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Exergetic comparison of two different cooling technologies for the power cycle of a thermal power plant

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ABSTRACT

Exergetic analysis is without any doubt a powerful tool for developing, evaluating and improving an energy conversion system. In the present paper, two different cooling technologies for the power cycle of a 50 MWe solar thermal power plant are compared from the exergetic viewpoint. The Rankine cycle design is a conventional, single reheat design with five closed and one open extraction feedwater heaters. The software package GateCycle is used for the thermodynamic simulation of the Rankine cycle model. The first design configuration uses a cooling tower while the second configuration uses an air cooled condenser. With this exergy analysis we identify the location, magnitude and the sources or thermodynamic inefficiencies in this thermal system. This information is very useful for improving the overall efficiency of the power system and for comparing the performance of both technologies.

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1. Introduction

As interest for clean renewable electric power technologies grows, a number of parabolic trough power plants of various configurations are being considered for deployment around the globe. The first parabolic trough power plant in Europe, Andasol-1, in southern Spain, went into operation in November 2008 and Andasol-2 and Andasol-3 are currently under construction.

Each Andasol power plant consists of a solar field, a thermal storage tank and a conventional power plant section. The power cycle used in the Andasol plants is a traditional Rankine cycle. Induced draft cooling towers are used as condenser cooling technology. The principal heat transfer process in a wet cooling tower is evaporation. As a result, approximately 1 kg of water must be evaporated for each kilogram of steam condensed. Therefore water consumption can be significant. For example: an 80 MWe parabolic trough solar plant, operating with a capacity factor of 27%, will consume about 725 tons of water per year [1]. For sites which have a limited supply of water, water consumption adversely impacts the operating costs of the plant.

There are alternative means for condensing steam that do not require makeup water. An A-frame air cooled condenser, for example, condenses steam through several finned tubes with forced air convection on the outer surfaces of the tubes. The primary advantage of air cooled condensing is the elimination of water consumption for cooling water makeup. Another advantage is the elimination of the cooling tower plume. Elimination of the cooling tower plume presents a unique benefit at solar thermal power plants, as condensation from the cooling tower plume can reduce the optical efficiency of the solar collector mirrors closest to the cooling tower. The primary disadvantage of air cooled condensing is that heat transfer by forced air convection is a less effective heat transfer process than evaporative heat transfer. Therefore larger heat exchanger areas and greater fan power will be required to achieve heat rejection from the cycle comparable to the design state.

The thermodynamic inefficiencies associated with an energy conversion system are assessed with the aid of an exergy analysis conducted at the component level [2,3]. The exergy analysis reveals two things: the destruction of exergy within a system component, and the exergetic efficiency, which in turn shows how effectively the exergetic resources supplied to a component have been used.

Several previous exergy studies have evaluated the performance of thermal power plants. Sengupta et al. [4] conducted an exergy analysis of a 210 thermal power plant. Habib and Zubair [5] performed a second law analysis of regenerative Rankine power plants

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Nomenclature

$\dot{E}$	exergy flow rate [kW]
$\dot{m}$	mass flow rate [kg/s]
p	pressure [bar]
T	temperature [°C]
$\dot{W}$	electric power [MW]
y	exergy destruction ratio [%]

Greek letters

ε	exergetic efficiency
η	energetic efficiency [%]

Subscripts

D	destruction
F	fuel
j	jth stream
k	kth component
P	product
L	loss
0	environment

Superscripts

CH	chemical
PH	physical
TOT	total

Abbreviations

ACC	air cooled condenser
CND	condenser
CT	cooling tower
DA	deaerator
ECON	preheater
EVAP	steam generator
FWHT	feedwater heater
HTF	heat transfer fluid
SPHT	superheater
ST	steam turbine

with reheating. Dincer and Muslim [6] conducted a thermodynamic analysis of reheat cycle power plants. Tsatsaronis and Winhold [7] presented a formulation on exergoeconomic analysis and evaluation of energy conversion plants applied to a Coal-Fired Steam Power Plant. In Ref. [8] exergetic and thermoeconomic analyses for a 500-MW combined cycle plant were performed. More recently, Aljundi [9] presented an energy and exergy analysis of a steam power plant in Jordan. Related to solar thermal power plants, Singh et al [10] presented a second law analysis based on an exergy concept for a solar thermal power system. Singh et al evaluated the respective losses as well as exergetic efficiency for typical solar thermal power systems under given operating conditions. They found that the main energy loss takes place in the condenser of the heat engine, and their exergy analysis shows that the collector–receiver assembly is the part where the losses are maximum. Gupta and Kaushik [11] carried out the energy and exergy analysis for the different components of a proposed conceptual direct steam generation solar thermal power plant. In Ref. [12], a 35 MW solar thermal power plant was analyzed with the aid of exergoeconomics.

This paper deals with the comparison of wet and dry cooling technologies for the power cycle of Andasol-1 by means of exergy analysis. The solar field is not considered in the study. Through an exergy analysis, the real thermodynamic inefficiencies (exergy destruction and exergy loss) of the power cycle are identified. This information, which cannot be provided by other means (e.g. an energy analysis), is very useful for improving the overall efficiency of the power system or for comparing the performance of both cooling technologies. The results obtained here are expected to provide information that will assist in decision-making regarding alternative cooling technologies.

2. Description of the plant

The power plant has a net power capacity of 50 MWe. The cycle is a conventional, single reheat design with five closed and one open extraction feedwater heaters. The GateCycle flow diagram is shown in Fig. 1.

In direct operation mode, a heat transfer fluid (HTF, Therminol-VP1) is circulated through the solar field to the steam generation system, where steam is produced at a temperature of 373 °C and at a pressure of 100 bar. The HTF fluid acts as the heat transfer medium between the solar field and the power block; it is heated up in the solar collectors and cooled down while producing steam in the steam generator. The steam generation system consists of two parallel heat exchanger trains (preheater (ECON1)/steam generator (EVAP1)/superheater (SPHT1)) and two reheaters (SPHT2), again connected in parallel. The superheated steam travels first through the high pressure turbine (ST1), where it expands and propels the turbine blades. One extraction is taken from the high pressure turbine to preheat feedwater in one closed feedwater heater (FWH5). On exiting the high pressure turbine, the steam is directed through a reheater, where it is superheated to approximately the same temperature reached at the outlet of the superheater (373 °C) and at a pressure of about 16.5 bar. The superheated stream then passes through the low pressure steam (ST2–3), where again the steam expands and propels the turbine blades. Five steam extractions are taken from the low pressure turbine: one is directed to the deaerator (DA1) and the remaining four are fed to feedwater heaters (FWH1–4). The steam leaving the low pressure turbine, at 0.063 bar, is condensed in a surface condenser by heat exchange with circulating water. The condenser water is cooled using an induced draft cooling tower. The condensed steam (feedwater) is pumped to a sufficiently high pressure (8.38 bar) to allow it to pass through the three low pressure feedwater heaters and into the deaerator. The feedwater is pumped again at the outlet of the deaerator to a pressure slightly higher than the boiling pressure in the steam generator (103 bar). Feedwater passes through the two high pressure feedwater heaters before returning to the preheater to complete the cycle.

3. Thermodynamic evaluation

3.1. Simulation and modelling

The software package GateCycle 5.61 [13] was used for the thermodynamic simulation of the Rankine Cycle. Table 1 gives an overview on the main parameters and assumptions used in the thermodynamic simulation. Main plant operation data (detailed in Section 2) were fed to the software as input variables. The results of the simulation were compared and validated using simultaneously plant operation data and EES Thermodynamic software [14].

The thermodynamic properties were calculated based on: IAPWS IF97 Steam Tables [15] for water, JANAF Tables [16] for ambient air and NIST Tables [17] for Therminol-VP1 streams.

The power cycle is modelled assuming that all components are adiabatic, except the steam generator system, and operating at

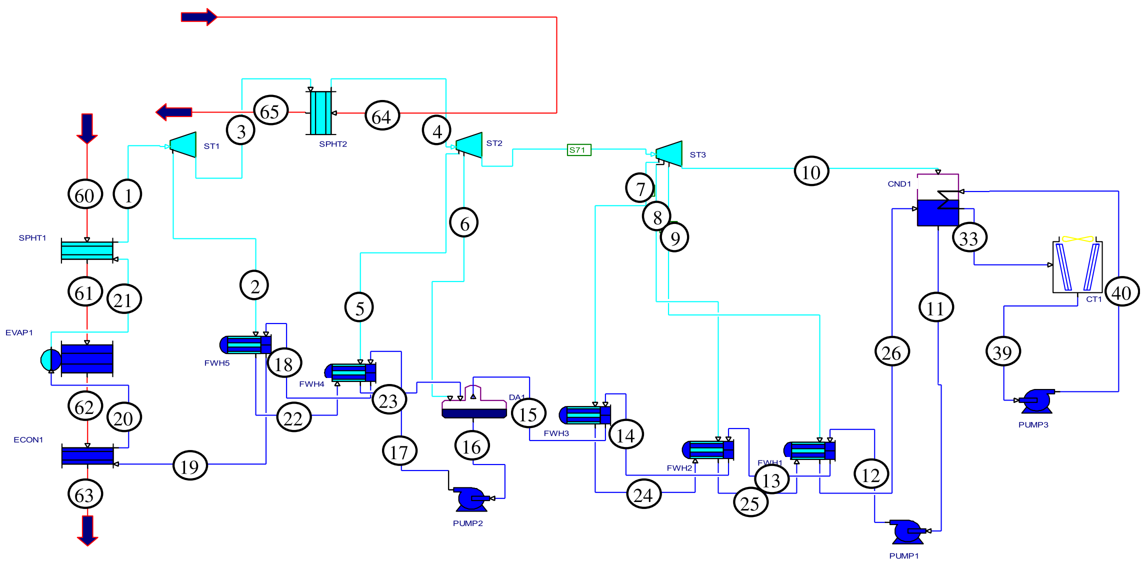

Fig. 1. GateCycle flow diagram 50 MWe Rankine cycle with wet heat rejection.

steady state. Heat loss in the steam generator is calculated from plant operation data. Changes in potential and kinetic energy of fluid streams are assumed negligible. Gland steam production, as well as steam losses through line leaks, is neglected. Pressure losses in the steam lines to the feedwater heaters were set to zero. Also, negligible changes in fluid state between the outlet of one component and the inlet of the next are assumed. In addition, the following assumptions are made: feedwater exits the preheater as saturated liquid ($x = 0$); steam exits the steam generator as saturated vapour ($x = 1$); condensed steam exits the heater as saturated liquid ($x = 0$); feedwater exits the deaerator saturated liquid ($x = 0$); feedwater exits the condenser as saturated liquid ($x = 0$).

3.2. Evaluation of the condensing operation pressure

For the overall power process, the energetic efficiency is defined by:

$$\eta = \frac{\dot{W}_{net}}{m_{60}(h_{63} - h_{60}) + m_{64}(h_{65} - h_{64})} \quad (1)$$

where $\dot{W}_{net}$ includes the auxiliary elements of the cooling system studied in each case (refrigeration pump + cooling tower or air cooled condenser).

Table 1
Main parameters and assumptions used in the thermodynamic simulation.

Isoentropic efficiency of the high pressure steam turbine	0.852
Isoentropic efficiency of the low pressure steam turbine	0.85
Isoentropic efficiency of the pumps	0.75
First extraction line inlet pressure (bar)	33.5
Second extraction line inlet pressure (bar)	14
Third extraction line inlet pressure (bar)	6.18
Fourth extraction line inlet pressure (bar)	3.04
Fifth extraction line inlet pressure (bar)	1.17
Sixth extraction line inlet pressure (bar)	0.37
Terminal temperature difference in feedwater heaters (°C)	4
Drain cooler approach in feedwater heaters (°C)	5
Temperature of the thermodynamic environment (°C)	25
Pressure of the thermodynamic environment (bar)	1
Dry bulb temperature of ambient air (°C)	20
Relative humidity of ambient air (%)	60
Cooling tower approach (°C)	6.8
Cooling tower range	9.7

The efficiency of a Rankine cycle is defined, in large part, by the pressure and the temperature of the steam both entering and leaving the turbine. The steam conditions at the turbine outlet are defined by the temperature at which the steam is condensed and the latent heat of vaporization can be transferred to the environment. The lowest ambient temperature available is the wet bulb temperature; thus, most power plants use an evaporation process to provide the cooling water source for the condenser. For sites which have a limited supply of water, heat can be rejected to the environment by condensing turbine exhaust steam at the dry bulb, rather than the wet bulb, temperature. However, heat transfer by forced air convention is less effective than evaporative heat transfer; therefore, larger heat exchanger areas and greater fan power will be required to achieve the same heat rejection.

For the Andasol-1 design turbine exit pressure (0.063 bar), energetic efficiency decreases from 34.2% to 32% when an air cooled condenser instead of a cooling tower is used. A configuration with an air cooled condenser, even if it were possible to condense steam in the air cooled condenser at the ambient air temperature, would be inefficient. The increase in gross power output from the cycle would be outweighed by the increase in parasitic fan power required to reach this temperature. In this paper, a comparison between dry and wet cooling technologies from an exergetic viewpoint is carried out with the consideration that both technologies, operating with similar parameters, not only reject the same amount of heat but also have the same net cycle power. It is necessary, therefore, to determine the turbine exit pressure with which both configurations produce an as close as possible net cycle power. Fig. 2 shows the tendency of net cycle power with the increase of condensing pressure using both technologies. As condensing pressure increases, net cycle power of both technologies get closer. A condensing pressure of 0.2 bar and, consequently, a net cycle power of 45 MWe was chosen to perform the comparison. Higher condensing pressures only cause unnecessary decrease in energetic efficiency. The condensing pressure of 0.2 bar is neither a real operation pressure nor an optimum operation pressure proposal; it is only used for simulation and comparison purposes.

This study does not include optimization of the design and configuration of the air cooled condensing unit. The performance characteristics of an existing A-frame air cooled condensing unit [13] were used as reference. The thermodynamic properties of the

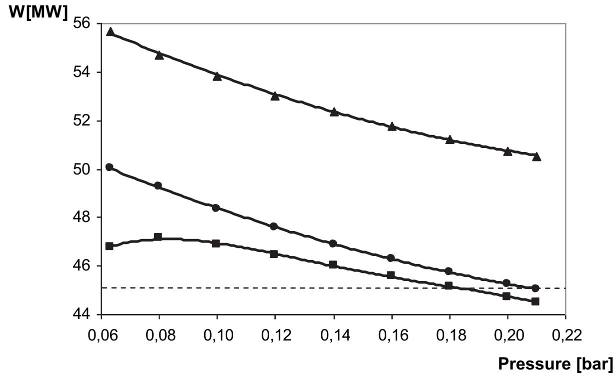

Fig. 2. Gross cycle power (—▲—), net cycle power with cooling tower (—●—) and net cycle power with air cooled condenser (—■—), versus condensing pressure (----- 45 MW).

working fluid used in the frame air cooled condensing unit were obtained on the basis of data provided by the GateCycle software.

4. Exergy analysis

In an exergy analysis, an exergy balance is formulated for the kth component at steady state conditions:

$$\dot{E}_{\mathrm{F},k} = \dot{E}_{\mathrm{P},k} + \dot{E}_{\mathrm{D},k} + \dot{E}_{\mathrm{L},k} \tag{2}$$

Here it is assumed that the system boundaries used for all exergy balances are at temperature T_0 of the reference environment and thus there are no exergy losses associated with one component [18]. Therefore the exergy destruction in the kth component is calculated only through the fuel and the product for the component. Exergy losses appear only at the level of the overall system, for which the exergy balance becomes:

$$\dot{E}_{\mathrm{F,tot}} = \dot{E}_{\mathrm{P,tot}} + \sum_k \dot{E}_{\mathrm{D},k} + \dot{E}_{\mathrm{L,tot}} \tag{3}$$

The exergetic efficiency of the kth component is:

$$\varepsilon = \frac{\dot{E}_{\mathrm{P},k}}{\dot{E}_{\mathrm{F},k}} = 1 - \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F},k}} \tag{4}$$

In addition to the exergy destruction rate, $\dot{E}_{\mathrm{D},k}$, and the exergetic efficiency, the thermodynamic evaluation of a system component is based on the exergy destruction ratio, $y_{\mathrm{D},k}$, which compares the exergy destruction in the kth component with the fuel exergy supplied to the overall system, $\dot{E}_{\mathrm{F,tot}}$:

$$y_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{F,tot}}} \tag{5}$$

This ratio expresses the percentage of the decrease in the overall system efficiency due to the exergy destruction in the kth system component:

$$\varepsilon_{\mathrm{tot}} = \frac{\dot{E}_{\mathrm{P,tot}}}{\dot{E}_{\mathrm{F,tot}}} = 1 - \sum_k y_{\mathrm{D},k} - \frac{\dot{E}_{\mathrm{L,tot}}}{\dot{E}_{\mathrm{F},k}} \tag{6}$$

Alternatively, the component exergy destruction rate can be compared with the total exergy destruction rate within the system, $\dot{E}_{\mathrm{D,tot}}$, giving the ratio:

$$y^*_{\mathrm{D},k} = \frac{\dot{E}_{\mathrm{D},k}}{\dot{E}_{\mathrm{D,tot}}} \tag{7}$$

$\dot{E}_{\mathrm{D},k}$ is an absolute measure of the inefficiencies in the kth component whereas ε_k, $y_{\mathrm{D},k}$ and $y^*_{\mathrm{D},k}$ are relative measures of the same inefficiencies. In ε_k the exergy destruction within a component is related to the fuel for the same component whereas in $y_{\mathrm{D},k}$ the exergy destruction within a component is related to the fuel for the overall system. In $y^*_{\mathrm{D},k}$ the exergy destruction within a component is related to the exergy destruction in the overall system.

The characterization of fuel and product for a component is arbitrary but of capital importance in order to give an appropriate definition of the exergetic efficiency. The product is determined by considering the desired result produced by the component and fuel by the resources expended to generate the result. In this study, fuel and product were calculated considering physical and chemical exergies of the material stream separately based on the definitions given in [18].

In the formulation of fuel and product for the cooling tower (Fig. 3), it was necessary to split the physical and the chemical exergy of the stream of wet air in the respective parameters of the components: dry air (N_2, O_2, CO_2 and Ar) and H_2O:

$$\begin{aligned} \dot{E}_{\mathrm{F,CT}} &= \dot{W}_{\mathrm{FAN}} + \left(\dot{E}_{33} - \dot{E}_{39}\right) + \left(\dot{E}_{37} - \dot{E}_{38}\right) \\ &+ \dot{E}_{35} - \dot{E}_{34} + \left(\dot{E}^{\mathrm{PH}}_{\mathrm{air},50} - \dot{E}^{\mathrm{PH}}_{\mathrm{air},51}\right) + \left(\dot{E}^{\mathrm{CH}}_{\mathrm{H_2O},50} - \dot{E}^{\mathrm{CH}}_{\mathrm{H_2O},51}\right) \end{aligned} \tag{8}$$

$$\dot{E}_{\mathrm{P,CT}} = \left(\dot{E}^{\mathrm{CH}}_{\mathrm{air},51} - \dot{E}^{\mathrm{CH}}_{\mathrm{air},50}\right) + \left(\dot{E}^{\mathrm{PH}}_{\mathrm{H_2O},51} - \dot{E}^{\mathrm{PH}}_{\mathrm{H_2O},50}\right) \tag{9}$$

Herein, the exergies of the flow streams (Tables 2 and 3) were calculated according to the definitions given in [19]. For the specific physical exergy:

$$e^{\mathrm{PH}} = (h - h_0) - T_0(s - s_0) \tag{10}$$

where subscript 0 represents environmental conditions. The physical exergy was obtained based on the thermodynamic properties calculated by GateCycle [13]. The values of the standard chemical exergy of chemical compounds were taken from [20].

The results of the thermodynamic analysis are shown in Tables 2 and 3.

5. Results and discussion

Tables 4 and 5 show the exergy destruction, exergetic efficiency and exergy destruction ratios for each main plant component.

The results are presented in descending order of the parameter $y^*_{\mathrm{D},k}$, which compares the exergy destruction in the kth component

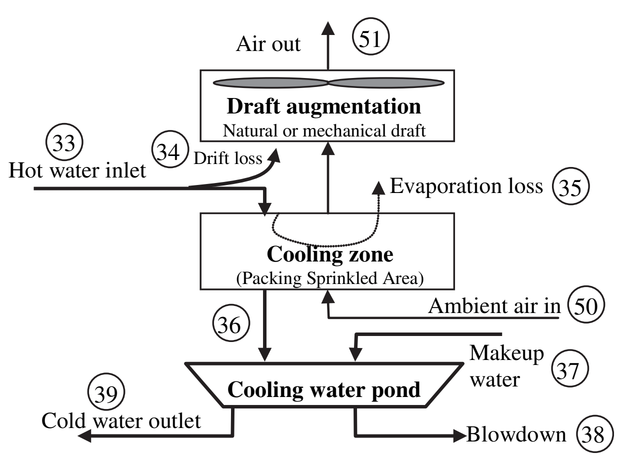

Fig. 3. GateCycle schematic of a cooling tower.

Table 2
Thermodynamic and exergetic data of streams for 0.063 bar of condensing pressure.

	$\dot{m}$ (kg/s)	T (C)	p (bar)	$\dot{E}_j^{PH}$ (kW)	$\dot{E}_j^{CH}$ (kW)	$\dot{E}_j^{TOT}$ (kW)
1	60.08	373.00	100.00	72,210	120	72,330
2	6.25	241.46	33.48	6123	12	6135
3	53.83	208.48	18.50	46,913	108	47,021
4	53.83	373.40	16.50	57,811	108	57,919
5	2.70	352.12	13.99	2778	5	2784
6	2.71	254.59	6.18	2248	5	2253
7	2.69	186.44	3.04	1839	5	1845
8	2.80	108.24	1.17	1433	6	1438
9	1.96	73.99	0.37	651.4	4	655
10	40.96	37.05	0.06	3551	82	3633
11	48.42	37.05	0.06	44.59	97	141
12	48.42	37.15	8.38	85.74	97	182
13	48.42	62.69	8.28	484	97	581
14	48.42	97.51	8.18	1585	97	1682
15	48.42	128.84	8.08	3051	97	3147
16	60.08	159.99	6.18	6073	120	6193
17	60.08	161.71	103.00	6759	120	6879
18	60.08	189.68	102.50	9213	120	9333
19	60.08	234.85	102.00	13,941	120	14,062
20	60.08	309.00	101.20	24,342	120	24,462
21	60.08	313.00	101.00	63,737	120	63,857
22	6.25	194.68	33.38	969.8	12	982
23	8.95	166.71	13.89	994	18	1012
24	2.69	102.51	2.94	98.26	5	104
25	5.50	67.69	1.07	64.71	11	76
26	7.46	42.15	0.27	14.52	15	29
60	533.40	393.00	15.80	149,968	19,854,633	20,004,601
61	533.40	381.00	14.20	140,706	19,854,633	19,995,339
62	533.40	320.00	12.60	98,383	19,854,633	19,953,017
63	533.40	301.00	11.80	86,687	19,854,633	19,941,320
64	68.50	393.00	15.80	19,256	2,549,761	2,569,016
65	68.50	225.00	13.50	6050	2,549,761	2,555,811
Cooling tower						
33	2869.22	36.23	1.88	2790	5734	8524
34	0.06	36.25	0.95	0.05061	0.115	0.165
35	32.86	32.85	0.05	2002	66	2068
36	2836.19	28.83	0.95	293.4	5668	5961
37	65.73	27.00	1.00	2.024	131	133
38	31.81	28.79	0.95	3.22	64	67
39	2869.20	28.79	0.95	290.5	5734	6024
40	2869.20	28.80	1.88	560.2	5734	6294
50	1791.17	28.05	0.95	29	12,384	12,413
51	1824.04	32.85	0.95	4184	12,281	16,465
Air cooled condenser						
50	13,989	28.05	0.95	224	96,730	96,953
51	13,989	34.58	0.95	2176	96,730	98,906

with the exergy destruction in the overall process, $\dot{E}_{D,tot}$. As result, this parameter provides an insight into the exergetic meaning of each of the components of the overall system.

Exergy destruction ratio in condenser working at 0.063 bar and using a cooling tower represents only 6.8% of total exergy destruction whereas with an air cooled condenser it represents 25.5% of total exergy destruction. This result is mainly due to the huge amount of parasitic fan power required by the air cooled condenser to work under this pressure condition. Working at 0.2 bar, both components (wet and dry condenser) present a comparable and high value of exergy destruction: 32.73 % the wet condenser and 31.11 % the dry condenser. These values represent almost a third of the total exergy destroyed in the global system.

Taking into account all the components that integrate the wet cooling technology (condenser, cooling tower and pump) it is necessary to sum the exergy destroyed in all the components. In this case, the exergy destructed ratio amounts to almost 12% operating at 0.063 bar and 37.2% operating at 0.2 bar. As a result we can say that, from an exergetic point of view, the use of an air cooled condenser is not an efficient solution working at low exit turbine pressures but it becomes more competitive at higher pressures.

Exergy destructed in remaining components is not especially influenced by the use of different cooling technologies, apart from the condenser working pressure. The sequence followed by different components regarding the value of $y^*_{D,k}$ is similar in both operation alternatives.

Exergetic efficiency of each component assesses the fuel exergy wasted in the component as exergy destruction. Related to this concept, Tables 4 and 5 show the following:

- The exergetic efficiency of the cycle is similar using both Technologies and working with the two pressure operation alternatives in the condenser: approximately 70 percent.
- Each component presents also a similar value of the exergetic efficiency with both technologies and working under both condenser pressure conditions: 0.063–0.2 bar. The most important difference is presented in condenser: working at 0.063 bar the exergetic efficiency of the wet condenser is 63.3% whereas the air cooled condenser presents an exergetic efficiency of 25.8%. Thermodynamic inefficiencies associated with heat transfer are caused by mismatched heat capacity rates of the two streams and by a finite minimum temperature

Table 3
Thermodynamic and exergetic data of streams for 0.2 bar of condensing pressure.

	$\dot{m}$ (kg/s)	T (C)	p (bar)	$\dot{E}_j^{PH}$ (kW)	$\dot{E}_j^{CH}$ (kW)	$\dot{E}_j^{TOT}$ (kW)
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4	53.83	373.40	16.50	57,811	108	57,919
5	2.70	352.12	13.99	2778	5	2784
6	2.71	254.59	6.18	2248	5	2253
7	2.70	185.81	3.04	1839	5	1844
8	2.81	106.83	1.17	1432	6	1438
9	0.20	73.99	0.37	66.8	0.4	67.2
10	42.71	60.06	0.20	10,282	85	10,368
11	48.42	60.06	0.20	387	97	483
12	48.42	60.17	8.38	429	97	525
13	48.42	62.69	8.28	484	97	581
14	48.42	97.51	8.18	1585	97	1682
15	48.42	128.84	8.08	3051	97	3147
16	60.08	159.99	6.18	6073	120	6193
17	60.08	161.71	103.00	6759	120	6879
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21	60.08	313.00	101.00	63,737	120	63,857
22	6.25	194.68	33.38	970	12	982
23	8.95	166.71	13.89	994	18	1012
24	2.70	102.51	2.94	98	5	104
25	5.50	67.69	1.07	65	11	76
26	5.70	65.17	0.27	59	11	71
60	533.40	393.00	15.80	149,968	19,854,633	20,004,601
61	533.40	381.00	14.20	140,706	19,854,633	19,995,339
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63	533.40	301.00	11.80	86,687	19,854,633	19,941,320
64	68.50	393.00	15.80	19,256	2,549,761	2,569,016
65	68.50	225.00	13.50	6050	2,549,761	2,555,811
Cooling tower						
33	2254.50	38.78	1.88	3171	4505	7676
34	0.05	38.80	0.95	0.07	0.10	0.17
35	34.02	35.58	0.06	2786	68	2854
36	2220.41	28.83	0.95	230	4437	4667
37	65.73	27.00	1.00	2	131	133
38	33.98	28.78	0.95	3	68	71
39	2254.54	28.78	0.95	227	4505	4732
40	2254.50	28.78	1.88	438	4505	4943
50	1407.47	28.05	0.95	23	9732	9754
51	1441.49	35.58	0.95	4987	9648	14,635
Air cooled condenser						
50	4075.99	28.05	0.95	65	28,184	28,249
51	4075.99	51.02	0.95	4518	28,184	32,701

difference of ΔT_{min}. In an air cooled condenser, the mismatch between the heat capacity rates of the two streams is high and these inefficiencies are difficult to avoid. Working at 0.2 bar, the exergetic efficiency of the air condenser improves, even while remaining a low value: 40%, and the exergetic efficiency of the wet condenser becomes considerably worse (27.5%) due to the increase in the temperature difference between inlet and outlet streams and to the increase in refrigeration water mass flow rate.

Apart from the condenser, it is interesting to add further comments about the results provided by the exergetic analysis of the rest of the components of the plant:

- Exergetic efficiency of the steam turbines is similar operating under both pressure conditions (89–90%). Pumpś exergetic efficiency is within the range of 74% for condensate pump, and 87% for feedwater pump where $p_{COND} = 0.063$ bar for both. Exergy destruction here is caused basically by friction. Therefore only an increase in the isentropic efficiency can increase its exergetic efficiency.
- Heat exchangers in the heat recovery steam generator present high exergetic efficiency values under both condenser pressure conditions: between 82%, for the economizer, and 94%, for the superheater. Herewith, irreversibilities are mainly caused by heat transfer between material streams entering and exiting the heat exchanger. A small percentage is produced by pressure loss and heat loss with the surroundings. The high exergetic efficiency values presented in Tables 4 and 5 are logical according to the similar heat capacity rates and the small value of the minimum temperature difference.
- The feedwater heaters and the deaerator present consistent values of exergetic efficiency: the lower the minimum temperature difference, the lower the exergy destruction and the higher the exergetic efficiency.

The exergy destruction ratio, y_D, gives a useful reference in comparison of components of different systems with similar fuel exergy. Regarding to this value, through Tables 4 and 5 it is possible to indicate: In the process operating with a condenser pressure of 0.063 bar, the exergy destructed in the wet condenser represents only 1.7% of the fuel of the overall system while the destruction using an air cooled condenser is higher than 7%. Operating at

Table 4
Exergy destruction, exergetic efficiency, and exergy destruction ratios for each main plant component; $p_{COND} = 0.063$ bar.

	$\dot{E}_F$(kW)	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	ε	y_D (%)	$y^*_{D,k,tower}$(%)	$y^*_{D,k,air}$(%)
LP steam turbine – ST2–3	45,311	40,023	5288	0.883	6.91	28.06	23.97
Air cooled condenser – ACC1	7583	1959	5624	0.258	7.35		25.50
Evaporator – EVAP1	42,323	39,396	2927	0.931	3.83	15.53	13.27
Reheater – SPHT2	13,206	10,898	2308	0.825	3.02	12.25	10.46
HP steam turbine – ST1	19,174	17,355	1819	0.905	2.38	9.65	8.25
Economizerr – ECON1	11,697	10,400	1297	0.889	1.70	6.88	5.88
Wet condenser – CND1	3521	2230	1291	0.633	1.69	6.85	
Cooling tower – CT1	13,169	12,095	1074	0.918	1.40	4.49	
Superheater – SPHT1	9262	8472	790	0.915	1.03	4.19	3.58
Feedwater heater 5 – FWH5	5153	4728	425	0.918	0.56	2.26	1.93
Feedwater heater 2 – FWH2	1466	1101	365	0.751	0.48	1.94	1.65
Feedwater heater 1 – FWH1	702	398	303	0.568	0.40	1.61	1.37
Feedwater heater 4 – FWH4	2754	2454	300	0.891	0.39	1.59	1.36
Feedwater heater 3 – FWH3	1741	1465	276	0.841	0.36	1.46	1.25
Deaerator – DA1	2063	1844	219	0.894	0.29	1.16	0.99
Feedwater pump – PUMP1	789	687	103	0.870	0.13	0.55	0.47
Refrigeration pump – PUMP3	316	270	46	0.854	0.06	0.24	
Condensate pump – PUMP2	56	41	14	0.741	0.02	0.08	0.07
Total with cooling tower	76,486	55,725	18,846	0.754			
Total with air cooled condenser	76,486	52,471	22,058	0.712			

Table 5
Exergy destruction. exergetic efficiency and exergy destruction ratios for each main plant component; $p_{COND} = 0.2$ bar.

	$\dot{E}_F$(kW)	$\dot{E}_P$ (kW)	$\dot{E}_D$ (kW)	ε	y_D (%)	$y^*_{D,k,tower}$(%)	$y^*_{D,k,air}$(%)
Wet condenser – CND1	9955	2733	7222	0.275	9.44	32.73	
Air cooled condenser – ACC1	11,140	4455	6685	0.400	8.74		31.11
LP steam turbine – ST2–3	39,164	34,946	4218	0.892	5.51	19.12	19.63
Evaporator – EVAP1	42,323	39,396	2927	0.931	3.83	13.26	13.62
Reheater – SPHT2	13,206	10,949	2257	0.829	2.95	10.23	10.50
HP steam turbine – ST1	19,224	17,355	1869	0.903	2.44	8.47	8.70
Economizer – ECON1	11,697	10,400	1297	0.889	1.70	5.88	6.03
Cooling tower – CT1	13,743	12,763	980	0.929	1.28	4.25	
Superheater – SPHT1	9262	8742	520	0.944	0.68	2.36	2.42
Feedwater heater 5 – FWH5	5153	4728	425	0.918	0.56	1.93	1.98
Feedwater heater 2 – FWH2	1466	1101	365	0.751	0.48	1.65	1.70
Feedwater heater 4 – FWH4	2754	2454	300	0.891	0.39	1.36	1.40
Feedwater heater 3 – FWH3	1741	1465	276	0.841	0.36	1.25	1.28
Desgasificador – DA1	2063	1844	219	0.894	0.29	0.99	1.02
Feedwater pump – PUMP1	789	687	103	0.870	0.13	0.47	0.48
Refrigeration pump – PUMP3	248	211	38	0.849	0.05	0.17	
Feedwater heater – FWH1	72	55	17	0.765	0.02	0.08	0.08
Condensate pump – PUMP2	55	42	14	0.755	0.02	0.06	0.06
Total with wet condenser	76,486	50,814	23,046	0.699			
Total with air cooled condenser	76,486	50,271	21,491	0.719			

pressure condenser of 0.2 bar, terms are reversed: wet condenser is more inefficient: it destroys 9.44% of the fuel exergy whereas in the air cooled condenser the exergy destruction ratio is only one point higher as previously. The rest of the components show no noteworthy tendencies.

6. Conclusions and future work

The paper presents the exergetic comparison of dry and wet cooling technologies for the Rankine cycle of a solar thermal power plant.

Exergy analysis is a useful tool to locate and evaluate the thermodynamic inefficiencies of the analyzed process. The third part of the inefficiencies is located in the heat rejection process. Other part is located in the steam generation process and another in the expansion process. The condenser is the component where more exergy of the total fuel exergy is wasted as exergy destruction. Related with the total fuel exergy, at the condensing pressure of 0.063 bar, the air cooled condenser destroys four times the exergy destroyed in the condenser with cooling tower. At 0.2 bar both configurations destroy a similar amount of exergy. From an exergetic point of view, the use of an air cooled condenser is not an efficient solution to working at low exit turbine pressures. It becomes, however, more competitive at higher pressures.

This paper is a part of a series of works that analyzed dry and wet cooling technologies for the Rankine cycle of a solar thermal power plant. An advanced exergetic analysis will also be conducted to evaluate the avoidable and unavoidable part of the exergy destruction as well as the mutual influence of the components of the process. An exergoeconomic analysis will also be performed to evaluate the cost of the inefficiencies and the real potential for reducing them.

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