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Thermoeconomic Evaluation of a Basic Optimized Chemically Recuperated Gas Turbine Cycle

Lourenço Gobira ALVES*, Silvia Azucena NEBRA -
Energy Dept. of Mech. Engineering Faculty,
State University of Campinas.
DE/FEM/UNICAMP, POBox: 6122,
CEP 13083-970 - Campinas – SP
Fax: +55 – 19 – 32893722
E-mail: lourenco@fem.unicamp.br, sanebra@fem.unicamp.br

Abstract

In a thermoeconomic analysis, the productive structure determines the cost distribution. A chemically recuperated gas turbine cycle, 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 three types of analyses: the Basic approach, the Functional Analysis approach and the Functional Analysis with Cost Negentropy Redistribution approach. The negentropy cost redistribution reduces the cogeneration cost, changing the exergetic costs through the cycle by about 1%.

Key words: Thermoeconomic, optimization, chemical recovery, gas turbine, power plant.

1. Introduction

Gas turbines that operate in simple cycles have low efficiencies because the turbine exhaust gases come out very hot and this energy is lost to the atmosphere. Better performance is reached with advanced cycles (Briesch et al., 1995), that take advantage of the energy contained in the turbine exhaust gases to improve the cycle or to transfer energy to combined cycles.

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 optimized in order to minimize the production of irreversibilities. The net operational profit, understood as the difference between the sale of the products and the cost of the inputs, was calculated.

A thermoeconomic analysis was performed to estimate the exergy cost in each flow. Three methodologies were utilized, the Basic approach, the Functional Analysis approach and the Functional Analysis with Negentropy Cost Redistribution 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 with and without negentropy (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. At this condition, the approach with negentropy results in a penalty to processes where an increase in entropy occurs

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[*] Author to whom all correspondence should be addressed.

and a credit where the entropy decreases. With negentropy the costs are modified by about 1%.

2. Cycle Model

The cycle presented in this work is a basic gas turbine cycle with chemical recovery. Details can be seen in *Figure 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.

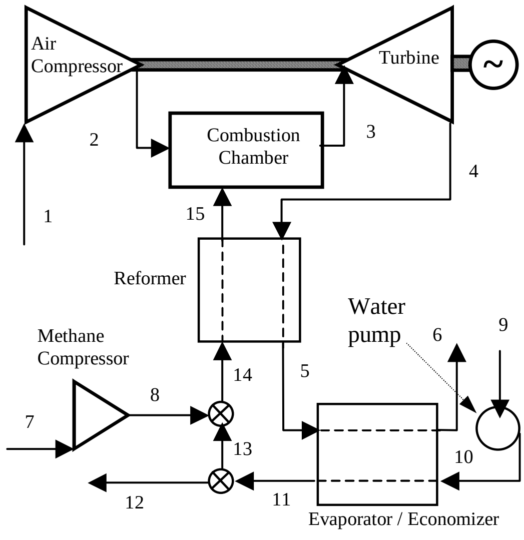

Figure 1. Cycle schematic.

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

$$CH_4 + H_2O \rightarrow CO + 3\,H_2 \qquad (1)$$

$$CO + H_2O \rightarrow CO_2 + H_2. \qquad (2)$$

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. 800º K.

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

$$\alpha_1 CH_4 + \alpha_2\, H_2O \rightarrow \beta_1\, CH_4 + \beta_2\, CO + \beta_3\, CO_2 + \\ + \beta_4\, H_2 + \beta_5\, H_2O, \qquad (3)$$

where α_1 and α_2 are known amounts of the process feed. β_1 to β_5 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.:

$$m_1\,(h_{2,iso} - h_1) - \eta_c\, W_c = 0 \qquad (4)$$

$$s_1 = s_{2,iso} \qquad (5)$$

$$m_3\,(h_3 - h_{4,iso})\,\eta_t - W_t = 0 \qquad (6)$$

$$W_e = (W_t - W_c)\,\eta_G - W_{mc} - W_p = 0 \qquad (7)$$

$$s_3 = s_{4,iso} \qquad (8)$$

$$m_1 h_2 + m_{15} h_{15} - m_3\, h_3 + \Delta H_{r,c} = 0 \qquad (9)$$

$$m_2 + m_{15} - m_3 = 0 \qquad (10)$$

$$m_3(h_4 - h_5) + m_{15}(h_{15} - h_{14}) + \Delta H_{r,r} = 0 \qquad (11)$$

$$m_{15} - m_8 - m_{13} = 0 \qquad (12)$$

$$m_{11} - m_{12} - m_{13} = 0 \qquad (13)$$

$$m_{11}(h_{11} - h_{10}) = m_3(h_5 - h_6) \qquad (14)$$

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 in order to identify the performance of each one of these volumes. The following were assumed:

- Standard atmosphere for the calculation of the physical exergies: $T_0 = 25°C$, $P_0 = 101.3$ 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 (e_x), were made using (Szargut, 1988):

$$e_{ph} = (h - h_0) - T_0 (s - s_0) \qquad (15)$$

$$e_x = e_{ph} + e_{ch} \qquad (16)$$

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 I. FUEL AND PRODUCT THROUGH THE CYCLE.

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:

$$\varepsilon = \frac{P}{F} \qquad (17)$$

$$I = P - F \qquad (18)$$

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 TABLE I.

4. Optimization

The objective function is the operational net profit, understood as 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

$$z = C_e W_v + C_v m_v - C_c m_c - C_a m_a \qquad (19)$$

where W_v is the electrical energy produced by the cycle.

The costs and prices are given in TABLE II.

TABLE II. 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 °C): The turbine cannot sustain greater temperatures without risk of thermal fatigue.
- $\frac{P_2}{P_1} \le 14.6$: 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 °C): This value must be high enough to avoid gas condensation, which can liquefy acid vapors in the stack walls.
- $m_{12} \ge 0$: This relation indicates that the system needs to provide at least the steam necessary for its own use.
- $650° \text{ K} < T_{15} < 923° \text{ K}$: 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.
- $T_4 - T_{15} = 20$ °K: 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:

1. R_p: Air compressor ratio $\frac{P_2}{P_1}$
2. m_9: Feedwater.
3. m_7: Methane mass.
4. 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 III. presents the optimal thermodynamic values for each point shown in *Figure 1*. The chemical composition of the flows is presented in the TABLE IV.

TABLE III. OPERATIONAL PARAMETER VALUES OF THE CYCLE AT THE OPTIMAL CONDITIONS.

Substance	T (ºK)	m (kg)	P (kPa)
1 (air)	298	1	101.3
2 (air)	689	1	1500
3 (gases)	1400	1.166	1477
4 (gases)	829	1.166	105.4
5 (gases)	689	1.166	103.4
6 (gases)	413	1.166	101.3
7 (methane)	298	0.020	499.5
8 (methane)	415	0.020	2138
9 (water)	298	0.146	199.6
10 (water)	298	0.146	2074
11 (steam)	485	0.146	1972
12 (steam)	–	0	–
13 (steam)	485	0.146	1972
15 (synthesis gas)	809	0.166	1950

TABLE IV. 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

The chemical and physical exergies of each stream are shown in the TABLE V. This system shows a high exergetic and energetic performance. TABLE VI gives the exergetic and the energetic performance of each subsystem.

TABLE V. OPTIMAL EXERGY THROUGH THE CYCLE

Flow	E_{ph} (kJ)	E_{ch} (kJ)	E_x (kJ)
1	0	0	0
2	383	0	383
3	1294	63	1356
4	332	63	394
5	202	63	265
6	24	63	87
7	5	1056	1061
8	11	1056	1067
9	0	7	7
10	0	7	7
11	133	7	140
13	133	7	140
15	155	1155	1310

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 VI. 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

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.

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 write for each control volume that:

$$E^*_{in} = E^*_{out}, \qquad (20)$$

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

$$E^*_1+E^*_C=E^*_2 \qquad (21)$$

$$E^*_2+E^*_{15}=E^*_3 \qquad (22)$$

$$E^*_3=E^*_4+E^*_t \qquad (23)$$

$$E^*_4+E^*_8+E^*_{13}=E^*_{15}+E^*_5 \qquad (24)$$

$$E^*_7+E^*_{cm}=E^*_8 \qquad (25)$$

$$E^*_5+E^*_{10}= E^*_{12}+ E^*_{13}+E^*_6 \qquad (26)$$

$$E^*_9+E^*_{ba}=E^*_{10} \qquad (27)$$

where $E^*_C = k_p * W_{mc}$ and $E^*_t = k_p W_t$, are the cost of the compressor's mechanical energy and the electric energy, respectively, and k_p 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 TABLE VII. 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).

$$k1 = k7 = k9 = 1 \qquad (28)$$

$$k3 = k4 = k5 \qquad (29)$$

$$k6 = 0 \qquad (30)$$

$$k11 = k12 = k13 \qquad (31)$$

TABLE VII. 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 *Figure 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 TABLE VIII.

TABLE VIII. 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 *Figure 2*, represents the electric generator lost.

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

$$k_F = k_S = 1 \tag{32}$$

$$E^*_{W1} = E^*_{1T} \tag{33}$$

$$E^*_{C2} = E^*_{2T} \tag{34}$$

$$E^*_{T3} = E^*_{3W} \tag{35}$$

$$E^*_{T4} = E^*_{4T} \tag{36}$$

$$E^*_{T4} = E^*_{4T} + E^*_{4Ch} \tag{37}$$

$$k_{T4} = k_{4T} \tag{38}$$

$$E^*_{T5} = E^*_{5T} \tag{39}$$

$$E^*_{W6} = E^*_{6T} \tag{40}$$

$$E^*_{W7} = E^*_{7T} \tag{41}$$

$$E^*_L = E^*_{C8} + E^*_{T8} \quad (42)$$

$$k_{C8} = k_{T8} = 0 \quad (43)$$

$$E^*_{1T} + E^*_{2T} + E^*_{4T} + \\ + E^*_{5T} + E^*_{6T} + E^*_{ST} = E^*_{FT} \quad (44)$$

$$E^*_T = E^*_{T3} + E^*_{T4} + E^*_{T5} + E^*_Q + E^*_{T8} \quad (45)$$

$$E^*_{3W} = E^*_{W1} + E^*_{W6} + E^*_{W7} + E^*_{We} \quad (46)$$

$$E^*_S = E^*_{ST} + E^*_{FT} \quad (47)$$

$$k_{ST} = k_{SC} = 1 \quad (48)$$

$$E^*_F = E^*_{FT} + E^*_{FC} \quad (49)$$

$$k_{FT} = k_{FC} = 1 \quad (50)$$

$$E^*_{FC} + E^*_{SC} + E^*_{4Ch} = E^*_C \quad (51)$$

$$E^*_C = E^*_{C2} + E^*_{C8} \quad (52)$$

TABLE IX. 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 *Figure 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 IX summarizes the results for the two situations: no saturated steam produced (case 1) and 20% of steam sold (case 2).

6.3 Functional analysis with negentropy cost redistribution

Physical exergy has two components, the enthalpy $(h - h_0)$ and the negentropy $-T_0(s - s_0)$. When the negentropy is included, the cost redistribution consists of a credit when the process decreases the entropy or a penalty when the process increases in entropy (Lozano and Valero, 1993; Frangopoulos, 1987). The functional diagram with negentropy included is given in *Figure 3*.

In contrast to *Figure 2*, *Figure 3* presents two new control volumes, 16 and 17, i.e. a negentropy junction and a negentropy distributor, respectively.

The stack consumes the residual exergy and produces the negentropy for the cycle. The role of the stack is to remove the gases, which still have some exergy, to the environment. This generates all the negentropy consumed in the cycle.

For each control volume process, the negentropy flow consumption is given by

$$ss_i = T_0 (s_i - s_{i-1}), \quad (53)$$

where “i-1” is the flow(s) coming from the upstream control volume. The negentropy cost is expressed as

$$s^*_i = k_{s,i} * s_{s,i} \quad (54)$$

where s^*_i is the negentropy cost and $k_{s,i}$ is the unitary negentropy cost.

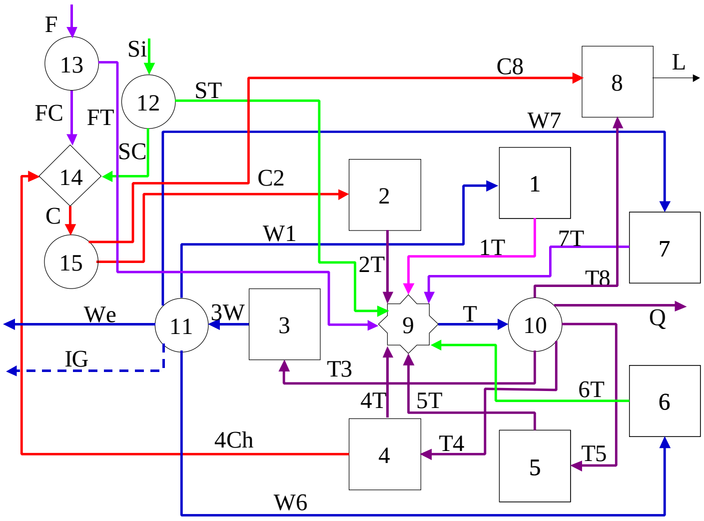

Figure 2 – Cycle functional diagram

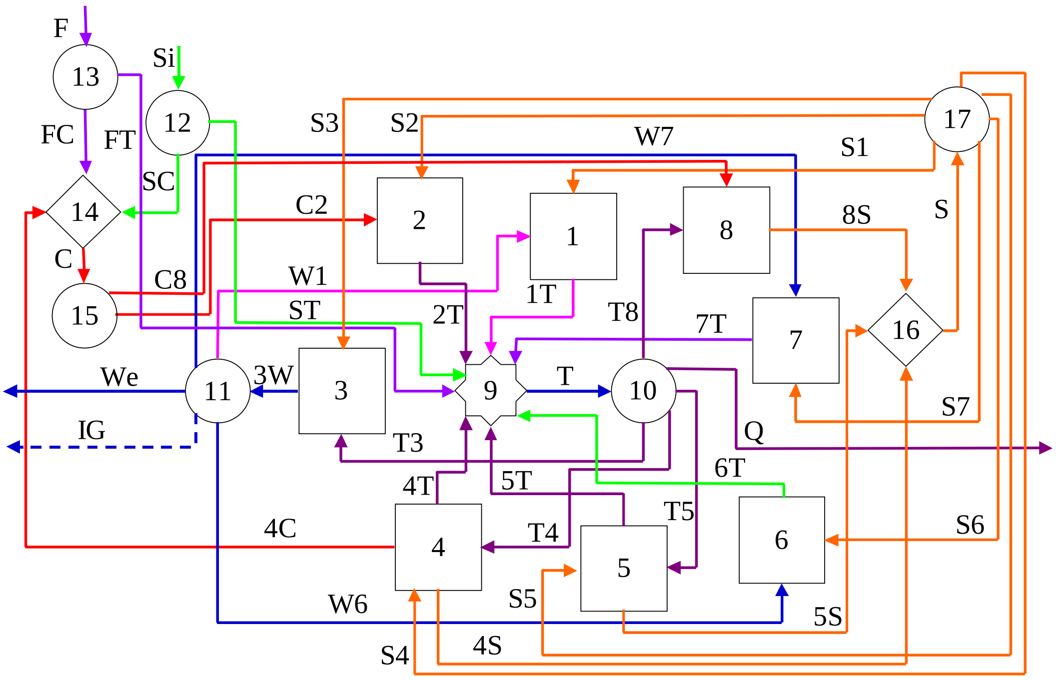

Figure 3. Cycle functional diagram with negentropy redistribution costs

Rewriting the equations for control volumes 1 to 8 and 16 and 17 results in

$$S^*_S=S^*_{8S}+S^*_{4S}+S^*_{5S} \quad (55)$$

$$S^*_S=S^*_{S1}+S^*_{S2}+S^*_{S3}+S^*_{S4}+S^*_{S5}+S^*_{S6}+S^*_{S7} \quad (56)$$

$$k_{s,1S} = k_{s,2S} = k_{s,3S} = k_{s,4S} = k_{s,5S} = k_{s,6S} = k_{s,7S} \quad (57)$$

$$E^*_{W1}+S^*_{SI}=E^*_{1T} \quad (58)$$

$$E^*_{C2}+S^*_{S2}=E^*_{2T} \quad (59)$$

$$E^*_{T3}+S^*_{S3}=E^*_{3W} \quad (60)$$

$$E^*_{T4}+S^*_{SIV}=E^*_{4T}+E^*_{4C}+S^*_{IVS} \quad (61)$$

$$k_{s,S4} = k_{s,4S} \quad (62)$$

$$E^*_{T5}+S^*_{SV}=E^*_{5T}+S^*_{VS} \quad (63)$$

$$k_{s,S5} = k_{s,5S} \quad (64)$$

$$E^*_{W7}+S^*_{SVII}=E^*_{7T} \quad (65)$$

$$E^*_{W6}+S^*_{S6}=E^*_{6T} \quad (66)$$

$$E^*_{C8}+E^*_{T8}=S^*_{8S} \quad (67)$$

All the products of control volume 17 have the same unitary exergetic cost so that it is possible to write equation 60. Other important equalities are equations 65 and 67, which are related to the reformer and the evaporator. These subsystems have negentropy as fuel (4S and 5S) and product (S4 and S5). These flows have equal unitary negentropy costs.

The exergetic costs values when negentropy is considered were recalculated and the results obtained are shown in the TABLES X and XI when there is no saturated steam for selling, case 1, and in the TABLES XI and XII for 20% of saturated steam for selling, case 2.

TABLE X. EXERGETIC COSTS WITH NEGENTROPY (CASE 1)

Flow	k	E^*	Flow	k	E^*
C	1.132	1308	7T	2.296	12.62
C2	1.132	1238	T	2.024	2620
4C	2.677	243.1	T3	2.024	1954
F	1	1063.032	T4	2.024	257.9
FC	1	1058	T5	2.024	359.2
FT	1	5.032	Q	-	0
Si	1	7.26831	3W	2.119	1964
SC	1	7.254	W1	2.143	880.4
ST	1	0.01431	W6	2.143	0.6496
1T	2.317	887.3	W7	2.143	12.26
2T	1.744	1322	W_e	2.143	1070
4T	2.024	21.34	C8	1.132	70.7
5T	2.807	370.5	T8	2.024	48.87
6T	2.409	0.6572			

TABLE XI. NEGENTROPY COSTS (CASE 1)

Flow	ks	S*	Flow	ks	S*
4S	0.2502	19.89	S3	0.2502	9.699
5S	0.2502	53.37	S4	0.2502	26.46
8S	0.767	119.6	S5	0.2502	64.75
S	0.4298	192.8	S6	0.2502	0.007585
S1	0.2502	6.964	S7	0.2502	0.3572
S2	0.2502	84.59			

TABLE XII. EXERGETIC COSTS WITH NEGENTROPY (CASE 2)

Flow	k	E^*	Flow	k	E^*
C	1.114	1235	7T	2.264	12.11
C2	1.114	1177	T	2	2487
4C	2.717	199.9	T3	2	1860
F	1	1033.895	T4	2	215.1
FC	1	1029	T5	2	365
FT	1	4.895	Q	2	53.68
Si	1	5.90052	3W	2.094	1868
SC	1	5.886	W1	2.116	869.5
ST	1	0.01452	W6	2.116	0.6508
1T	2.287	875.8	W7	2.116	11.78
2T	1.707	1250	W_e	2.116	986.5
4T	2	20.55	C8	1.114	58.45
5T	3.01	322.3	T8	2	46.41
6T	2.377	0.6578			

TABLE XIII. NEGENTROPY COSTS (CASE 2)

Flow	ks	S* (kJ)	Flow	ks	S* (kJ)
4S	0.2251	14.79	S3	0.2251	8.435
5S	0.2251	48.15	S4	0.2251	20.16
8S	0.6982	104.9	S5	0.2251	59.08
S	0.3904	167.8	S6	0.2251	0.0069
S1	0.2251	6.265	S7	0.2251	0.326
S2	0.2251	73.52			

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%.

Nomenclature

C_a	Feedwater cost (US$/kg).
C_c	Acquisition price of the fuel, i.e. natural gas (US$/kg).
C_e	Electric power price (US$/kWh).
C_v	Saturated steam price (US$/kg).
E_{ch}	Chemical exergy (kJ).
e_{ch}	Specific chemical exergy (kJ/kg).
E_{ph}	Physical exergy (kJ).
e_{ph}	Specific physical exergy (kJ/kg).
e_x	Specific total exergy (kJ/kg).
E_x	Total exergy (kJ).
F	Fuel exergy (kJ).
h	Enthalpy (kJ/kmol).
I	Irreversibility (kJ).
iso	Subscript of the isentropic condition.
K_p	Chemical equilibrium constant.
m	Mass flow (kg).
m_a	Feedwater mass (kg).
m_c	Fuel mass used (kg).
m_v	Steam mass available for sale (kg).
N	Molar mass (kmol).
N_{tot}	Total number of moles (kmol).

P	Component of subsystem product exergy (kJ).
P_0	Reference pressure (kPa).
s	Entropy (kJ/kmol-K).
T	Temperature (K).
T_0	Reference temperature (K).
W_c	Air Compressor power required (kJ).
W_e	Electric power generated (kJ).
W_{mc}	Methane compressor power required (kJ).
W_p	Water pump power required (kJ).
W_t	Turbine power (kJ).
W_v	Available electric power. (kWh).
z	Optimization objective variable (US$).
η	Energy (thermal) efficiency.
ε	Second law efficiency.
$\Delta H_{r,c}$	Combustion reaction enthalpy (kJ).
$\Delta H_{r,r}$	Reform reaction enthalpy (kJ).

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