Thermoeconomic Evaluation of a Basic Optimized Chemically Recuperated Gas Turbine Cycle

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Abstract. In a thermoeconomic analysis, the productive structure determines the cost distribution. In this work, a chemically recuperated gas turbine cycle previously optimized in order to reduce the production of irreversibilities is analyzed using to thermoeconomic techniques. The exergy flows are disaggregated into chemical and physical exergies. The internal cost distribution is compared in two types of analyses: the Basic approach, where the flow costs are accumulated subsystem by subsystem and the Functional Analysis approach, where the each subsystem plays a global role in the cycle. A cogeneration possibility is considered, and the results are discussed.

Keywords. Thermoeconomics, optimization, chemical recovery, gas turbine, power plant.

1. Introduction

Chemically recuperated gas turbine cycle is one of the advanced cycles that in a near future can be used to produce electric power at high thermodynamic performances (Briesch et al., 1995). This cycle take an additional advantage when cogeneration is desired. Its evaporator can supply the cycle with steam surpluses to sell.

Cycles with chemical recovery were studied with respect to several aspects. Kesser (1994) explored the relationships between the reformer and the turbine for two compressor pressure air ratios. Souza-Santos (1997) studied variants of the cycle with chemical recovery taking into account the chemical composition of the natural gas. The physical and chemical aspects of the reformer can be seen in Adelman et al. (1995), and Carcasci et al. (1998). Sanchez Prieto et al. (2000) did an Exergy analysis of the cycle to determine its exergetic efficiency.

In this work, a basic chemically recuperated cycle was studied. It was analyzed according to the first and second law of thermodynamics. The cycle was previously optimized in order to minimize the production of irreversibilities, Alves and Nebra (2002).

The net operational profit, understood as the difference between the sale of the products and the cost of the inputs, was calculated. The equipmente costs and maintenance were not analysed.

A thermoeconomic analysis was performed to estimate the exergy cost in each flow. Two methodologies were utilized, the Basic approach and the Functional Analysis approach.

In the Basic approach (Valero and Lozano, 1993), each control volume corresponds to one subsystem in the cycle, and there are no other control volumes. The exergetic cost is accumulated subsystem by subsystem, according to the physical flow.

The Functional Analysis approach (Frangopoulos, 1983, 1987) incorporated a disaggregation of exergy into chemical and physical exergies. The results with and without negentropy were compared. The possibility of producing saturated steam was considered.

This work aimed to diagnose the cost evolution along the cycle identifying each subsystem contribution analysing two different approachs.

2. Cycle Model

The cycle presented in this work is a basic gas turbine cycle with chemical recovery. Details can be seen in Fig. (1).

In order to make the case the most general possible, the air compressor flow was taken as the reference. The ratio between the mass of methane and the mass of air can vary between 1:100 and 1:40, and the ratio between the mass of methane and the mass of water can vary between 1:3 and 1:7.5. These ratios were taken as the operational base to the turbine. It is not possible to operate economically below this level of methane, and superior levels would make the temperature surpass 1400ºK, the maximum temperature acceptable at the turbine inlet without blade cooling (Saravanamuttoo et al., 2001). The amount of injected water also needs to be limited due to the physical parameters of the turbine.

The natural gas is composed of several substances, the main one being methane. For simplicity, methane was assumed as the only component of the natural gas. In the same way, the composition of the dry air was assumed as 21% oxygen and 79% nitrogen.
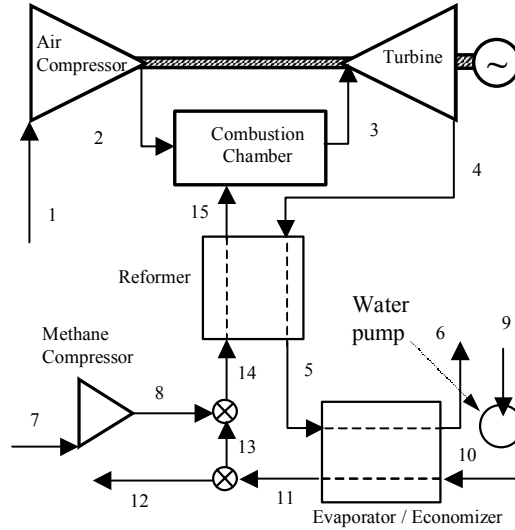
Figure 1. Cycle schematic.

After absorbing heat in the reformer, the methane reacts with the steam according to the following reactions:

$$CH_4 + H_2O \rightarrow CO + 3\,H_2 \tag{1}$$
$$CO + H2O \rightarrow CO2 + H2. \tag{2}$$

The ratio between steam and methane must be high enough to prevent the formation of carbon deposits. In industrial reformers, this range is between 3 and 5. This proportion is used widely for the production of hydrogen and for the synthesis of ammonia and methanol (Carcasci et al., 1998). For this case, the catalyst must be active at low temperatures, i.e. 800 K.

The global reaction, written as a function of the methane and the steam that enter as raw materials, is expressed as

$$\alpha_1 CH_4 + \alpha_2\,H2O \rightarrow \beta_1\,CH_4 + \beta_2\,CO + \beta_3\,CO_2 + \beta_4\,H_2 + \beta_5\,H_2O, \tag{3}$$
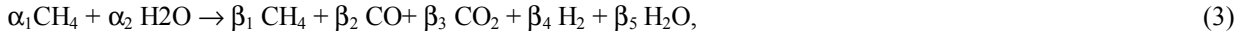
where $\alpha1$ and $\alpha2$ are known amounts of the process feed. $\beta1$ to $\beta5$ are obtained by the stoichiometric balance calculation and equilibrium equation (for more details see Alves and Nebra, 2002). All the thermodynamic properties were calculated based on the Janaf tables using the software EES®.

An analysis of each component of the cycle supplies the system equations for the compressor (4 and 5), turbine (6 - 8), combustion chamber (9 and 10), reformer (11 - 13) and evaporator (14), i.e.:

$$m_1\,(h_{2,iso} - h_1) - \eta_c\,W_c = 0 \tag{4}$$
$$s_1 = s_{2,iso} \tag{5}$$
$$m_3\,(h_3 - h_{4,iso})\,\eta_t - W_t = 0 \tag{6}$$
$$W_e - (W_t - W_c)\,\eta_G - W_{mc} - W_p = 0 \tag{7}$$
$$s_3 = s_{4,iso} \tag{8}$$
$$m_1 h_2 + m_{15} h_{15} - m_3\,h_3 + \Delta H_{r,c} = 0 \tag{9}$$
$$m_2 + m_{15} - m_3 = 0 \tag{10}$$
$$m_3 (h_4 - h_5) + m_{15}(h_{15} - h_{14}) + \Delta H_{r,r} = 0 \tag{11}$$
$$m_{15} - m_8 - m_{13} = 0 \tag{12}$$
$$m_{11} - m_{12} - m_{13} = 0 \tag{13}$$
$$m_{11}(h_{11} - h_{10}) = m_3(h_5 - h_6) \tag{14}$$

where the compressor isentropic efficiency, η_c, is 0.85 and W_c is the power required by the compressor. The turbine isentropic efficiency is 0.9, $\eta_G = 0.98$ is the electric generator efficiency, W_e is the generated electric power, W_{mc} and W_p are the power consumed in the methane compressor and the water pump, $\Delta H_{r,c}$ is the combustion reaction enthalpy at 25ºC (gas enthalpy reference), and $\Delta H_{r,r}$ is the reforming reaction enthalpy.

3. Exergetic Analysis

The importance of the exergetic analysis is to diagnose how much of the theoretical maximum work the system is able to produce. The method of exergetic analysis used consists of evaluating each bearer of energy through the system and identifying its chemical composition, physical state, and flow rate.

In order to analyze the cycle, the system was divided into seven control volumes to identify the performance of each one of these volumes. The following considerations were assumed:

Standard atmosphere for the calculation of the physical exergies: T0 = 25ºC, P0 = 101.3 kPa and a relative humidity of the air of 70%.

Efficiency determination of each control volume in agreement with the fuel, product and losses, concept of Kotas (1985).

Calculations of the physical exergy (e_{ph}), chemical (e_{ch}), and total exergy (ex), were made using (Szargut, 1988):

$$e_{ph} = (h - h0) - T0 (s - s0) \tag{15}$$
$$ex = e_{ph} + e_{ch} \tag{16}$$

where e_{ch} is defined based on the methodology and tables reported by Szargut (1988).

The gas mixtures from the reformer exit were assumed to be mixtures of ideal gases with specific heat varying with temperature. The gas compositions through the cycle were determined based on mass balances and the assumption of complete combustion in the combustion chamber.

Table 1. Fuel and product through the cycle for each piece of equipment.

Control Volume	F	P
Air compressor	W_c	Ex_2-Ex_1
Combustion Chamber	$m_{15} (e_{x15}-e_{x3})$	$m_2 (e_{x3}-e_{x2})$
Turbine	Ex_3-Ex_4	W_t
Reformer	Ex_4-Ex_5	$Ex_{15}-Ex_8-Ex_{13}$
Methane compressor	W_{mc}	Ex_8-Ex_7
Evaporador	Ex_5-Ex_6	$Ex_{11}-Ex_{10}$
Water pump	W_p	$Ex_{10}-Ex_9$
Cycle	$Ex_1+Ex_7+Ex_9$	W_e+Ex_{12}

The exergetic efficiency, ε, and the irreversibility, I, for each component were calculated according to the following:

$$\varepsilon = \frac{P}{F} \tag{17}$$
$$I = P - F \tag{18}$$

where, P is the product and F the thermodynamic fuel of each subsystem, i.e. control volume. A summary of the F and P for each subsystem appears in Tab. (1).

4. Optimization

The objective function is the operational net profit, understood as the revenue from the sale of electric power and saturated steam minus the cost of the inputs, i.e. feedwater and natural gas. This is expressed as

$$z = C_e W_v + C_v m_v - C_c m_c - C_a m_a \tag{19}$$

The costs and prices are given in Tab. (2), in equation 19 Wv is the electrical energy produced by the cycle.

Table 2. Costs and prices of the cycle

	Cost / Price
C_e	44,87 US\$/MWh (Aneel, 2001)
C_c	0.0778 US\$/kg (Gazeta Mercantil, 2000)
C_v	0.009 US\$/kg (Guarinello Jr et al., 2000)
C_a	0.00022 US\$/kg (Guarinello Jr et al., 2000)

Restrictions on the system

- $T_3 < 1400$ K (1127 ºC): The turbine cannot sustain greater temperatures without risk of thermal fatigue.

- $P_2/P_1 \leq 14.8$: Assumed maximum air compressor operational compression ratio (Saravanamuttoo et al., 2001).

- $h_4 = h_3 - \eta_t (h_3 - h_{4,iso})$: The adiabatic expansion links the variation of temperature to pressure. This relationship supplies the smallest value possible for the temperature T_4. The lower the temperature at the point 4, the higher the electric generation but the lower the capacity to generate steam.

- $T_6 > 413$ K (140 ºC): This value must be high enough to avoid gas condensation, which can liquefy acid vapors in the stack walls.

- $m_{12} \geq 0$: This relation indicates that the system needs to provide at least the steam necessary for its own use.

- 650 K $< T_{15} <$ 923 K: The reforming temperature should be between these two limits. Below 650 K the reforming does not occur and above 923 K the chemical equilibrium equations used are not valid.

- $T_4 - T_{15} = 20$ ºK: Imposed condition of the heat transfer in the reformer (Pinch Point).

- $0.1 \leq m_{15} \leq 0.25$: Turbine operational conditions that regulate the flow and the excess of air

- $0.3 \leq m_{13} \leq 0.16$: Turbine operational conditions.

Using these constraints and appropriate exergy and mass balances, the objective function (equation 19) is maximized with respect of set decision variables using EES® software. The decision variables are:

- R_p: Air compressor ratio P_2/P_1
- m_9: Feedwater.
- m_7: Methane mass.
- T_{15}: Reforming temperature reaction.

5. Operational Results

The optimal solution shows that the largest operational net profit is obtained with the largest possible temperature in the turbine inlet associated with the largest compression ratio (14.8) in the air compressor. This solution shows that in this cycle, for the present prices of electric power and saturated steam, the electricity production is more profitable than steam production, which makes the system prioritize electric production without steam surpluses. The maximum net profit found is 0.00457 US$ per kilogram of compressed air. The optimal steam-to-methane mol ratio optimal is 7.2.

Table (3) presents the optimal thermodynamic values for each point shown in Fig. (1). The chemical composition of the flows is presented in the tab. (4).

This system shows a high exergetic and energetic performance. Tab. (5) gives the exergetic e and the energetic h performance of each subsystem.

On the chemical reforming side, the evaporator/economizer group is the largest source irreversibility. This is due to the high temperature difference between the hot gases and the steam. When compared to the evaporator/economizer, the reformer generates much less irreversibility. This is due to the optimization of the reformer, where there are small temperature differences. This conclusion agrees with those by Harvey and Kane (1997), Kesser (1994) and Sanchez Prieto et al. (2000).

Table 3. Operational parameter values of the cycle at the optimal conditions.

Substance	T (ºK)	m (kg)	P (kPa)	E_{ph} (kJ)	E_{ch} (kJ)	Ex (kJ)
1 (air)	298	1	101.3	0	0	0
2 (air)	689	1	1500	383	0	383
3 (gases)	1400	1.166	1477	1294	63	1356
4 (gases)	829	1.166	105.4	332	63	394
5 (gases)	689	1.166	103.4	202	63	265
6 (gases)	413	1.166	101.3	24	63	87
7 (methane)	298	0.020	499.5	5	1056	1061
8 (methane)	415	0.020	2138	11	1056	1067
9 (water)	298	0.146	199.6	0	7	7
10 (water)	298	0.146	2074	0	7	7
11 (steam)	485	0.146	1972	133	7	140
12 (steam)	-	0	-	-	-	-
13 (steam)	485	0.146	1972	133	7	140
15 (synthesis gas)	809	0.166	1950	155	1155	1310

Due to the characteristics of this system it presents a high efficiency, similar to that of combined cycles. The optimal efficiency of our cycle is lower than that of Kesser (1994) whose overall cycle electrical efficiency is 47.6% versus ours for 46.5%. This difference is due to the lower maximum temperature at the turbine inlet assumed in this work. If the turbine has no blade cooling, the temperature at the combustion chamber exit must be lower, which results in higher irreversibility. These performance results were calculated for the optimal conditions, therefore without steam for sale. The cycle performance is related to electrical efficiencies.

Table 4. Mole fractions of the optimal flows in the cycle

Substance	Flow 3	Flow 14	Flow 15
CH_4	-	0.1357	0.0878
CO_2	0.0286	-	0.0356
CO	-	-	0.0020
O_2	0.1047	-	-
N_2	0.6095	-	-
H_2O	0.2572	0.8643	0.7259
H_2	-	-	0.1486

Table 5. Optimal component performance

Equipment	ε (%)	I (kW)	I (%)	η (%)
Air compressor	93.2	27.8	4.9	85
Combustion chamber	69.2	336.6	58.8	100
Turbine	94.9	48.7	8.5	90
Reformer	79.7	26.2	4.6	100
Methane compressor	96.1	0.2	0.0	94.6
Evaporator	74.3	45.8	8.0	100
Water pump	90.0	0.0	0.0	90
Stack	0	86.8	15.2	0
Cycle	46.5	572.2	100	46.8

6. Thermoeconomic Analysis

Three methodologies were applied in order to estimate the exergetic costs, the Basic Analysis approach and the Functional Analysis approach with and without negentropy cost redistribution. Two scenarios are proposed. Case 1 corresponds to the operational conditions calculated in the previous section, which represent the optimum conditions. For the second case the same compression ratio in the air compressor and the same inlet turbine temperature are assumed, but an extra quantity of saturated steam in the evaporator (flow 12) is assumed. Flow 12 is chosen to be 20% of the total evaporator flow.

In the case 2, 0.030 kg of saturated steam and 0.1286 kWh of electric power are produced. The exergetic cycle efficiency goes to 47.25%. The electric power decreases because there is less water expansion in the turbine. Less water injected in the combustion chamber decreases the irreversibility and increases the efficiency.

6.1. Basic analysis approach

In the Basic Analysis approach, each subsystem corresponds to one control volume in the cycle, and the exergetic cost is accumulated subsystem by subsystem. The basic hypothesis (Lozano and Valero, 1993), is that the exergetic cost of the inputs is equal to the output costs in each control volume. Thus, it is possible to write for each control volume that:

$$E^*_{in} = E^*_{out}, \tag{20}$$

where $E^* = k E_x$ is the exergetic cost, and k is the unitary exergetic cost of the flow. Applying this concept to the seven control volumes it results in

$$E^*_1 + E^*_C = E^*_2 \tag{21}$$
$$E^*_2 + E^*_{15} = E^*_3 \tag{22}$$
$$E^*_3 = E^*_4 + E^*_t \tag{23}$$
$$E^*_4 + E^*_8 + E^*_{13} = E^*_{15} + E^*_5 \tag{24}$$
$$E^*_7 + E^*_{cm} = E^*_8 \tag{25}$$
$$E^*_5 + E^*_{10} = E^*_{12} + E^*_{13} + E^*_6 \tag{26}$$
$$E^*_9 + E^*_{ba} = E^*_{10} \tag{27}$$

where $E^*_C = kp * Wmc$ and $E^*_t = kp Wt$, are the cost of the compressor's mechanical energy and the electric energy, respectively, and kp is the unitary cost of power.

These seven equations, however, are not enough to determine all the costs. Additional hypotheses are necessary to determine all 15 unitary cost variables given in Tab. (6). These hypotheses are

1 – All the flows that enter the cycle have a unitary cost equal to 1 (equation 28).

2 – The burned gases have the role of providing energy to the cycle, to the turbine, the reformer, and the evaporator. These flows have the same exergetic unitary cost (equation 29).

3 – Flows that exit the cycle without utility have null exergetic unitary cost (equation 30).

4 – No losses in the division of flow 11, so that the exergetic unitary cost is the same for flows 11 to 13 (equation 31).

$$k1 = k7 = k9 = 1 \tag{28}$$
$$k3 = k4 = k5 \tag{29}$$
$$k6 = 0 \tag{30}$$
$$k11 = k12 = k13 \tag{31}$$

Table 6. Exergetic costs in the basic analysis approach

	No saturated steam		20% saturated steam	
	k	E^*	k	E*
k1	1	0	1	0
k2	2.309	884.4	2.175	832.9
k3	2.044	2772	1.925	2493
k4	2.044	805.9	1.925	709.1
k5	2.044	541.6	1.925	498.9
k6	0	0	0	0
k7	1	1061	1	1033
k8	1.006	1074	1.006	1044
k9	1	7.295	1	7.403
k10	1.05	7.95	1.045	8.029
k11	3.923	549.6	3.566	507
k12	-	-	3.566	101.4
k13	3.923	549.6	3.566	405.6
k15	1.441	1887	1.342	1660
k_p	2.153	1069	2.027	938.7

6.2. Functional analysis

The Functional Analysis approach, exergy is disaggregated into two forms: chemical, e_{ch}, and physical, e_{ph}. In the cycle, there are two chemical inputs, water and fuel (methane). In addition to the chemical inputs, the reformer promotes a chemical reaction between methane and water, increasing the chemical exergy in the flow that traverses the reformer on the cold side.

If there is no saturated steam exiting the cycle, all the water injected (flow 9) traverses the reformer. If saturated steam is produced, the water flow at the evaporator exit is divided into two parts. The part that leaves the cycle e.g., the saturated steam, received energy for the evaporation and does not contribute to any chemical reaction. Thus, the cycle provides only physical exergy to this flow. The part that goes to the reformer reacts with the methane modifying the flows chemical composition. Therefore the water's chemical input is due only to the steam that goes to the reformer (flow 13).

The function of each control volume is well determined. The compressors and pump use mechanical (or electrical) power to increase the physical exergy of the flows. The turbine provides mechanical power through the expansion of gases. The evaporator transfers heat exergy from the hot side to the cold side. The reformer uses the hot-side heat exergy to improve the chemical and physical exergy of the flow on the cold side.

The cycle's functional diagram is shown in the Fig. (2). The control volumes have a functional utility. They may be a physical or a virtual subsystem. The control volumes are

1 – Air compressor
2 – Combustion chamber
3 – Turbine
4 – Reformer
5 – Evaporator
6 – Water pump
7 – Methane compressor
8 – Stack
9 – Physical exergy junction
10 – Chemical exergy distributor
11 – Power distributor
12 – Water exergy distributor
13 – Methane exergy distributor

14 – Chemical exergy junction
15 – Chemical exergy distributor.
Fuels and Products of the control volumes are shown in the Tab. (7).

Table 7. Fuels and products of each control volume (CV).

CV	Fuel	Product
1	$W1= W_c$	$1T= Ex_2 - Ex_1$
2	$C2 = E_{ch,15} - E_{ch,3}$	$2T= E_{ph,3} - (E_{ph,2} + E_{ph,17})$
3	$T3= E_{ph,3} - E_{ph,4}$	$W3= W_t$
4	$T4= E_{ph,4} - E_{ph,5}$	$4T= E_{ph,15} - E_{ph,13} - E_{ph,8}$ $4Ch= E_{ch,15} - E_{ch,13} - E_{ch,8}$
5	$T5= E_{ph,5} - E_{ph,6}$	$5T= E_{ph,11} - E_{ph,10}$
6	$W6= W_p$	$6T= E_{ph,10} - E_{ph,9}$
7	$W7= W_{mc}$	$7T= E_{ph,8} - E_{ph,7}$
8	$C8= E_{ch,6}$ $T8= E_{ph,6}$	$L= 0$
9	$1T; 2T; 4T; 5T; 6T; 7T;$ $ST= E_{ph,9}$ $FT= E_{ph}7$	$T=1T+2T+4T+ 5T+ 6T+ 7T+ST+FT$
10	T	$T8; T3; T4; T5$ $Q= E_{ph,12}$
11	$3W$	$W1; W6; W7$ W_e
12	$S= E_{x,9}$	$ST;$ $SC= E_{ch,9}$
13	$F= E_{x,7}$	$FC= E_{ch,7}$ $FT= E_{ph,7}$
14	$FC; SC; 4Ch$	$C = FC+ SC+ 4Ch$
15	C	$C8; C2$

IG, in Fig. (2), represents the electric generator losses.

It's necessary to make some hypotheses in order to resolve the system of equations. As usual, it was assumed that the total exergetic cost, in each control volume, is constant. All control volume products of the same type have the same cost. Each cycle input has a unitary exergetic cost equal to one. Flow without utility has a unitary exergy cost zero. Making these hypotheses it is possible to write the following

$$k_F = k_S = 1 \tag{32}$$
$$E^*_{W1} = E^*_{1T} \tag{33}$$
$$E^*_{C2} = E^*_{2T} \tag{34}$$
$$E^*_{T3} = E^*_{3W} \tag{35}$$
$$E^*_{T4} = E^*_{4T} \tag{36}$$
$$E^*_{T4} = E^*_{4T} + E^*_{4Ch} \tag{37}$$
$$k_{T4} = k_{4T} \tag{38}$$
$$E^*_{T5} = E^*_{5T} \tag{39}$$
$$E^*_{W6} = E^*_{6T} \tag{40}$$
$$E^*_{W7} = E^*_{7T} \tag{41}$$
$$E^*_{L} = E^*_{C8} + E^*_{T8} \tag{42}$$
$$k_{C8} = k_{T8} = 0 \tag{43}$$
$$E^*_{1T} + E^*_{2T} + E^*_{4T} + E^*_{5T} + E^*_{6T} + E^*_{ST} = E^*_{FT} \tag{44}$$
$$E^*_{T} = E^*_{T3} + E^*_{T4} + E^*_{T5} + E^*_{Q} + E^*_{T8} \tag{45}$$
$$E^*_{3W} = E^*_{W1} + E^*_{W6} + E^*_{W7} + E^*_{We} \tag{46}$$
$$E^*_{S} = E^*_{ST} + E^*_{FT} \tag{47}$$
$$k_{ST} = k_{SC} = 1 \tag{48}$$
$$E^*_{F} = E^*_{FT} + E^*_{FC} \tag{49}$$
$$k_{FT} = k_{FC} = 1 \tag{50}$$
$$E^*_{FC} + E^*_{SC} + E^*_{4Ch} = E^*_{C} \tag{51}$$
$$E^*_{C} = E^*_{C2} + E^*_{C8} \tag{52}$$
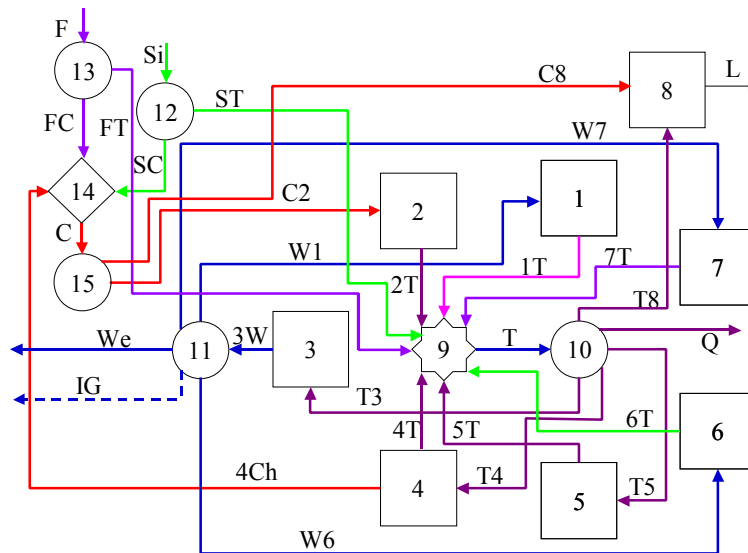
Figure 2 – Cycle functional diagram

Table 8. Functional analysis unitary exergetic costs

	Case 1		Case 2	
	k	E^* (kJ)	k	E^* (kJ)
C	1.129	1305	1.112	1232
C2	1.194	1305	1.167	1232
4C	2.628	241.2	2.664	198.3
F	1	1061	1	1032
Fc	1	1056	1	1028
Ft	1	5.023	1	4.888
S	1	7.295	1	5.926
Sc	1	7.281	1	5.911
St	1	0.014	1	0.015
1T	2.309	884.4	2.28	873.3
2T	1.726	1305	1.687	1232
4T	2.044	23.08	2.018	22.01
5T	2.749	364.3	2.753	370.2
6T	2.392	0.6551	2.362	0.6564
7T	2.241	12.3	2.213	11.82
T	2.005	2595	1.981	2515
T3	2.044	1966	2.018	1870
T4	2.044	264.3	2.018	220.3
T5	2.044	364.3	2.018	370.2
Q	2.044	0	2.018	54.4
3W	2.129	1966	2.103	1870
W1	2.153	884.4	2.126	873.3
W6	2.153	0.6551	2.126	0.6564
W7	2.153	12.3	2.126	11.82
W_e	2.153	1069	2.126	984.2
C8	0	0	0	0
T8	0	0	0	0
L	0	0	0	0

It was assumed that the main function of the combustion chamber is to convert the chemical exergy, contained in the "fuel", into physical exergy. Therefore, the exergetic fuel of this subsystem was considered as the chemical exergy difference between the combustion chamber inlet and outlet flows. Furthermore an input exergetic unitary cost equal to 1 was assumed for the water and methane for their physical and chemical components. These particular hypotheses were made for control volumes 4 (the reformer) and 8 (the stack).

In control volume 4, there are two products: chemical exergy 4C and physical exergy 4T. It was assumed that the reformer's main function is to increase the chemical exergy of flow 15 (see Fig. 1). Therefore, this hypothesis implies that the unitary exergy costs of the inlet and outlet physical exergy flows are equal.

When the flow has no utility, its unitary exergetic cost is zero. The costs of flows T8 and C8 are thus zero. They represent the cycle exergy losses by the stack.

Table (9) summarizes the results for the two situations: no saturated steam produced (case 1) and 20% of steam sold (case 2).

7. Conclusions

The purpose of this work was the optimization and thermoeconomic analysis of a simplified gas turbine cycle with chemical recovery. The optimisation showed that the difference between the electric power and saturated steam prices made it more advantageous to produce and trade electric power than saturated steam. When saturated steam is produced at about 20% of the steam production in the evaporator, the electric production decreases by about 5%.

It was shown that disaggregating the exergy into chemical and physical exergies is particularly important for processes where the chemical composition varies, as in the combustion chamber and the reformer. In these subsystems, the primary function of these components is to modify the chemical composition. Therefore, disaggregating the exergy is a convenient way to evaluate the performance of these control volumes.

Finally, in this work the negentropy modifies the costs by about of 1%.

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