EXERGETIC AND THERMOECONOMIC ANALYSIS OF SUPERCRITICAL CYCLE FOR COAL POWER PLANT

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Abstract. *Historically, coal-fired power stations had been limited in its efficiency by the use of the conventional Rankine cycle. However, recent technological development has made it possible to increase the efficiency of the steam cycle in these power plants. One of these technologies is the Supercritical Rankine Cycle, which has improved the thermal efficiency of these power plants moreover has reached some reduction of the atmospheric emissions without the need of additional environmental equipment. An exergetic and thermoeconomic analysis in order to evaluate the exergy destruction and the exergetic efficiencies in each component of a supercritical coal-fired power station is performed in this paper. The exergetic efficiency of the overall plant is found to be 40.4%. The thermoeconomic equations used in this paper may also be utilized in the exergoeconomic analysis in order to estimate the production costs depending on various input costs in a supercritical cycle.*

Keywords: *supercritical cycle, thermoeconomic, exergy, exergy cost, exergetic efficiency.*

1. INTRODUCTION

Due to the continuous increase in electrical energy demand in most countries of the world, the main challenge today is to increase the efficiency of power plants in order to ensure an electricity source in the future and at the same time to fulfill environmental parameters that have become more demanding recently.

Nowadays, the main energy sources used in the generation of electricity are fossil fuels, hydropower, nuclear energy and renewable resources under development; whereas coal is responsible for 40% of electric generation in the world (World Coal Institute, 2005).

At the end of 2006 the reserves of coal reported in the world were around 909,064 million tonnes, mostly found in Europe, Asia and North America. Besides, the lower coal cost and its relative stability, being compared to oil and natural gas, which has increased of over 200% and 75% in the last five years respectively, in comparison to 46% coal, make this fuel one of the most safe and attractive ways to generate electrical power (BP Statistical Review of World Energy, 2007).

However, the biggest challenge is the reduction of CO_2 emissions, which has led engineers to develop new technologies for coal power generation (Clean Coal Technologies) that can supply this energy demand, so it is becoming necessary to optimize and improve these power installations to get the maximum efficiency for equipment and minimize the fuel consumption in order to generate cheapest electrical power and less atmospheric emissions.

2. SUPERCRITICAL SYSTEM CONFIGURATION

There is a defined relationship between the operational temperature and the system optimal pressure regarding steam cycles. The supercritical pressure cycle is in general used to obtain higher thermodynamics efficiency with boilers using fossil fuels, operating in higher pressures and steam temperatures. Figure 1 shows the physical structure of a supercritical coal power plant with two stages of reheating (Suresh *et al*. 2006; Kjaer, 2006). In this plant coal and air are fed to the combustor (1) where it releases its calorific power, the gases generated exchange heat with the feedwater into the steam generator (4) resulting in the generation of supercritical steam at 29 MPa and 855 K. This steam expands itself firstly in a very-high-pressure turbine (6) at 9.4 MPa and then is reheated at 853 K before making a second expansion into the high pressure turbine (5). The steam at the outlet of the high pressure turbine (1.9 MPa / 612 K) is led to the boiler for a second stage of reheating at 853 K and is expanded in an intermediate pressure turbine (7). The steam expansion ends in two low pressure turbines (8) with the following conditions (0.0083 MPa/315.35 K) and steam exhaust quality of 0.962 (Bernero, 2002). The extraction of turbines are used in a ten stage preheating system (9-18) in order to raise the boiler feedwater temperature up to 573 K in the steam generator. The thermal energy in the flue gas leaving the steam generator is recovered in air preheaters (3). The operational conditions of the plant are summarized in Tab. 1 and the characteristics of the considered fuel are presented in Tab. 2.

2.1. Assumptions

1) The kinetic and potential energy are neglected; 2) Reference state temperature (T_0) and pressure (P_0) are 298.15 K and 101.325 kPa; 3) Temperature and pressure of fuel and air inlets are 298.15 K and 101.325 kPa; 4) Isentropic

efficiency of pumps/fans: 75%; 5) Isentropic efficiency of turbines 86% (Bernero, 2002); 6) Excess air: 20%; 7) Steady state operation; 8) Load condition 100%.

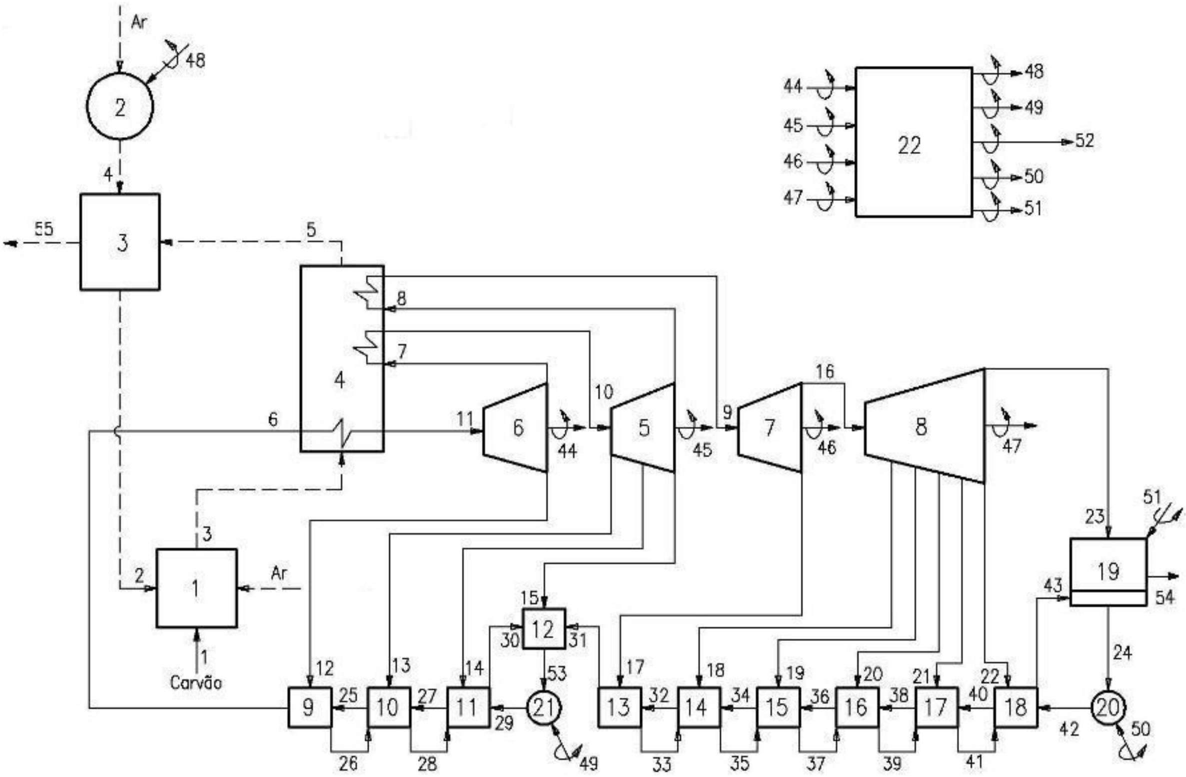

Figure 1. Physical structure of supercritical coal power station.

Table 1. Operating conditions of the power plant.

Operating condition	Value
Mass flow rate of fuel	46.285 kg/s
Mass Flow Rate of air Primary e Secondary Tertiary	 512.94 kg/s 18.31 kg/s
Stack gas temperature	416.45 K
Feed water inlet temperature to boiler	300 K
Steam flow rater	381.55 kg/s
Steam temperature	582/580/580 ^{0}C
Steam pressure	29/9.416/5.788 MPa
Pressure condenser	8.3 kPa

Table 2. Properties of Bituminous coal used in Power plant.

Ultimate analysis (dry basis)	
Carbon	71.20%
Hydrogen	4.98%
Nitrogen	1.27%
Sulfur	3.65%
Oxygen	9.30%
Ash	9.60%
Total	100.00%
Higher heating value	27386 kJ/kg
Lower heating value	26189 kJ/kg

Reference: Phyllis database for biomass and waste (2008).

3. EXERGETIC ANALYSIS

According to Aljundi (2008), the exergy is a thermodynamic function which can be defined as the maximum capacity of a system to perform useful work, once it leads to a certain final state balanced with its surroundings. The exergy (B_i) can be used as a quality and quantity measure of energy which involves the first and second thermodynamic laws, so an exergetic analysis is useful to identify, locate and quantify the thermodynamic efficiencies of a thermal system (Torres and Valero, 2001).

The calculation of exergy is related to enthalpy (h), entropy (s), temperature (T), pressure (P), composition, velocity and reference state. A process that generates electrical power can be considered as a system consisted of lots of units related to each other by material flows (m) like water, combustion gas and fuel and energy flows in heat form (Q) and shaft work (W). Equations (1-4) show some typical expressions for exergy calculation of these thermodynamic flows (Zaleta *et al. 2007*):

For pure substances (water) with negligible potential and kinetic energy changes.

$$B_i = m_i\left[\left(h_i - h_0\right) - T_0\left(s_i - s_0\right)\right] \tag{1}$$

For solid fuel based on its elemental composition (Hydrogen (x_H), Carbon (x_C), Oxygen (x_O), Nitrogen (x_N) and sulfur (x_S).

$$B^{ch} = m_i\left[\left(LHV\right)_i\left(1{,}0438 + 0{,}0013\frac{x_H}{x_C} + 0{,}1083\frac{x_O}{x_C} + 0{,}0549\frac{x_N}{x_C}\right) + 6740x_S\right] \tag{2}$$

Heat Flow

$$B_i = Q_i\left(1 - \frac{T_o}{T_i}\right) \tag{3}$$

Shaft Work

$$B_i = W_i \tag{4}$$

For exergy calculation of combustion gases, the exergy of the mixture is based on its molar flow (n_k) and is given by the Eq (5) where the subscript (k) represents each component of the gas mixture and the enthalpy (Δh - *kJ/kmol*) and entropy (Δs - *kJ/kmol.K*) variation are in function of temperature and are calculated by the Eqs. (6-7), where A, B, C and D are specific constants to each component (Carvalho *et al.* 1977).

$$B_i = \sum \; n_k\left[\left(h_i - h_0\right) - T_0\left(s_i - s_0\right)\right]_k \tag{5}$$

$$\left(h_i - h_0\right) = \left[\left(A \cdot T\right) + \left(B \cdot \frac{T^2}{1000}\right) + \left(C \cdot \frac{100000}{T}\right) + D\right] \cdot 4{,}186 \tag{6}$$

$$\left(s_i - s_0\right) = \left[\left(A \cdot \ln\frac{T}{T_0}\right) + \left(2B \cdot \frac{\left(T - To\right)}{1000}\right) + \left(50000C \cdot \left(\frac{1}{T^2} - \frac{1}{T_0^{\;2}}\right)\right)\right] \cdot 4{,}186 \tag{7}$$

The mass, energy and exergy balances, for each subsystem that compose the power generation cycle, considering a steady state and not taking into account the kinetic and potential energy changes, can be calculated as Eqs. (8-10) show, respectively (Moran and Shapiro , 2000).

$$\sum \dot{m}_e = \sum \dot{m}_s \tag{8}$$

$$\dot{Q} - .\dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_s h_s \tag{9}$$

$$\left(1 - \frac{T_0}{T}\right)\dot{Q} - \dot{W} = \sum B_e - \sum B_s + \dot{I} \tag{10}$$

4. THERMOECONOMIC AND EXERGY COST THEORY

As mentioned previously the exergy as a thermodynamic parameter reflects the minimum quantity necessary to obtain a product from a reference environment. Therefore, we can say that the exergy is independent of the process used to manufacture a product and represents a 'minimum cost' (in energetic terms) required to produce it. As all real processes are irreversible, verifying destruction or exergy loss, the exergy necessary to obtain a functional product, called "Exergy Cost" (B^*), is always going to be a function of the process used, incorporating exergy losses that belong to the process, and whichever it is, the exergy cost will always be higher than exergy (Lozano and Valero, 1987).

So, the energetic optimization of the process must be in order to minimize the value of the unitary exergetic cost (k^*), being defined as follows:

$$k^* = \frac{Exergy\ Cost}{Exergy} \tag{11}$$

For the calculation of exergetic costs (B^*) in a system, it is necessary, initially, to define the heat flow, work or material, in the control volumes, represented by solid material flows, gases or energy, defining those that act as exergy resources (R), necessary for the manufacturing of a certain product (P), by the system. In this way, the exergy contained in a product obtained in an analyzed system, is given by:

$$PRODUCT\,(P) = RESOURCES\,(R) - LOSSES\,(L) - DESTRUCTION\,(D) \tag{12}$$

In the process of power generation it is considered as resources (R): the exergy in the form of electrical energy provided for the starting of motor fans, pumps and compressors; exergy provided by fuel burning in the boilers; exergy provided by the feedwater system in the boilers and exergy provided by the steam in electricity generation. Exergy difference between the steam flow and entering water flow in the boiler and exergy as shaft work form are considered as products (P). The hot gases emitted by the chimney into the atmosphere and the flows without any recovery are considered losses (L), and in the exergetic cost calculation this value is considered null. The same thing happens to the heat losses for the environment, by the heat transfer through boilers refractory coverings and turbines isolation.

4.1. Exergy Cost Theory

The exergetic cost theory is a practical method to determine the exergetic costs (B^*) in a system with n flows (mass, heat or power) through a system resolution of (*n x n*) equations. So, once it is established in the system the flows considered as resources, products and losses in each subsystem of the process, the (*n x n)* equation system is achieved by some assumptions as follow (Torres and Valero, 2001; Zaleta *et al.* 2007):

a) Exergetic cost is a conservative property in each subsystem:

$$\sum B_R^* = \sum B_P^* \tag{13}$$

b) The exergetic cost of the initial resource is considered equal to exergetic content:

$$\sum B_R^* = \sum B_F \tag{14}$$

c) A multi-product of the same nature in a subsystem must have a exergetic equivalent unit cost, it means:

$$\frac{B_{P,a}^*}{B_{P,a}} = \frac{B_{P,b}^*}{B_{P,b}} = \frac{B_{P,c}^*}{B_{P,c}} \tag{15}$$

d) Any not-exhausted resource in a subsystem must have the unit cost of its resource or average cost in case of multi-resource, such that:

$$\frac{\sum B_R^*}{\sum B_R} = \frac{B_{R,non-ex}^*}{B_{R,non-ex}} \tag{16}$$

e) Also, the overall system losses, in turn, are assessed a null exergy cost, due to a non-external valuation:

$$\sum B_L^* = 0 \tag{17}$$

5. RESULTS AND DISCUSSION

The exergetic and thermoeconomic analysis of the cycle presented in this paper introduces the exergetic efficiency as an evaluation parameter of the real performance from the thermodynamic point of view. Once the products and the resources in the analyzed thermal system are identified, the irreversibility and exergetic costs for each subsystem are determined. Using the previous expressions Eqs. (1-10), the plant was analyzed and the properties of each thermodynamic flow were calculated and summarized in Tab. 3.

Table 3. Thermodynamic properties of supercritical cycle.

Flow	P (kPa)	T (k)	m (kg/s)	(h-ho) (kJ/kg)	(S-So) (kJ/kg.K)	b (kJ/kg)	B (kW)
1	101.3	298.15	46.285	-	-	24,999.5	1157101.86
2	112	659.25	512.94	371.4	0.784	137.6504	70606.40
3	112	2173.15	573.09	1346580.84	1362.192	940443.3	940443.30
4	115	311.85	512.94	13.8	0.009	11.11665	5702.17
5	102	692.95	573.09	250072.924	531.587	91580.26	91580.26
6	32368	573.15	381.55	1220.8	2.7983	386.4869	147464.06
7	9416	673.85	349.42	3007.2	5.8792	1254.317	438283.28
8	1900	611.95	300.86	3008.4	6.5588	1052.894	316773.62
9	1710	853.15	300.86	3542.4	7.3586	1348.433	405689.68
10	8945	853.15	349.42	3479.1	6.5362	1530.332	534728.60
11	29008	855.15	381.55	3289.3	5.821	1553.769	592840.50
12	9416	673.85	32.13	3007.2	5.8792	1254.317	40301.19
13	5788	778.15	20.87	3331.3	6.5462	1379.55	28791.22
14	3456	696.75	12.76	3173.4	6.564	1216.343	15520.54
15	1843	611.35	14.93	3008.3	6.5727	1048.649	15656.34
16	1130	789.35	288.73	3407.2	7.3844	1205.541	348075.89
17	1130	789.35	12.13	3407.2	7.3844	1205.541	14623.21
18	601	694.55	11.16	3211	7.4081	1002.275	11185.39
19	305	602.65	10.32	3024.2	7.4402	805.9044	8316.93
20	147	528.45	9.48	2878.2	7.5019	641.5085	6081.50
21	67	433.05	8.81	2692.8	7.4716	465.1425	4097.91
22	28	358.05	8.21	2550.8	7.5234	307.6983	2526.20
23	8.3	315.35	240.76	2381.1772	7.5597528	127.2369	30633.56
24	8.3	315.35	300.87	71.8	0.2347	1.824195	548.85
25	33028	540.05	381.55	1063.9	2.5151	314.0229	119815.45
26	8926	544.45	32.13	1085.5	2.611	307.0304	9864.89
27	33702	508.85	381.55	921.1	2.2417	252.7371	96431.86
28	5477	513.25	53	933.2	2.3317	238.0036	12614.19
29	34390	488.05	381.55	828.12	2.0535	215.869	82364.81
30	3203	490.85	65.75	828.13	2.1279	193.6966	12735.55
31	1177	451.75	300.87	651.91	1.7583	127.6729	38412.93
32	1214	425.65	300.87	538.34	1.5019	90.54852	27243.33
33	1062	430.05	12.13	557.25	1.5436	97.02566	1176.92
34	1251	401.05	300.87	433	1.2439	62.13122	18693.42
35	566	405.45	23.29	451.31	1.2911	66.36854	1545.72
36	1290	377.85	300.87	334.76	0.9917	39.08465	11759.40
37	289	382.35	33.62	353.04	1.0425	42.21863	1419.39
38	1330	356.05	300.87	243.14	0.742	21.9127	6592.87
39	139	360.45	43.1	260.67	0.7943	23.84946	1027.91
40	1371	335.45	300.87	156.98	0.4925	10.14113	3051.16
41	63	339.95	51.91	174.67	0.5489	11.01547	571.81
42	1413	316.05	300.87	75.99	0.2435	3.390475	1020.09
43	27	320.55	60.11	93.54	0.3031	3.170735	190.59
44	-	-	-	-	-	-	92781.59
45	-	-	-	-	-	-	133841.08
46	-	-	-	-	-	-	34981.59
47	-	-	-	-	-	-	233466.01
48	-	-	-	-	-	-	7013.66
49	-	-	-	-	-	-	19584.60
50	-	-	-	-	-	-	566.78
51	-	-	-	-	-	-	159.88
52	-	-	-	-	-	-	458390.46
53	1760	478.95	381.55	773.35	2.0183	171.5939	65471.64
54	101	307.95	13690	40.96	0.1353	0.620305	8491.98
55	101.325	416.45	573.09	71676.591	203.887	10887.68	10887.68

The destruction exergy rate (*I*), exergy destruction fraction (*%I*) and exergetic efficiency (η) of each equipment are calculated and shown in Tab. 4. It is important to stand out that in the calculation of the exergetic efficiency of the cycle not only the irreversibility of heat transfer in the steam generator and the exergy destruction associated with combustion are considered, but exergy lost with exhaust gases from the furnace are also included (Aljundi, 2008).

Table 4. Exergy destruction and exergy efficiency of the power plant components.

Subsystem		**I (kW)**	**% I**	**η**
1	Combustor	287264.95	42.88	23.40
2	Fan	1311.49	0.20	81.30
3	Air preheaters	15788.36	2.36	82.76
4	Steam generator	218125.22	32.56	88.16
5	HP-Turbine	24145.80	3.60	78.87
6	VHP- Turbine	21474.45	3.21	81.20
7	IP1- Turbine	8008.97	1.20	81.37
8	IP-LP- Turbine	51768.39	7.73	81.85
9	FWH10	2787.70	0.42	98.26
10	FWH9	2658.32	0.40	98.03
11	FWH8	1332.13	0.20	98.79
12	Deaerator	1333.19	0.20	98.00
13	FWH6	2276.69	0.34	94.56
14	FWH5	2266.67	0.34	92.70
15	FWH4	1509.24	0.23	93.02
16	FWH3	1306.46	0.20	90.73
17	FWH2	1012.29	0.15	87.62
18	FWH1	876.35	0.13	78.72
19	Condenser	21943.21	3.28	29.18
20	Pump	95.53	0.01	83.15
21	Pump	2691.43	0.40	86.26
Power cycle		674422.83	100.00	40.42

Table 4 shows that 40.42% of resources that enter in the system are converted into electrical energy, once the combustor and steam generator are responsible for 75.4% of exergy destruction. That destruction is mostly due to irreversibilities generated by the combustion process and low heat transfer in the steam generator, while the destruction exergy rate in the condenser is only 3.28% even with the low exergetic efficiency of this equipment. The remaining exergy destruction of 21.32% is concentrated mostly in the expansion of steam 15.74%.

Once the thermodynamic properties of flows are known and considering each subsystem of Fig. 1 as a control volume, the linear system is solved according to Eq. (13-17) by the inverse matrix method determining the values of K* and B* of each exergy flow and the partial inefficiency of each equipment ($\delta i = I/FT$); the results are summarized in Tab. (5-6). The highest unitary exergetic costs were found in the output flow of the condenser (dissipative equipment), where its product is assessed a null exergy cost, due to a non-external valuation.

Table 5. Exergetic cost and unit exergetic cost.

Flow	B (kW)	B* (kW)	K*	Flow	B (kW)	B* (kW)	K*
1	1157101.86	1157101.86	1	28	12614.19	26046.53	2.06
2	70606.39	144056.99	2.04	29	82364.81	264205.08	3.20
3	940443.3	1301158.84	1.38	30	12735.55	26221.35	2.05
4	5702.17	17350.29	3.04	31	38412.93	157655.28	4.10
5	91580.26	126706.69	1.38	32	27243.33	130601.56	4.79
6	147464.06	413488.16	2.80	33	1176.92	2367.95	2.01
7	438283.27	918788.39	2.09	34	18693.42	108838.76	5.82
8	316773.62	650676.62	2.05	35	1545.72	3109.97	2.01
9	405689.67	816240.9	2.01	36	11759.40	91851.05	7.81
10	534728.59	1098372.38	2.05	37	1419.39	2855.79	2.01
11	592840.50	1242792.05	2.09	38	6592.87	78827.53	11.95
12	40301.18	84484.77	2.09	39	1027.91	2068.14	2.01
13	28791.21	59139.30	2.05	40	3051.16	69664.95	22.83
14	15520.54	31880.34	2.05	41	571.81	1150.48	2.01
15	15656.33	31880.34	2.03	42	1020.09	63815.26	62.5
16	348075.89	700322.92	2.01	43	190.59	383.47	2.01
17	14623.21	29421.66	2.01	44	92781.59	239518.88	2.58
18	11185.38	22504.81	2.01	45	133841.08	324795.75	2.42
19	8316.93	16733.53	2.01	46	34981.59	86496.31	2.47
20	6081.50	12235.87	2.01	47	233466.01	573886.89	2.45
21	4097.90	8244.91	2.01	48	7013.66	17350.29	2.47
22	2526.20	5082.67	2.01	49	19584.60	48448.11	2.47
23	30633.55	61634.20	2.01	50	566.78	1402.08	2.47
24	548.84	62413.17	113.71	51	159.88	395.50	2.47
25	119815.45	349683.48	2.91	52	458390.46	1157101.86	2.47
26	9864.88	20680.09	2.09	53	65471.64	215756.98	3.29
27	96431.85	295910.61	3.06	-	-	-	-

Table 6. Unit exergetic cost and inefficiency partial of each equipment.

Subsystem		F (kW)	P (kW)	I (kW)	K*	δi
1	Combustor	1227708.25	940443.30	287264.95	1.31	0,2483
2	Fan	7013.66	5702.17	1311.49	1.23	0.0011
3	Air preheaters	80692.58	64904.22	15788.36	1.24	0.0136
4	Steam generator	848863.04	630737.82	218125.22	1.35	0.1885
5	HP-Turbine	157986.88	133841.08	24145.80	1.18	0.0209
6	VHP- Turbine	114256.04	92781.59	21474.45	1.23	0.0186
7	IP1- Turbine	42990.57	34981.59	8008.97	1.23	0.0069
8	IP-LP- Turbine	270611.19	233466.01	37145.18	1.16	0.0321
9	FWH10	30436.30	27648.61	2787.70	1.10	0.0024
10	FWH9	26041.91	23383.59	2658.32	1.11	0.0023
11	FWH8	15399.18	14067.05	1332.13	1.09	0.0012
12	Deaerator	66804.82	65471.64	1333.19	1.02	0.0012
13	FWH6	13446.29	11169.60	2276.69	1.20	0.0020
14	FWH5	10816.59	8549.91	2266.67	1.27	0.0020
15	FWH4	8443.27	6934.02	1509.24	1.22	0.0013
16	FWH3	6472.98	5166.52	1306.46	1.25	0.0011
17	FWH2	4554.00	3541.71	1012.29	1.29	0.0009
18	FWH1	2907.42	2031.07	876.35	1.43	0.0008
19	Condenser	30435.18	8491.98	21943.21	3.58	0.0190
20	Pump	566.78	471.25	95.53	1.20	0.0001
21	Pump	19584.60	16893.17	2691.43	1.16	0.0023
22	Generator	495070.28	476001.06	19069.21	1.04	0.0165
Power Cycle		1157101.86	458390.46	674422.84	2.52	0.5828

6. CONCLUSIONS

The thermoeconomic analysis presented in this paper is appropriate to deepen into an effective use of resources, providing the location, causes and magnitude of inefficiencies and irreversibility generated in a supercritical cycle. This information should be used in the design of new and more efficient energy systems and to increase the efficiency of existing ones. Considering that the exergetic cost theory is a systematic tool for identifying all sources and costs for optimizing thermal systems of power generation.

The Efficiency improvement in the supercritical power plant analyzed (40.4%) compared with the efficiencies in conventional coal power plants (35-38%) can represent a reduction of about 10% in the CO_2 emission considering the same installed capacity.

7. REFERENCES

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5. RESPONSIBILITY NOTICE