ENERGETIC AND EXERGETIC ANALYSIS OF A THERMAL SYSTEM AIMING COGENERATION

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Abstract *This paper presents an analysis focusing two industrial system for energy recovery using process gas. The first one operates as a bottoming cycle and the second as a topping cycle. Both systems are provided with a Heat Recovery Steam Generator – HRSG. The main difference between the systems is the inclusion of an expander instead of a pressure reduction control valve and an orifice chamber, upstream the HRSG. In a first approach the concepts of cogeneration and its modalities are described, taking into account the version of different authors.The operational performance of the boiler in both cases was prior simulated by using EES® and Excel softwares and the impact of including the expander was verified. The simulation also included the temperature profile through the boiler. The boiler simulation is not part of this paper. An energetic and exergetic analysis is done emphasizing the benefits concerning the exergy gain by using topping cycle with the expander. To proceed this kind of analysis the 1*st *and 2*nd *law of thermodynamics are used. Economical profits in using the expander instead of pressure reduction valve and orifice chamber are also briefly commented.*

Keywords*: Exergy, cogeneration, thermal system, expander*

1. Introduction

Considering the scenario of reduction in low cost new hydraulic sources and an increasing on oil prices, the identification and use of hot industrial gases flow to produce power are becoming very important in industries. In this context is defined cogeneration. ANEEL, Brazilian Electricity Regulatory Agency, in the Resolution n° 21 of January, 29th, 2000, defines cogeneration as a combined production of useful heat and power, from chemical energy available on one or more fuels.

In this way, Brazilian oil refinery REGAP has installed in its Fluidized Catalyst Cracking unit, FCC, a recovery system to convert the energy from hot flow gases of the catalyst regenerator vessel in electricity. Usually there are in similar FCC units only a Heat Recovery Steam Generator, HRSG, to produce process steam, in a bottoming cycle.

Passing the gas flow first in an expander before passing in the waste heat boiler, getting power from hot gases, the system changes to a cogeneration Plant in a topping cycle.

Figure 1 shows the schemes of original system, in red, and actual in blue.

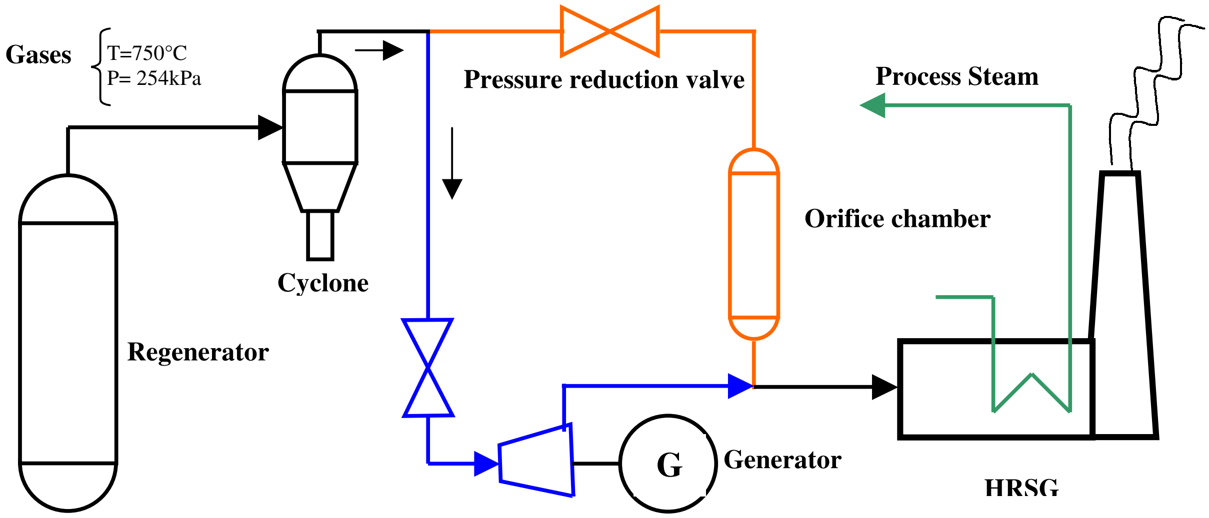

Figure 1. Scheme of the original and new system

The main difference between both is an expander in the flow gases path in parallel with a differential pressure control valve and an orifice chamber.

In the actual system the pressure loss in the expander controls pressure in the regenerator while produces power. According 1st Law of Thermodynamics may be wrongly concluded there is a good use of the energy from flow gases producing only steam in a HRSG. However an analysis based on 2nd Law of Thermodynamics shows a loss of Exergy when producing only steam in a waste heat boiler. In other words, the ability of the system to produce power is definitively lost in the original scheme. So, an Exergetic analysis based on 2nd Law of Thermodynamics is an important tool to complement and consolidate studies of Thermal systems in order not to waste the capability of generating power.

This kind of analysis permits to make a map of opportunities in producing power and detecting irreversibilities in the industrial process.

In the next topics an energy and exergy balance of the system with and without the expander will be done comparing in both cases the efficiencies and losses of energy and exergy. A brief review in the concepts of cogeneration, topping and bottoming cycle will be made and some comments about the economical analysis that have supported the decision to install the expander and a refreshing in some parameters will be made also.

2. Cogeneration and the topping and bottoming cycles

Cogeneration is a wide term to define a simultaneous production of power and heat in a thermal plant. In fact there are many possibilities and several degrees of complexity in an industrial cogeneration plant (Corrêa Neto, 2001). According to PURPA, Power Utilities Regulatory Policies Act, law edited in 1978 in the USA, cogeneration is a combined production of electricity and heat obtained by using energy from a fuel. The main target of cogeneration is to make better the use of fuels, producing power and recovering waste heat rejected from the system to some kind of application in the process. Cogeneration is also a benefit in reducing greenhouse gases as it contributes in the reduction of consumption in fossil fuels as well.

Normally thermal cycles can be classified as topping and bottoming ones. In the topping cycle, in which most of the heat is supplied the waste heat it produces is utilized in a second process which operates at a lower temperature level and is referred as a bottoming cycle (Kehlhofer, 1991). According to Corrêa Neto (2001), the topping cycle is that in which first of all fuel is burned in a thermal machine generating power and the waste heat rejected is used in some way in the process. In the bottoming cycle the rejected thermal energy from industrial processes is used in waste heat boilers producing steam. The steam generated is then used to power a generator.

Kehlhofer exemplifies the advantages of cogeneration and why to combine thermal cycles to improve efficiency based on Carnot maximum efficiency, as follows:

$$\eta_c = (T_w - T_k) / T_w \tag{1}$$

Where η_c is the Carnot maximum efficiency, T_w is the temperature of the heat supplier and T_k is the ambient temperature. Obviously the efficiencies of real processes are lower than Carnot efficiency because of losses involved. In this paper will be emphasized the distinction between energetic and exergetic losses.

Energetic losses are mainly associated to radiation or convection of heat. On the other hand, exergetic losses are caused by irreversible processes accordingly to the 2nd Law of Thermodynamics. The best way to improve process efficiency is reducing these losses increasing the cycle maximum temperature or reducing heat waste at the lower temperature as possible. All this considered becomes clear the advantages in choosing the cogeneration on topping cycle in order to optimize system efficiency minimizing exergetic losses. It is important to remember that from the economical point of view it is determinant evaluate the gap between the Public Utility price of electricity and that generated on site. As higher this difference is more profitable the investment will be.

In this way the installation of the expander in a refinery FCC unit intends to match both targets: lower exergetic losses and reduce the expenses with electricity. The expander selected in this project was a 9,3 MW four stages MAN TURBO, model EH071/150, with a wide range in the inner gas temperature limits and in the size and amount of catalyst particles.

2.1. Exergetic an Energetic analysis

As said before, the tools available to make a consistent analysis taking in account both quantitative and qualitative issues are the 1st and 2nd Law of Thermodynamics. The first law deals with quantities taking in account energy inputs and outputs in an energy conservation idea. By the 1st Law there is no limit to convert heat in power since the total amount of energy remains the same. On the other hand, according the 2nd Law, none thermal machine, real or ideal, in a cyclic process, can convert all input heat supplied in power. Some amount of the heat supplied shall be rejected to a cold thermal energy reservoir. The 1st and 2nd Laws for a control volume can be stated by Eq. (2) and Eq. (3) below:

$$\Delta\dot{E}_{system} + \Delta\dot{E}_{surroundings} = 0 \tag{2}$$

$$\Delta\dot{S}_{system} + \Delta\dot{S}_{surroundings} \geq 0 \tag{3}$$

In these equations $\dot{E}$ and $\dot{S}$ represent respectively the energy and entropy rate of the open system and surroundings. Here below are described the main equations used to proceed the analysis of the thermal system.

Mass balance for a control region:

$$\sum \dot{m}_e - \sum \dot{m}_s = 0 \tag{4}$$

Where $\dot{m}_e$ and $\dot{m}_s$ represents respectively the mass flow to and from the control region.

Steady-flow energy balance (1st Law):

$$\sum \dot{Q}_{vc} - \sum \dot{W}_{vc} + \sum \dot{m}_e h_e - \sum \dot{m}_s h_s = 0 \tag{5}$$

Where $\sum \dot{Q}_{vc}$ represents the total heat flow to or from the control region, $\sum \dot{W}_{vc}$ is the total shaft work to or from the control region and h_e and h_s are the input and output specific mass enthalpy.

Steady-flow Exergy balance (2nd Law):

$$\sum \left[\dot{Q}_r (T_r - T_0)/T_r\right] - \dot{W}_{vc} + \sum \dot{m}_e \varepsilon_e - \sum \dot{m}_s \varepsilon_s = I_{vc} \tag{6}$$

The first term of the equation represents the exergy due to a heat transfer $\dot{Q}_r$ at a control surface temperature T_r to ambient at a constant temperature T_0, $\dot{W}_{vc}$, is the shaft work rate to or from the control region and ε_e and ε_s are the input and output specific mass exergy. The term I_{vc} represents the irreversibility rate or the rate of exergy loss of the system.

The physical exergy of the stream of gases will be calculated according Eq. (7) below:

$$\varepsilon_{ph} = (h_e - T_0 s_e) - (h_0 - T_0 s_0) \tag{7}$$

The first term ε_{ph} represents the physical exergy obtainable when the stream of gas is brought from its initial state to the environmental state defined by pressure P_0 and temperature T_0 (Kotas, 1995). The enthalpy and entropy of gas in initial state is h_e and s_e and h_0 *and* s_0 for the environmental state.

The efficiencies will be evaluated according the following equations:

$$\eta_{vc} = \left[\dot{m}_{steam}(h_s - h_e) + \dot{W}_{vc}\right] / \left(\dot{m}_{gas} h_{ge} + \dot{m}_{water} h_e\right) \tag{8}$$

$$\Psi_{vc} = 1 - I_{vc} / \sum \Delta E_e \tag{9}$$

Where η_{vc} is the energetic efficiency and Ψ_{vc} is the rational efficiency in the control volume. The terms h_s, h_e and h_{ge} are respectively the enthalpy of the superheated steam produced by the boiler, the enthalpy of the incoming water at the boiler economizer and the enthalpy of the flue gas at the regenerator output; $\dot{m}_{steam}$, $\dot{m}_{water}$ and $\dot{m}_{gas}$ are respectively the mass flow rate of steam, water and flue gas and $\Sigma\Delta E_e$ is the total input exergy of the gas stream in the control volume.

2.2. Comments on energetic and exergetic performances of the thermal systems

The original and actual thermal systems are represented by two control volumes in order to define properties in each section as can be seen in Fig. 2 and Fig. 3. Thermodynamical properties of both control volumes are described in Tab. 1 and Tab. 2.

In this section will be discussed the results obtained using the methodology and Equations described in order to compare the energetic and exergetic performance in each control volume. Based on the properties showed in Tab. 1 and Tab. 2 data were calculated using the software EES® making possible to prepare tables with data of each stream on the CV1 and CV2 as appears on Tab. 3 and Tab. 4.

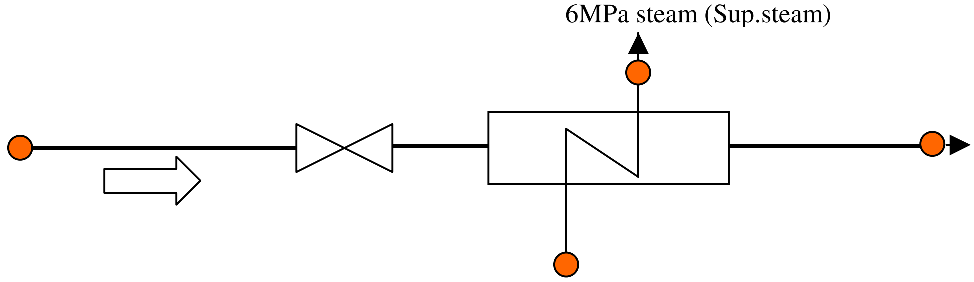

Figure 2. Control volume for the original system – CV1

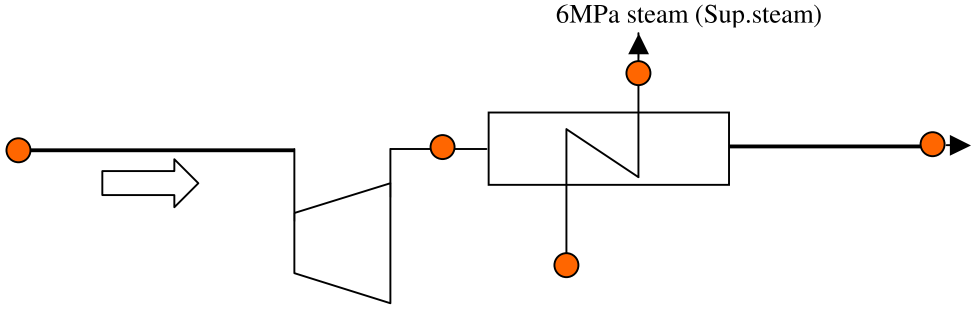

Figure 3. Control volume for the actual system – CV2

Table 1. Properties in the CV1

Local	Description	Temp (^oC)	Pressure (kPa)	Flow $\dot{m}$ (kg/h)	Enthalpy h (kJ/kg)	Entropy s (kJ/kg.K)	Exergy ε (kJ/kg)
1	Regen gas	720	364	127762	945.1	1.755	509
2	Stack gas	285	94	127762	430.2	1.458	82.6
3	BFW	140	8000	26894	594.2	1.731	82.74
4	Sup. steam	394	6000	25500	3161	6.517	1224

Table 2. Properties in the CV2

Local	Description	Temp (^oC)	Pressure (kPa)	Flow m (kg/h)	Enthalpy h (kJ/kg)	Entropy s (kJ/kg.K)	Exergy ε (kJ/kg)
1	Regen gas	720	364	127762	945.1	1.755	509
2	Expander output	552	96	127762	739.5	-	-
3	Stack gas	281.8	94	127762	426.7	1.451	80.93
4	BFW	140	8000	15540	594.2	1.731	82.74
5	Sup. steam	412.5	6000	14800	3209	6.587	1250

As can be seen on Tab. 3, 53.60% of the input exergy of the gas flow in CV1, the original system, was destroyed, and a great part of this, i.e. 21.57%, was lost in the isenthalpic pressure reduction at the control valve and orifice chamber. The exergy of the flue gas at the chimney, 15.69%, was lost by dissipation to the atmosphere and the other losses, 16.34%, were distributed in boiler components as superheater and boiler bank.

According to Modesto & Nebra (2002), the irreversibility at boiler bank and superheater appears due to the large gap on the temperature profiles of the streams of gas and steam across the boiler. The energy losses on CV1 were concentrated in the flue gas at the chimney, 40.20%, and other minor losses were radiation losses at boiler and pipes surfaces.

When comparing Tab. 3 and Tab. 4 it can be seen that there is a large reduction on irreversibility losses in the control volume CV2. In the control volume CV2, the exergy lost in the isenthalpic pressure reduction in the control valve and orifice chamber in CV1 is now converted in power by using the expansion of the gas through the expander. In the same way, part of the exergy of the steam of CV1 is transferred into power reducing once more the irreversibility of the process. In the new composition of the exergetic streams the amount of exergy associated to the superheated steam is reduced from 46.40% to 27.90% and appears power with 39.61%, converted in electricity.

Table 3. Energy and exergy distribution on CV1

VC1 - Original	Energy (kJ/h)	Exergy (kJ/h)	Energy (%)	Exergy (%)
Flue gas from regenerator	$1.207.10^{8}$	$6.503.\ 10^{7}$	88.31	96.69
Boiler feedwater	$1.598.\ 10^{7}$	$2.225.\ 10^{6}$	11.69	3.308
Gas at stack	$5.497.\ 10^{7}$	$1.055.\ 10^{7}$	40.20	15.69
Superheated steam	$8.062.\ 10^{7}$	$3.121.\ 10^{7}$	58.96	46.40
Power	-	-	-	-
Exergy loss at valve/orifice chamber	-	$1.450.\ 10^{7}$	-	21.57
Other losses	$1.09.\ 10^{6}$	$1.099.\ 10^{7}$	0.83	16.34
Total irreversibilities losses	-	$3.605.\ 10^{7}$	-	53.60

Table 4. Energy and exergy distribution on CV2

VC2 - Actual	Energy (kJ/h)	Exergy (kJ/h)	Energy (%)	Exergy (%)
Flue gas from regenerator	$1.207.10^{8}$	$6.503.\ 10^{7}$	92.90	98.06
Boiler feedwater	$9.234.\ 10^{6}$	$1.286.\ 10^{6}$	7.104	1.939
Gas at stack	$5.451.\ 10^{7}$	$1.034.\ 10^{7}$	41.94	15.59
Superheated steam	$4.749.\ 10^{7}$	$1.851.\ 10^{7}$	36.54	27.90
Power	$2.627.\ 10^{7}$	$2.627.\ 10^{7}$	20.21	39.61
Exergy loss at valve/orifice chamber	-	-	-	-
Other losses	$1.664.\ 10^{6}$	$1.120.\ 10^{7}$	1.315	16.90
Total irreversibilities losses	-	$2.155.\ 10^{7}$	-	32.50

Table 5. Irreversibilities and energy and exergy efficiencies for CV1

Control volume	I_{VC} (kJ/h)	I (%)	η energetic (%)	ψ exergetic (%)
CV1(Original)	$3.605.10^{7}$	100	57.8	46.4
Stack flue gás	$1.055.10^{7}$	29.26	-	-
Control Valve and orifice chamber	$1.450.\ 10^{7}$	40.22	-	-
Other losses	$1.099.\ 10^{7}$	30.48	-	-

Table 6. Irreversibilities and energy and exergy efficiencies for CV2

Control volume	I_{VC} (kJ/h)	I (%)	η energetic (%)	ψ exergetic (%)
CV2(Actual)	$2.155.10^7$	100	56.1	67.5
Stack flue gas	$1.034.\ 10^7$	47.98	-	-
Control Valve and orifice chamber	-	-	-	-
Other losses	$1.120.\ 10^7$	51.97	-	-

The total irreversibility losses have changed from 53.60% in CV1 to 32.50% in the actual system, represented by the CV2. However, the exergy loss in the stack remains almost the same in both systems.

Flue gas temperatures in the waste heat boiler stack, obtained by simulations, are very close on both systems. So, the energetic efficiencies on both systems are not so different, staying in 57% approximately. This low efficiency is due to the high stack temperature, a bit higher than 280°C. The 40% range energy losses are concentrated in the stack and 1% are due to radiation losses in boiler and pipes.

The exergetic efficiency goes from 46.4% in the CV1 to 67.5% in CV2, because of the large power conversion in the expander. Table 5 and 6 shows the percentual distribution of irreversibility in each control volume considering each one of the exergy destroyers.

At this point some pictures of the machine and installation will be showed to illustrate the final installation. Figure 4 shows the rotor of the expander at factory in Oberhausen, Germany and Fig. 5 shows the machine being prepared to be transported to site in the process FCC unit.

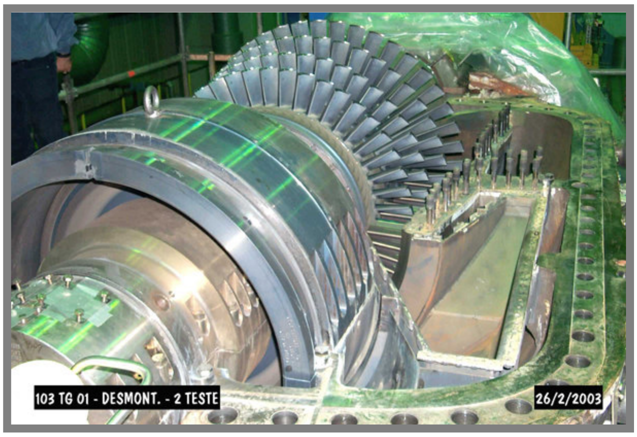

Figure 4. Expander rotor at the factory

Figure 5. Expander being transported to the FCC unit

Finally Fig. 6 shows the expander mounted and ready to operate, close to the 3rd stage separator cyclone.

Figure 6. Expander ready to operate

3. Brief comments on Economical Evaluation

According to Miranda (1999), a business analysis is a complex financial decision requiring to set all costs and benefits involved and their temporal distribution, or the time value of money. It is important to know the rules concerned the cash flow and how to compare quantities out of step on time. It is important to know very well the scenario in which things will be happening. Some questions shall be made about the project: is it technologically adequate? What are the benefits involved with? Is the project aligned with Company strategies? What is the environment impact of its installation? It is also important to define the parameters that will be determinants to check the project feasibility.

In the present case the main economical parameters taken in account were the Net Present Value, NPV and the Internal Rate of Return of the cash flow, IRR. According Ribeiro Júnior (2002) the NPV can be calculated according the Eq. (10).

$$NPV = I - \Sigma[(R_i - C_i) / (1 + r)^i] \quad (10)$$

Where I represents the initial investment, R_i are the benefits of the project at time i, C_i are the costs at time i, r is the interest rate considered for the project and the sum is made for the total periods of the economical life of the project. The meaning of this parameter is to bring to the present time the cash inflows and outflows of the project taking the interest rate in account (the minimum available interest rate for the company) and taking out the investment done. The *IRR* is the interest rate that annuls the *NPV*. Obviously a good project must exhibit a *NPV* higher than zero and an *IRR* higher than the minimum interest rate available for the company.

The main parameters that impact thermal power plant projects are at least the investment, operational cost, the price of the electricity and delays to start normal operation. The cash flow for the expander project had considered the first three events and a sensibility analysis was done in order to determine the weight of deviations in these parameters on final results. Considering that the critical and more expensive components for this project are not manufactured in Brazil is very important to have in mind eventual changes between the relationship on Brazilian and foreign currency mainly because the incomings of the project are in local currency.

It is interesting to prepare a sensibility diagram showing the impact of the main parameters on the profitability of the project to make easier to decide going on or aborting the project. For this particular project it was prepared three different scenarios: Base, Optimist and Pessimist. For each one, deviation on investment, power production, delays, price of electricity from Public Utility, operational costs, fuel price, were made and the impact in the cash flow was calculated. Such diagram is showed in Fig. 7. The figures concerning this project are available in Tab. 7, for the Base Scenario.

Table 7. Economical figures for the Base scenario

Total investment (MM US$)	11.4
Power produced (MW)	8.90
Electricity price (US$/MWh)	46.48
Annual revenue (MM US$)	3.70
Annual operational cost (MM US$)	0.50
IRR (%)	23.00
NPV (MM US$)	3.13
Minimum interest rate (%)	15.00

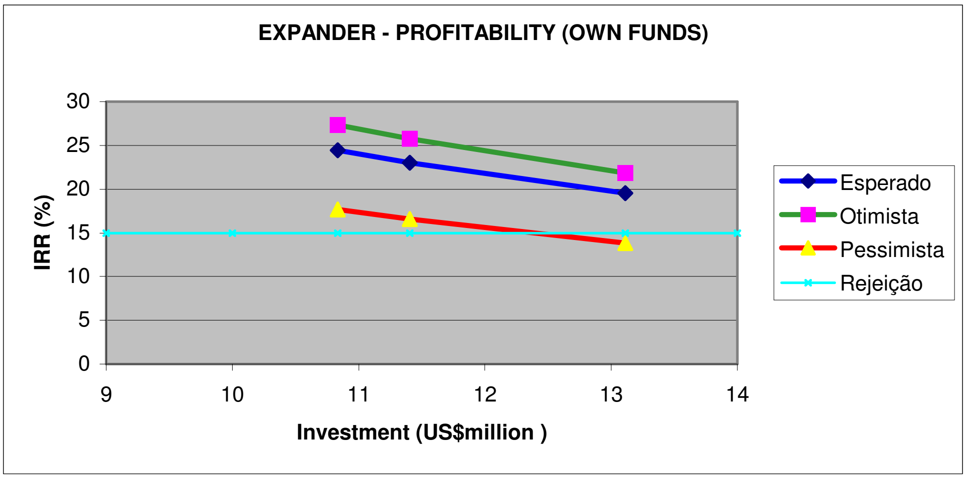

Figure 7. Sensibility diagram

4. Conclusion

The use of an expander instead of a pressure reduction valve and orifice chamber in a thermal system of refinery FCC plants is a good option to minimize exergetic losses and to reduce expenses in the electricity Public Utility bill. However, numerical simulations and operational results point to inexpressive improvement in the energetic efficiency of the thermal system.

From the economical point of view the project is attractive taking the actual prices of electricity paid to Public Utility and a near future scenario of increasing prices in account.

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