant during one complete hour. Annual production is the sum of the power at each hour. In the figure 5, the example of site 1, design at average temperature is given. Power and production are reduced by the power and respectively production, produced at the average temperature. The coldest temperature gives the greatest power, but a poor occurrence. Average temperatures give average power values but have the highest occurrences. Hottest temperatures give the lowest power values but have poor occurrences. The consequence is illustrated in figure 6. The power and contribution to annual production are not correlated. Then at which temperature should equipment be designed: at a frequent temperature or at a temperature that gives the best power? To find out, different design temperatures are tested in the paragraph 4.

4. OFF-DESIGN REGARDING THE CHOSEN DESIGN TEMPERATURE

The method described in the paragraph 3 is applied for different design temperatures. These temperatures are chosen among temperature site occurrences, every 0.5 K. All results are reduced by a reference production. This reference is the production of a complete year in site 1, in the case of design at the average temperature of this site. These results are shown in figure 7. The shape of curves are non monotonous. This is due to the chosen criterion of entropy production equipartition. This is different to minimizing the entropy production. The algorithm of optimization is based on a simplex method, with

[4] temperature reduced by the Carnot factor

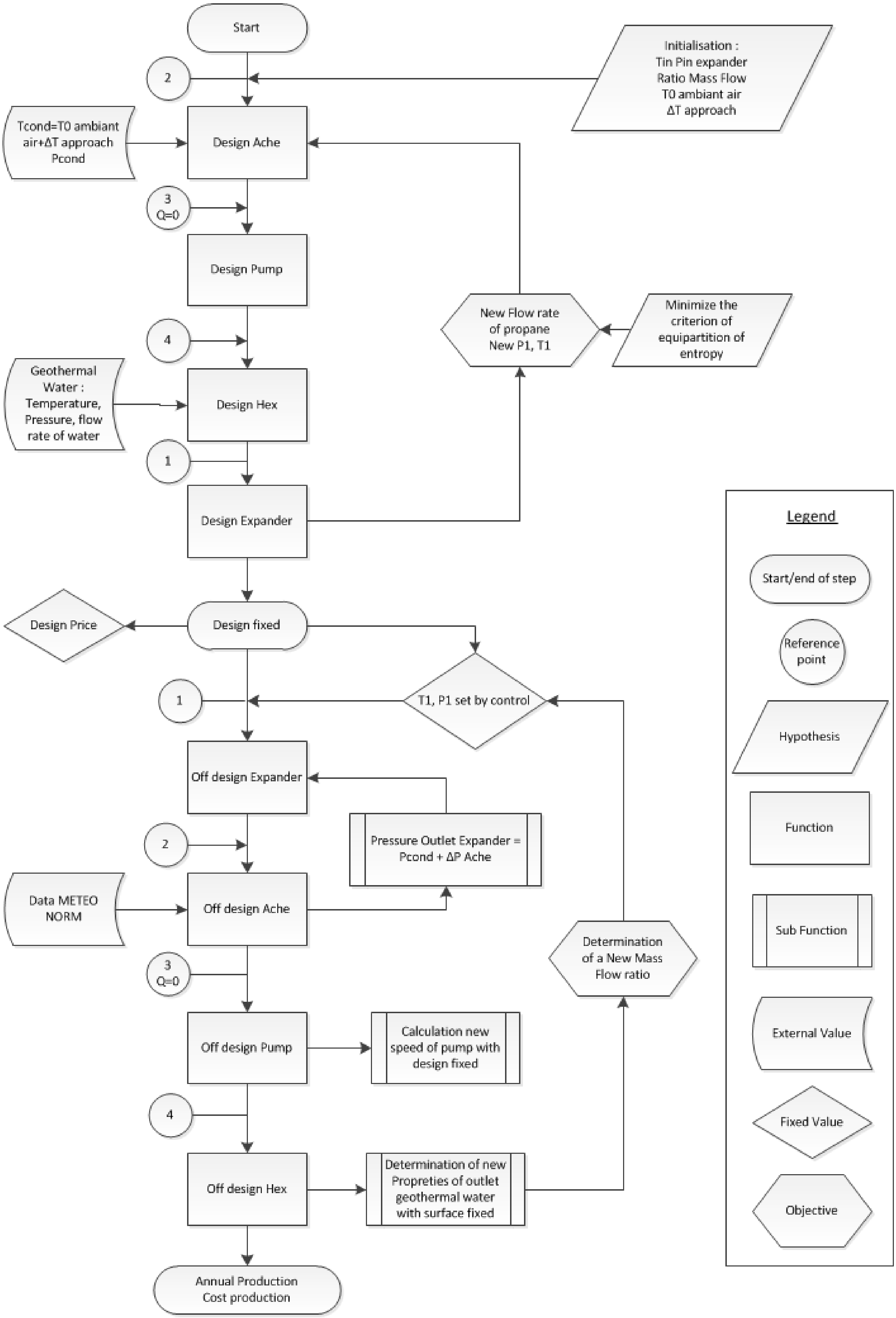

Figure 4: Algorithm logic diagram

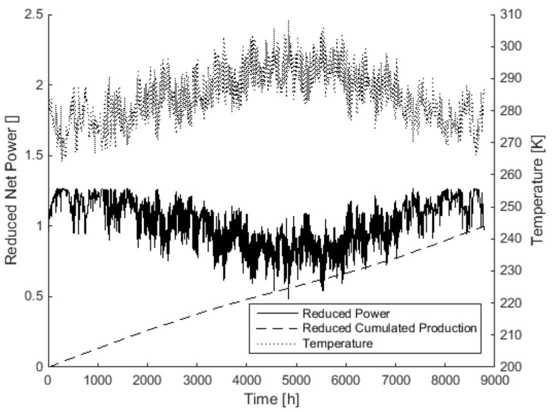

Figure 5: reduced power and production and temperature throughout the year

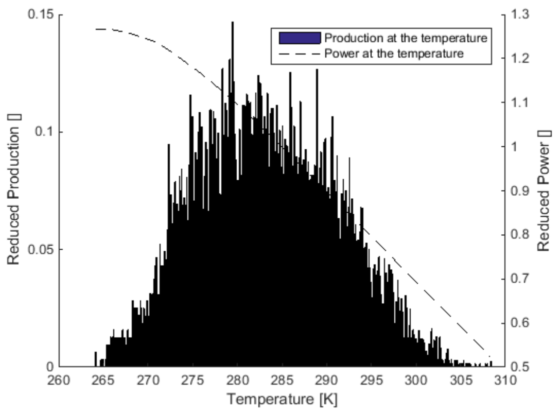

Figure 6: Reduced power and contribution to production vs ambient temperature

three parameters: turbine inlet pressure and temperature and water over propane mass flow ratio. Entropy production is illustrated by the gap between both thermal paths. On the other hand, entropy production equipartition is illustrated by the parallelism between both thermal paths. Sometimes, regarding the equipartition, it is preferable to move the thermal paths away from each other rather than bringing them together while keeping them parallel. This is clearly shown in the example of figure 8:

- The point 1 of figure 7 corresponds to the annual production of a power plant designed at 269.6 K and running at site 1. Its design case is shown with a dotted line in figure 8.
- The point 2 of figure 7 corresponds to the annual production of a power plant designed at 273.6 K and running at site 1. Its designed case is shown with a continuous line in figure 8.

The characteristics of both cases are summarized in the table 3. For only 4 K of design temperature difference, the gap between the two thermal paths of the continuous line case shows that the heat exchanger is smaller than the one of the doted line case. In consequence the heat exchanger surface of case 1 is three times bigger than that of case 2. Both annual productions are coherent regarding heat exchanger sizes. More generally, pressure rise flattens isobaric curves, while flow ratio changes the slope of the water thermal path. A temperature rise increases the length of the thermal path, thus the range of temperatures taken into account to calculate the average gap between the two fluid temperatures. Finally, by using the equipartition criterion to design equipment and if we assess the production of the designed cycle, we realize that it is not the best method to design equipment that will finally mainly work at off-design conditions. The method is missing a criterion to distinguish between plants that are designed at different ambient temperatures. Thus in paragraph 5, we add some economical criteria.

Case	Design ambient temperature	Propane reduced mass flow	Heat exchanger reduced surface	Turbine inlet temperature	Turbine inlet pressure	Reduced annual production
(Units)	(K)	( )	( )	(K)	(kPa)	( )
1	270	0.91	2.43	421	6900	1.19
2	274	1.28	0.81	408	4500	0.94

Table 3: Comparison between two different design ambient temperature

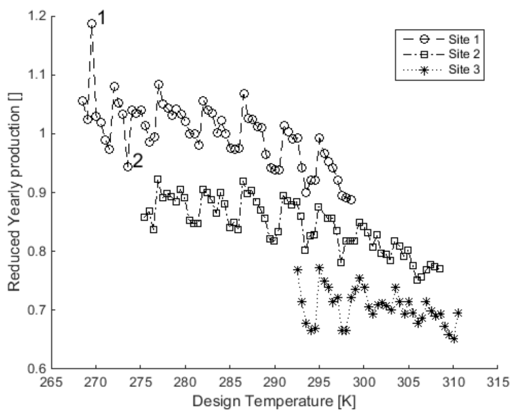

Figure 7: Reduced annual production ("Sum of net kWh produced in one year")

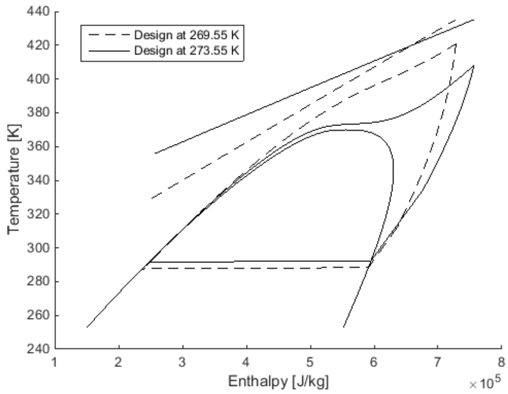

Figure 8: Thermal path of cycles after optimization for two different design ambient temperatures

5. BUSINESS PLAN

One kind of business plan is now implemented. The prices of the four main items of equipment are evaluated by means of Marshall and Swift Index (Turton et al., 2008), then shifted to match some MW ORC scale. From the sum of these items of equipment, by a Lang method (Lang, 1947), updated by Peters et al. (2003), the rest of the cost is estimated by adding rate of this sum. Of course, the business plan depends very sensitively to the place where the plant is built, the local fee, and the price for the kWh sold. The average ambient temperature of site 1 is taken as reference. The results for the other case are then reduced by this reference. The reduced production cost of the reference is then equal to 1, the sames goes for its production at the end of one year. As previously done for the calculation of yearly production in paragraph 4, the cost of production is also estimated at different design temperatures. The results are shown in figure 9. This business plan is done for a period of twenty years. The life time of the power plant is assumed to be equal to this duration. The cost of production (see equation 2) is the sum of all cost of charges, salaries and fees to be payed during one year in the power plant. One twentieth of the plant cost is added as depreciation. This amount is divided by the number of kWh produced in one year.

$$Cost_{Production} = \frac{\sum_{year}(fees,\ charges,\ salaries) + \frac{1}{20}(capital\ investment)}{\sum_{year}(sold\ electricity)} \tag{2}$$

6. FILTER APPLIED TO DESIGN TEMPERATURE REGARDING BUSINESS PLAN

As shown in figure 7, the yearly production is non monotonous with regards to the design temperature when entropy production equipartition is the optimization criterion chosen. However, once the design is done, off-design is regular and follows the ambient temperature. The business plan brings a new criterion for the evaluation of the results obtained at different design temperatures. Especially the cost of production reveals some more interesting design temperature. We decide to focus on the low production cost plants and the ambient temperature at which they have been designed, as shown in figure 9. The filter keeps all local minima [5]. We plot them with a continuous line for each site. These continuous line reveal more monotonous shapes, with global minima. These minima appear at average minus 1.5 times the standard deviation for site 1 and 2, and the average minus 2 times the standard deviation for site 3. A best design temperature found below the average temperature is coherent with figure 6 data [6].

[5] the filtered points correspond to the lower bound of the points cloud

[6] The ambient air distribution is symmetric around average, but power is higher at below average temperature

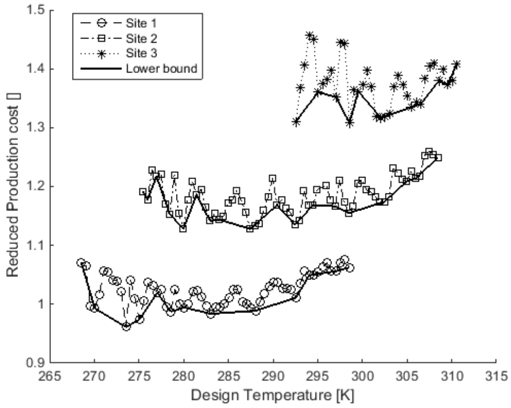

Figure 9: Reduced cost of production ("production cost of kWh")

7. CONCLUSION

Focusing production and comparing between sites, as expected, the coldest site reaches the best result. The entropy production equipartition, used only for design does not have regular trend. Despite this non monotonous characters, it is possible to extract a global regular behavior by filtering data. Due to the sensitivity of the equipartition criterion, it remains necessary to perform the calculations on the entire range of ambient temperature. Each climate has its own optimum. However, based on these three tests, no generic law (rule of thumb) seems to be able to predict a priori the optimal design temperature. Thus, the use of such numerical simulator, coupled with an optimization method seems to be essential in the context of the design of plant like ORC system. The results presented here are closely linked to the choice of the steam generator design criterion, but taking into account other constraints or criteria is easily implementable in the simulator. Thus, in forthcoming publication, we intend as well to test the following points:

- Use a particle swing optimization method to determine the best power plant design.
- Do a systematic calculation on the neighborhood of the assumed optimum found in paragraph 6.
- Use the criterion of entropy production equipartition in off-design control setting [7].
- Add a constrain to the equipartition criterion to force the mean to reach a target value, figuring indirectly the size of the heat exchanger.
- Calculate the best approach[8] on the condenser regarding the cost of production.

The combination of multiple criteria is interesting to study. Especially when thermodynamics criterion have benefit to economical optima, which finally is one of the most efficient means to deploy ORC with the support of policymaker.

NOMENCLATURE

T	temperature	(K)	Ache	condenser
T_0	temperature of cold sink	(K)	Hex	geothermal heat exchanger
Q	mass quality		**Subscript**	
P	pressure of working fluid,	(kPa)	*cond*	condensation
Δ	difference		*in*	inlet
EMTD	maximal exergy at given temperature		*hot*	hot source

[7] Actual control is at fixed turbine inlet temperature and pressure.
[8] Differential temperature between ambient air and condensation temperature.

REFERENCES

Augustine, C., Field, R., Dipippo, R., Gigliucci, G., Fastelli, I., and Tester, J. (2009). Modeling and analysis of sub- and supercritical binary rankine cycles for low- to mid-temperature. In *Geothermal resource council*, volume 33, pages 689--693.

Balje, O. (1981). *Turbomachines: A guide to design selection and theory*. Wiley-Interscience.

Lang, H. (1947). Cost relationships in preliminary cost estimation. *Chemical Engineering*.

Lemon, E. (2013). *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties (REFPROP),Version 9.1*.

Marcuccilli, F. (2007). Benefits of using radial inflow turbines for organic rankine cycles. In *Proceeding of World Geothermal congress*. World Geothermal Congress, Geothermal Resource Council.

Marcuccilli, F. (2010). Optimizing binary cycles thanks to radial inflow turbines. In *Proceeding of World Geothermal congress*. Proceedings European Geothermal Congress.

Nir, A. (1991). Heat transfer and friction factor correlations for crossflow over staggered finned tube banks. *Heat Transfer Engineering*, 12(1):43--58.

Peters, M., Timmerhaus, K., and West, R. (2003). *Plant Design and Economics for Chemical Engineers*. Companies, The Mcgraw-hill, 5th edition.

Remund, J. and Kunz, S. (1997). *METEONORM: Global meteorological database for solar energy and applied climatology*. Meteotest.

Sauret, E. (2011). Candidate radial-inflow turbines and high-density working fluids for geothermal power systems. *Energy*, 36-7:4460--4467.

Schuller, S. (2011). Best exergy point for orc. In *Proceeding of European Geothermal congress*, volume 35, pages 1343--1349. European Geothermal Congress.

Schuller, S., Josset, C., Auvity, B., and Bellettre, J. (2014). Optimisation par le critère d'équipartition de production d'entropie d'un cycle orc supercritique équipant une source d'eau chaude géothermale et influence sur les performances.

Schuster, A., Karellas, S., and Aumann, R. (2010). Efficiency optimization potential in supercritical Organic Rankine Cycles. *Energy*, 35(2):1033--1039.

Tondeur, D. (2006). Optimisation thermodynamique Équipartition de production d'entropie. In *Thermodynamique et Énergétique*, number 8017 in BE, pages 1--15. Techniques de l'ingénieur.

Troskolanski, A. (1977). *Les turbopompes*. Eyrolles.

Turton, R., Bailie, R., Whiting, W., and Shaeiwitz, J. (2008). *Analysis, synthesis and design of chemical processes*. Pearson Education.

Walraven, D., Laenen, B., and D'haeseleer, W. (2014). Optimum configuration of shell-and-tube heat exchangers for the use in low-temperature organic rankine cycles. *Energy Conversion and Management*, 83:177--187.