EFFECTS ON ACTUAL PROFITS WHEN USING EXHAUST GAS TURBINE IN HEAT RECOVERY APPLICATION-CASE STUDY

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Abstract. *The advantages of using the exhaust gas of gas turbines (GT) to drive a power generator are well described in literature. The main applications are in combined or cogeneration cycles with steam production for process. The effects of pressure drop rising on GT may become important, considering the efficiency loss of GT, which affects the economic performance of the proposed heat recovery application. In the present work, part of the GT exhaust is directed to an existent fired heater burning natural gas, substituting the combustion air presently used at ambient temperature. The study considers the interaction between pressure drop, cost of the insulated duct feeding the heater, the natural gas saving and the cost of electricity that this part of exhaust gas would produce if used together with the main stream in a combined cycle.*

keywords*: gas turbines, heat recovery, profits*

1. Introduction

The poor efficiency of a Brayton cycles is a consequence of the work consumed in the air compressor. Many efforts has been spent in order to improve its efficiency with good results. These efforts resulted in the idealization of some yet innovative cycles, such as the steam and water injection, air cooling and chemical recuperation. Another way of reducing the energy loses is by giving an useful use for the heat rejected by the gas turbine (GT) in an industrial process. One typical application is the combined cycle (CC), in which hot gases from the GT generate steam in a heat recovery steam generator (HRSG) and, at the same time, produce electric or mechanical power by means of a steam turbine (ST). Alternatively, the heat can be directly used for heating purposes in applications such as dryers, chemicals reactors or, as studied in this work, in a burner of a process fluid heater.

It must be mentioned that when the objective of the plant is the electricity generation, the least waste heat must be rejected to the environment and also, if the purpose is only heat generating, the electric power generation is null. The first case can be associated with a Rankine cycle using all the steam in a condensing turbine and the second with a steam generator associated to an industrial process. All intermediate cases can be carried out by the Rankine cycle. The CC may be considered as belonging to the "electric power only" classification, but when the steam from the HRSG is partially used with heating objectives, it can be classified in the "intermediate" position. Its has no sense in using a combined or a Brayton cycle for producing heat exclusively.

The advantages of using the TG exhaust gases from the energetic and exergetic point of view are apparently obvious in a first analysis, but sometimes these advantages are not too clear and a more accurate study must be conducted. The influence of the implicit pressure drop on the gas turbine system efficiency when an extra equipment are installed downstream it is the more important aspect to be investigated. As a rule of thumb some authors (Ganapathy, 1993) indicate that the GT efficiency decreases 1% per 100 mmH_2O increase of the discharge pressure and that HRSGs can incur in a pressure drop in the range of 125 to 150 mmH_2O. The costs of the additional fuel consumed by the GT, or the minor electric power output can become comparable to the advantages estimated in the first analysis and, depending on the extra equipment costs, the proposed heat recovery alternative can be rejected.

In this work it is considered the feasibility of substituting the combustion air of an industrial fired heater burner by using part of the rich oxygen exhaust gases from a GT. To obtain a more precise evaluation, the economic advantages are verified by means of an investment and operational cost analysis.

2. A proposed scheme alternative

Chemical and petrochemical industries may be highly energy intensive, according to the processes they operate; an important application of petrochemical industry is the petroleum processing, that is done by means of a heat treatment of the raw material for producing sub-products, such as gas, kerosene, liquefied gas, fuel oil among others. A case-study is here presented for the analysis of a Heater associated to the petroleum processing and a natural gas compressing station that takes part of the same industrial complex.

As the exhaust gas has nearly 17% O_2 weight base and a temperature of 460 ^oC, it seems to be interesting to substitute the atmospheric combustion air by part of these hot gases. Two technical constraints must be considered when calculating the amount of gas to be used, the minimum O_2 excess and the maximum allowable flame temperature due to the maximum obtainable inside film temperature in the heaters tubes, which contributes to the heat dissipation.

The evaluation of pros and cons of the alternative economic restrictions must be considered taking into account that all of the decision variables interact with the others, such as the pressure drop in the required duct and burner and the incoming gas temperature in burner; the higher the temperature, the lower the NG consumed on burner, but at the same time, the higher the temperature the larger the duct diameter in order to maintain the outlet GT gases at a reasonable pressure. If this constraint is not controlled, the GT efficiency may decrease to an unacceptable level. The scheme of both, the present installation and proposed alternative, is presented in Fig. (1).

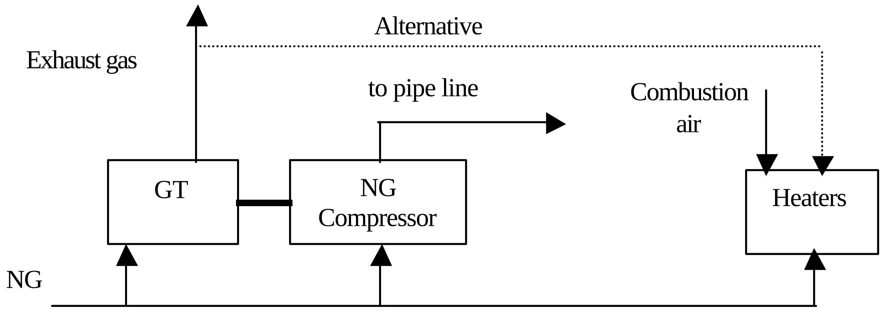

Figure 1. Present installation and proposed alternative

The scheme of Fig. (1) represents a NG compressing station associated to a NG processing plant. The first unit has the finality of compensating the pressure drop in the long GN ducts and is composed mainly by the GT and NG compressor, the second unit produces fuels as liquefied petroleum gas (LPG). The present installation uses NG and air in some equipment, as the Heaters shown in the figure. The alternative, shown in dotted line, consists on taking part of the exhaust mass flow gas to be directed to the Heaters in order to substitute the atmospheric combustion air.

Tables (1) and (2) indicate the operations data considered in this work for the Heaters and GT, respectively.

Table 1. Present operation data of the Heaters

	HEATER 1	HEATER 2
NG consumption (kg/s)	0.0731	0.1583
Excess air	15%	15%
air/fuel mass ratio	19.88	19.88
combustion gas/fuel mass ratio	20.87	20.87
Combustion gas inlet temperature, flame temperature (^oC)	1798	1798
Combustion gas outlet temperature, stack temperature (^oC)	97	320
Product inlet temperature (^oC)	44.7	79.9
Product outlet temperature (^oC)	250	88.8

Table 2. Present operation data of the GT.

NG consumption (kg/s)	0.2515
Exhaust gas flow (kg/s)	18.05
Exhaust gas temperature (^oC)	459
Shaft power at present discharge pressure (kW)	3380
Power loss for each mm. w. c. (kW)	0.3

2.1 Thermodynamics analysis

The following considerations were taken into account for the analysis:

a) The specific heat of gases, Cp, is considered constant for any temperature after burner, both when atmospheric air or hot gases are sent to the combustion chamber.
b) Transport properties of atmospheric air and hot gases are also considered constant.
c) The exit temperature of atmospheric air and hot gases in the heater stack is fixed for each simulation.
d) Eventual heat looses to the environment are invariable.
e) Little alterations of LMTD has not influence on inlet-outlet product temperatures.

An energy balance is then presented, according to Fig. (2).

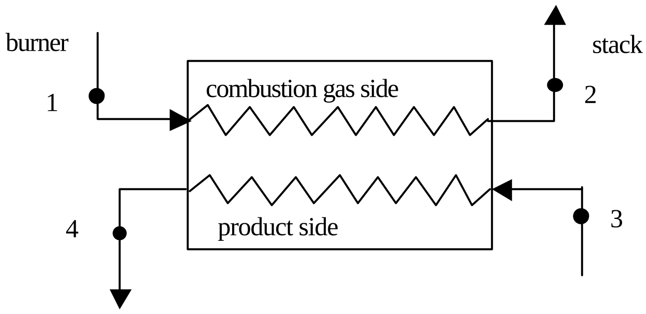

Figure 2. Fired Heater scheme

Equation (1) and (2) represent the heat balance of the existing installation and the proposed alternative, respectively:

$$\dot{m}_g C_{pg}(T_1 - T_2) = \dot{Q}_p \tag{1}$$

$$\dot{m}_{ga} C_{pg}(T_{1a} - T_2) = \dot{Q}_p \tag{2}$$

in which $\dot{m}_g$ and $\dot{m}_{ga}$ are the combustion gas mass flow for the present installation and the proposed alternative, respectively, in kg/s, $\dot{Q}_p$ is the heat delivered to the product, in kW, and subscript "a" indicates the alternative value. Temperatures T are all presented in ^oC.

From Eqs. (1) and (2), Eq. (3) is derived for calculating the necessary gas flow at different flame temperatures.

$$\dot{m}_{ga} = \dot{m}_g \frac{(T_1 - T_2)}{T_{1a} - T_2} \tag{3}$$

Logarithmic Mean Temperature Difference (LMTD) is calculated according to Eq. (4):

$$LMTD = \frac{(T_1 - T_4) - (T_2 - T_3)}{\ln\left(\frac{T_1 - T_4}{T_2 - T_3}\right)} \tag{4}$$

Considering different temperatures for the exhaust gas turbine and the values listed in Tab. (1), the results shown in Tab. 3 are presented relatively to the heaters operational behavior.

Table 3. Heater operation data

		Heater 1		Heater 2	
Exhaust GT temperature (^oC)	Flame temperature (^oC)	Combustion gas mass flow (kg/s)	LMTD (^oC)	Combustion gas mass flow (kg/s)	LMTD (^oC)
Atmospheric air	1798	1.53	441.5	3.30	748.5
227	1608	1.72	400.9	3.79	693.3
327	1668	1.65	413.8	3.62	710.9
427	1751	1.57	431.6	3.41	735.0

The values of exhaust GT temperature were chosen to facilitate the use of enthalpy from formation tables. The temperatures considered for the exhaust gases reveals that the temperature can be diminished by using the gases in a previous application or by diluting it with air injection. The last case may produce some differences on the calculus because of the higher O_2 fraction in the injected air.

2.2 Heat transfer

In this section, a brief analysis of the effects of different LMTD and gas flow on the heaters performance is carried out. The assumptions adopted in 2.1 (b), (c) and (d) are considered to validate the assumption 2.1 (e).

For the gas side operating in off-design conditions it is acceptable to impose the condition (Ganapathy, 1993a, Abdallah and Harvey, 2001):

$$\frac{U}{U_a} = \left(\frac{\dot{m}_g}{\dot{m}_{ga}} \right)^{0,65} \tag{5}$$

in which U is the overall heat-transfer coefficient, $W/m^2.^oC$.

Equation (6) expresses that the heat delivered to the product is the same in all the alternatives:

$$\dot{Q}_p = LMTD \cdot U \cdot A = LMTD_a \cdot U_a \cdot A \tag{6}$$

in which A is the total transfer area in m^2.

Equation (6) can also be wrote according to Eq. (7):

$$LMTD \cdot U = LMTD_a \cdot U_a \tag{7}$$

being necessary to check the consistency of Eq. (5). By applying Eq. (5) and the values of LMTD from Tab. (3), the required check is shown in Tab. (4).

Table 4. Heat transfer deviation

	Heater 1			Heater 2		
Flame temperature (^oC)	$LMTD/LMTD_a$	U/U_a	Total variation	$LMTD/LMTD_a$	U/U_a	Total variation
1798	1	1	1	1	1	1
1608	0.9081	1.0800	0.9807	0.9263	1.0936	1.0130
1668	0.9373	1.0529	0.9869	0.9498	1.0617	1.0084
1751	0.9775	1.0184	0.9955	0.9820	1.0212	1.0028

The values in column “Total variation” are the product of the two columns at left, and they are very close to the unity. If necessary, during the operational procedures, little adjustment in O_2 excess may be enough to obtain the desired response. This fact suggests that it is possible to accept the simplification 2.1 (e).

3. Fuel consumption.

The profit of some alternative can be assured when low NG consumption in the heater burners is obtained. The consumption of NG in the current installation was collected from the data sheet of the equipment and with the informed 15% excess air. The combustion gas mass flow to NG mass flow ratio can be calculated, as shown in Tab. (1). By doing similar balance and substituting the atmospheric air by the combustion GT gases, with 17,5% O_2 weight base, and maintaining the same 15% O_2 excess, the former ratio becomes 25.09 instead of 20.87 of Tab. (1). Then, from Tab. (3), taking the Heater 1 operating with a flame temperature of 1608 ^oC, for example, the NG consumption is expected to be 1.72/25.09=0.0686 kg/s against the current 0.0731 kg/s. The reduction of NG consume is of 6.85%. Similar calculus leads to the results listed on Tab. (5).

Table 5. Reduction on NG consumption

	Heater 1			Heater 2		
Flame Temperature (^oC)	Combustion gas flow (kg/s)	NG mass flow (kg/s)	NG consumption reduction (%)	Combustion gas flow (kg/s)	NG mass flow (kg/s)	NG consumption reduction (%)
1798	1.53	0.073	0	3.30	0.158	0
1608	1.72	0.068	6.85	3.79	0.151	4.43
1668	1.65	0.066	9.59	3.62	0.144	8.86
1751	1.57	0.063	13.7	3.41	0.136	13.92

Flame temperatures were calculated as adiabatic condition for different oxidant gas temperatures, as indicated in Tab. (3).

4. Economic analysis

This analysis accounts by the NG cost and the investment cost for the implementation of the proposed alternative, as well as it considers the effect on GT performance.

4.1 Fuel cost

For the values of Tab. (5), it is possible to calculate the annual cost of NG for each alternative. A typical cost of 3.00 US$/MMBtu (per million Btu) is used in this analysis, which is converted to 0.136 US$/kg when a LHV of 46360 kJ/kg is considered. Table (6) resumes the costs and savings for different alternatives.

Table 6. Annual costs and savings of NG consumption (US$)

	Heater 1		Heater 2		Total
Flame Temperature (^oC)	Annual cost	Annual savings	Annual cost	Annual savings	Annual savings
1798	313090	0	677645	0	0
1608	291645	21445	647623	30022	51467
1668	283067	30023	617601	60044	90067
1751	270200	42890	583290	94355	137245

4.2 Piping cost

The pipe cost may be determined by calculating the effects of pressure drop on the GT performance, for each of the considered combustion gas temperatures and duct diameter. Table (7) describes the necessary calculated data to help the decision-making about the recommended option: annual costs and profits.

Table 7. Pipe costs , pressure drop and additional costs for different exhaust gas from GT and pipe diameters

	Diameter (m)	0.4	0.5	0.6	0.7	0.8	0.9	1.0
Exhaust GT Temperature (^oC)	Total pipe cost (US$)	17800	22100	26500	30800	35200	39600	43900
227	Pressure drop mm.w. c.	553.5	176.5	68.79	30.95	15.49	8.41	4.87
327	Pressure drop mm.w. c.	621.2	198.4	77.33	34.79	17.40	9.00	5.00
427	Pressure drop mm.w. c.	662.2	211.6	82.49	37.10	18.55	10.07	5.83
		Power loss (kW)						
227		166.05	52.95	20.64	9.29	4.65	2.52	1.46
327		186.36	59.52	23.20	10.44	5.22	2.83	1.64
427		198.66	63.48	24.75	11.13	5.57	3.02	1.75
		Additional NG consumption (kg/s)						
227		0.0124	0.00394	0.00154	0.000691	0.000346	0.000188	0.000109
327		0.0139	0.00443	0.00173	0.000777	0.000389	0.000211	0.000122
427		0.0148	0.00473	0.00184	0.000829	0.000414	0.000225	0.000130
		Additional Annual Cost (US$/year)						
227		53000	16900	6590	2960	1480	805	466
327		59500	19000	7410	3330	1670	905	524
427		63400	20300	7900	3550	1780	965	558
		Total annual savings (US$/year)						
227		-1560	34600	44900	48500	50000	50700	51000
327		30600	71100	82700	86700	88400	89200	89500
427		73800	117000	129000	134000	135000	136000	137000

In Tab. (7), the set of data relative to the Additional NG consumption was calculated proportionally to the nominal power and corresponding consumption of the GT. Costs were calculated for 8760 hours per year, 24 hours per day. Total annual savings is the difference between the annual savings in Tab. (6) and the Additional Annual Cost indicated in Tab. (7). It can be perceived that when the pipe diameter is one meter, the effects of the pressure drop on the total profits are negligible, when compared to the results Tab. (6), but the total pipe cost is the biggest. Table (3) indicates the relation between gas temperature and flame temperature.

4.3 Profits estimative

In order to choose one of these alternatives, it is convenient to project the profits of them along with their life. Table (8) shows the results obtained for one and five years.

Table 8. Return of investments (US$/year). Interest 18%/year, n years, Temperature of combustion gas=T (^{o}C)

Diameter (m)	0.4	0.5	0.6	0.7	0.8	0.9	1.0
n=1_T=227	Negative	8445.69	13630.83	12110.93	8458.93	3992.55	-813.23
n=1_T=327	9590.23	44947.77	51412.73	50343.07	46875.96	42493.02	37729.29
n=1_T=427	52840.62	90861.27	98096.43	97299.79	93943.79	89611.34	84872.90
n=5_T=227	Negative	27482.41	36409.51	38640.18	38730.15	38014.34	36959.13
n=5_T=327	24876.38	63984.49	74191.42	76872.33	77147.18	76514.81	75501.65
n=5_T=427	68126.77	109897.99	120875.11	123829.04	124215.01	123633.13	122645.26

From Tab. (8) it is possible to observe that nor the maximum or the minimum diameter are the optimum choice, for the considered scenery. Figure (3) presents these results.

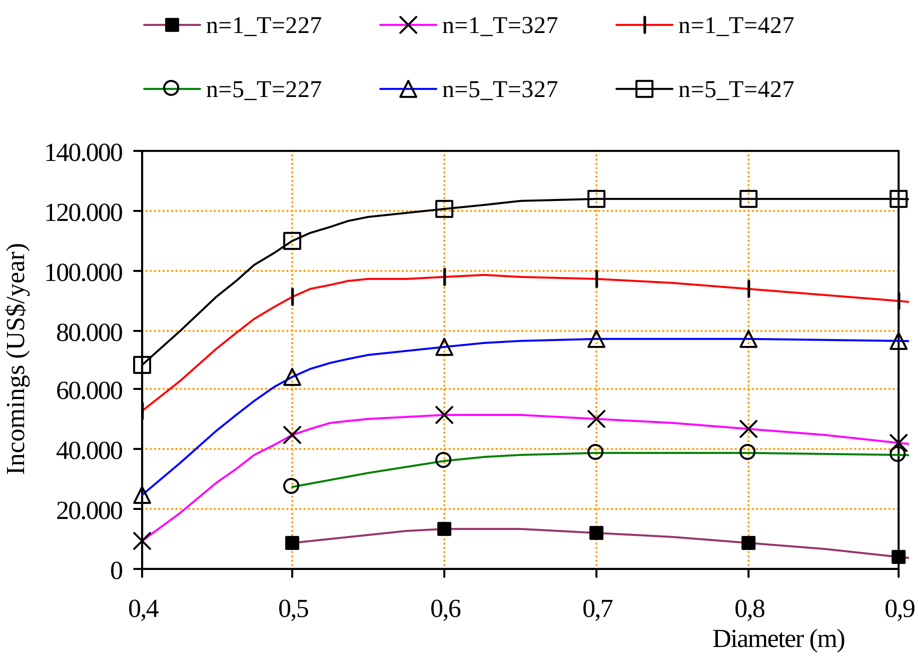

Figure 3. Return of investments (US$/year). Interest rate of 18%/year for a variable "n" years and temperature of combustion gas "T" (^{o}C)

From the former table and Fig. 3 it can be observed that for some diameters the incoming has not a significant increase, and in some cases it decreases. The "optimum" diameter can be approximated to 0,8 m because it maintains the incoming above 5 years.

Another aspect that must be analyzed is when the study considers combined cycle including the installation of a HRSG and a steam turbine. In this case, the steam production of the HRSG is reduced when part of the gases is deviated to the heaters, and the electric power on the steam turbine-generator too. By using a C_p=1,148 kJ/kg.^{o}C for the exhaust gases of GT and 459 ^{o}C for inlet temperature and 200 ^{o}C for stack temperature on the HRSG, depending on the HRSG and steam turbine efficiencies, it can be estimated a production of 640 kW in a steam turbine generator, when 18 kg/s of combustion gas passes through the HRSG producing superheated steam. As the estimate flow for the Heaters is of around 4,8 kg/s, it is expected a reduction of 4,8/18·640=170 kW in electric power production. Considering a typical cost of 60 US$ /MWh for the electricity purchased and that the energy deficit must be acquired from the grid, the additional cost is US$ 89.400 per year. From the Total Annuals Profit of Tab. (7), in this scenario, only the alternative of high temperature remains of interest, if the costs and advantages of the HRSG are compatible.

5. Conclusions

Combined cycles and gas turbine systems to be operated in electric-only mode or as a cogeneration unit have been extensively proposed for several applications, including the chemical and petrochemical industries. This paper showed, however, that the actual profits when using the exhaust gas turbine in heat recovery applications can only be validated after detailed analysis including thermodynamics, heat transfer and fluid mechanics.

Economic aspects are those that will conduct the decision-making process by identifying the best options, and it is necessary a cost estimation procedure the more accurate possible. Profits can be assured for specific values of pipe diameter and exhaust gas temperatures, which means that the design phase must be specially conducted for assuring steady-state values in accordance to the better operational condition. Strategic aspects must also be considered for checking the interactions between all the parts of an installation, actual or proposed.

6. References

Abdallah, H. and Harvey, S.,2001, "Thermodynamic analysis of chemically recuperated gas turbines", Int. J. Therm. Sci., Vol. 40, pp. 372-384.

Ganapathy, V.,1993, "Recovering Heat when generating Power", Chemical Engineering, February, pp.94-98.

Ganapathy, V.,1993a, "Simulation Aids Cogeneration System Analysis", Chemical Engineering, October, pp.27-31.

Vatavuk, W. M. and Neveril, R. B., 1980, "Estimating the size and cost of duct work", Chemical Engineering, December, Vol. 29, pp. 71-73.