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STUDY OF OPTIMIIZING A COMBINED CYCLE OF TWO PRESSURE LEVELS

Bachelor of Science Thesis

Examiner: University Lecturer Henrik Tolvanen
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ABSTRACT

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Examiner: University Lecturer Henrik Tolvanen

The topic of the thesis consists of a case calculation of power plants. The objective is to optimize combined cycle HRSG (Heat Recovery Steam Generator) pressure levels (two system pressures) for optimal efficiency. The **objective of HRSG** is to gain the maximum enthalpy difference at the same time as achieving a reasonable economic cost. If we want to get heat transfer by convection it is necessary to have gases with a higher temperature than the steam in any stage in order to be closer to the pinch points.

To develop **the case**, an example of a power plant formed by combined cycle of two pressure systems has been created. Consequently, the system has been analyze and studied in order to find the possible combinations for two pressure values which is the most optimal in order to enhance its performance.

The **mathematical simulation** has been done with Excel with the corresponding enthalpy functions in order to determinate the function for the maximum efficiency points for each pressure value in order to reach the maximum power generated.

Keywords: Combined cycle, heat recovery steam generator, pressure, efficiency, power, model.

RESUMEN

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El tema tratado se basa principalmente un caso de cálculo en centrales térmicas. El objetivo consiste en optimizar la caldera de recuperación (HRSG) de ciclo combinado en función de sus dos niveles de presión para la mejora de la eficiencia completa del ciclo. **El objetivo del HRSG** es obtener la máxima diferencia de entalpias entre la entrada y salida del ciclo de vapor lo que nos generara más potencia en dicho ciclo, siempre y cuando, al mismo tiempo mantenga un razonable coste económico de construcción y puesta en marcha.

En el desarrollo de **este trabajo**, primero estudiaremos el desarrollo de un modelo de ciclo combinado para dos niveles de presión. A continuación, analizaremos el sistema y estudiaremos las posibles combinaciones de esas dos presiones en la caldera de recuperación para obtener la máxima eficiencia energética.

Para la **simulación** de los cálculos, se utiliza un modelo de ciclo combinado en la herramienta del programa Microsoft Excel para estudiar las curvas de variación de eficiencia donde se han implementado las funciones entálpicas de los flujos de trabajo.

Palabras clave: Ciclo combinado, caldera de recuperación, entalpias, eficiencia, potencia

PREFACE

The knowledge required for this project is mainly based by course in thermal Engineering taught of the Polytechnic University of Madrid in the third academic year, as well as supported by the guiding university lecturer Henrik Tolvanen from the Tampere University of Technology. In addition, Excel was chosen in order to acquire a complete knowledge and understanding of the mathematical processes of combined cycles focusing on HSRG. The vast knowledge of this field helps to better understand thermodynamical process and energy generation of the student.

Pedro Pacios Llorca

Tampere, 25.04.2018

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LIST OF SYMBOLS AND ABBREVIATIONS

ABBREVIATIONS

Symbol	Meaning
AC	Air compressor
BC	Bryton Cycle
CC	Combined Cycle
CW	Condensed Water
EG	Exhaust Gases
GT	Gas Turbine
HP	High Pressure
HSRG	Heat Steam Recover Generator
LP	Low Pressure
RC	Rankine Cycle
ST	Steam Turbine
∆PP	Pinch Point

NOMENCLATURE **THERMODYNAMIC** STUDY.

C_{pa}	Specific heat of air at constant pressure	[kJ/kgºC]
C_{pg}	Specific heat of exhaust gases at constant pressure.	[kJ/kgºC]
$\frac{C^2}{2}$	Pressure drop	[J]
e	Sensible enthalpy.	[kJ/kgºC]
F	Fuel – Air flow ratio.	[-]
h	Enthalpy	[kJ/kg]
h_S	Isentropic Enthalpy	[kJ/kg]
H_C	Specific heat power of fuel	[MJ/kg]
m_a	Air mass	[kg]
$\dot{m}_a$	Flow air mass	[kg/s]
m_f	Fuel mass	[kg]
$\dot{m}_f$	Flow fuel mass	[kg/s]
m_g	Gas mass	[kg]
$\dot{m}_g$	Flow gas mass	[kg]
$\dot{m}_{st}$	Total steam flow	[kg/s]

$\dot{m}_{st\ LP}$	Low pressure steam flow	[kg/s]
$\dot{m}_{st\ HP}$	High pressure steam flow	[kg/s]
N	Net power	[kJ]
N_{Br}	Brayton cycle net power	[kJ]
N_{GT}	Gas turbine net power	[kJ]
N_c	Compressor net power	[kJ]
N_{Rk}	Rankine cycle net power	[kJ]
N_{ST}	Steam turbine net power	[kJ]
N_P	Pump net power	[kJ]
P	Pressure	[bar]
P_{amb}	Environment pressure	[bar]
q	Specific heat power	[kJ/s]
q_{cc}	Specific heat power of combustion chamber	[kJ/s]
q_{HSRG}	Specific heat power that HSRG provides	[kJ/s]
Q	Net heat power	[kJ]
Q_{Br}	Net heat power in Brayton cycle	[kJ]
Q_{Rk}	Nest heat power in Rankine cycle	[kJ]
Q_{cc}	Net heat power of combustion chamber	[kJ]
Q_{eg}	Net heat power of exhaust gases	[kJ]
Q_L	Heat loses between cycles coupling	[kJ]
Q_{HSRG}	Net heat power that heat steam recover generator provides	[kJ]
Q_{LP_eco}	Heat transferred in LP Economizer	[kJ]
Q_{LP_eva}	Heat transferred in LP Evaporator in HSRG	[kJ]
Q_{LP_sup}	Heat transferred in LP Superheater	[kJ]
Q_{HP_1eco}	Heat transferred in first part HP Economizer	[kJ]
Q_{HP_2eco}	Heat transferred in second part HP Economizer	[kJ]
Q_{HP_eva}	Heat transferred in HP Evaporator	[kJ]
Q_{HP_sup}	Heat transferred in HP Superheater	[kJ]
q	Specific heat power	[kJ/kg]
Sc	Control surface	[m^2]
s	Entropy	[kJ/kgºC]
T	Temperature	[ºC]
T_S	Isentropic temperature	[ºC]
U	Intern energy	[kJ]
v	Flow velocity	[m/s]
v_c	Control flow velocity	[m/s]

V_c	Control volume	[m^3]
w	Power indicated	[kJ/kg]
w_{Br}	Power indicated in Brayton cycle	[kJ/kg]
w_{Rk}	Power indicated in Rankine cycle	[kJ/kg]
W	Specific power	[kJ/s]
W_{Br}	Specific power of Brayton cycle	[kJ/s]
W_{GT}	Gas turbine specific power	[kJ/s]
W_C	Compressor specific power	[kJ/s]
W_{Rk}	Specific power of Rankine cycle	[kJ/s]
W_{ST_LP}	Steam turbine specific power for low pressure circuit	[kJ/s]
W_{ST_HP}	Steam turbine specific power for high pressure circuit	[kJ/s]
W_{SP}	Pump specific power kJ	[kJ/s]
S_C	Surface control	[m^2]
v	Flow velocity	[m^2/s]
T_S	Isentropic temperature	[ºC]
T	Temperature	[ºC]
U	Internal energy	[kJ]
Δ	Variation	[-]
ρ	Density	[kg/m^3]
ν	Specific volume	[m^3/kg]
η	Efficiency	[-]
η_{Cycle}	Overall cycle efficiency	[-]
η_{Br}	Brayton cycle efficiency	[-]
η_{Rk}	Rankine cycle efficiency	[-]
η_{CC}	Combustion chamber efficiency	[-]
η_S	Isentropic efficiency	[-]
η_m	Mechanical efficiency	[-]
η_{mTr}	Mechanical efficiency of transmission	[-]
τ	Tension matrix	[Pa]

1. INTRODUCTION

In general, a combined cycle can be defined as the integration of two or more power cycles in order to achieve greater efficiency in power generated in each individual cycle. However, the great progress made on gas turbines has created a denominated combined cycle which means a system composed of one or several gas turbines, a recovery energy system and a steam turbine. This results in the combination of the Brayton and Rankine cycles (LM Antón p. 101).

Since Industrial Revolution, society has been experiencing a high growth and development through the massive use of fossil fuels such as coal and petroleum; they are able to provide the necessary energy for this current progress.

However, nowadays it is a reality that fossil fuel reserves are limited and close to being exhausted. This fact will slow down world development which is continuously demanding more energy. In addition to this problem, there is the greenhouse effect created by the exhaust gas emissions mainly of NO_X, SO_2 and CO_2, which is progressing to levels as never known before.

In this context, governments have invested in renewable energies to avoid this problem. Therefore, it is necessary to keep another form of energy to satisfy consumer demand. This is why, combined cycle technology gains importance. Due to his high flexibility, efficiency and low emissions, it looks a viable alternative that is able to supply the growing demand, while producing a lower environmental impact.

The idea to use one cycle, which operates at high temperatures in order to feeding another on lower temperatures, was originated one century ago. Emmet, in 1925, thought the idea to introduce two combined cycles using mercury as work flow. However, the gas turbine development and the setting up on Rankine cycle with water-steam were which allowed the present combined cycles.

The first combined cycles for power plants where built in United States the seventeenth century. They usually used coal as a fuel and it got a 5% efficiency increase.

During 1970s, gas turbine was continual development, increasing the power and getting over the maximum possible temperature. In this case, the exhaust gases on the Brayton cycle had enough energy to produce steam in the Rankine cycle.

In the 1980s, the advance in gas turbines keeps progressing; therefore the power plants reached 40%-50% efficiencies and 600 Mw of power.

In the 1990s, the “F Generation and G Generation” gas turbines were created. They are composed by materials that are able to support higher temperatures and able to reach efficiencies from 55% to 60% (Electricity History, Endesa).

Currently, the electric market liberation was the reason that systems using lower investment costs were redirected in order to increase the company´s competitive edge. Other factors included climate change concern and the sustainability concept. (ICADE publication, Energy sector Liberation, 2012).

In this way, the majority of developed countries having increased power generation through renewable sources (Carry Elliot, The Guardian, 2015). On the other hand, technology has developed low CO_2 exhaust gases, which decreases atmospheric contamination.

The figure shows how the energy production has been developed, it is possible to appreciate the increase use of renewable energies during the last few years.

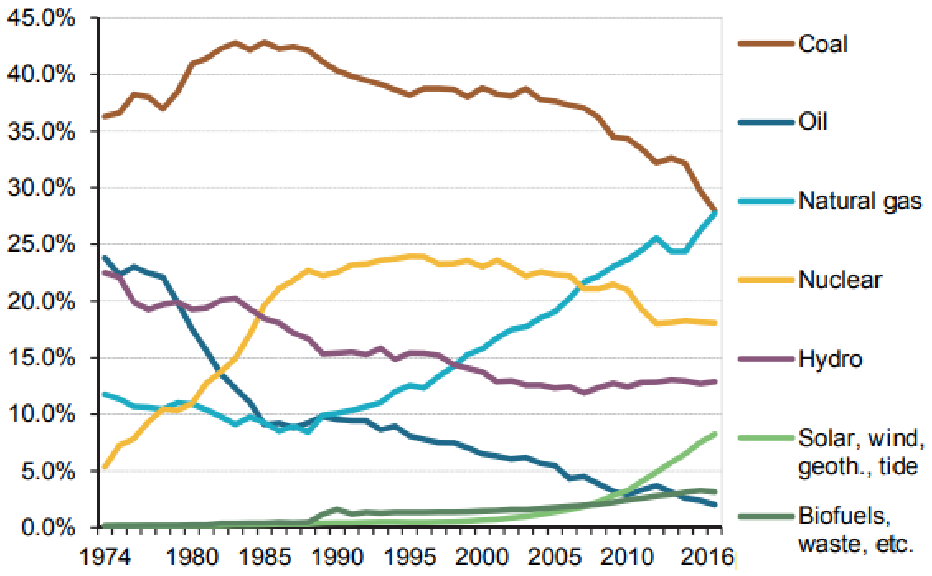

Figure 1. *Global electricity production by source. (Webstore IEA)*

However, Coal (28%), Natural Gas (27%) and Nuclear (18%) are 73% of the total electricity power generated. From this, it is possible to appreciate that the part generated by power plants is still approximately 75% of the total amount.

In this context of energy efficiency and low emission gases match the combined cycle. This fulfils a lot part of the currently increasing electricity demand.

(IEA, International Energy Agency), (History of the Power, Abby Harvey with Sonal Patel, 2017) and (CIA stats)

2. CONCEPTS OF COMBINED CYCLES

The cogeneration technology through “combined cycle” is fundamentally based on the use of two generation power cycles. One of them, at a higher level ,usually Brayton cycle, provides energy to another lower cycle level ,Rankine cycle. In this way, it is easier to achieve a greater benefit from the fuel input on the first up level cycle. (LM. Antón chapters 2, 3).

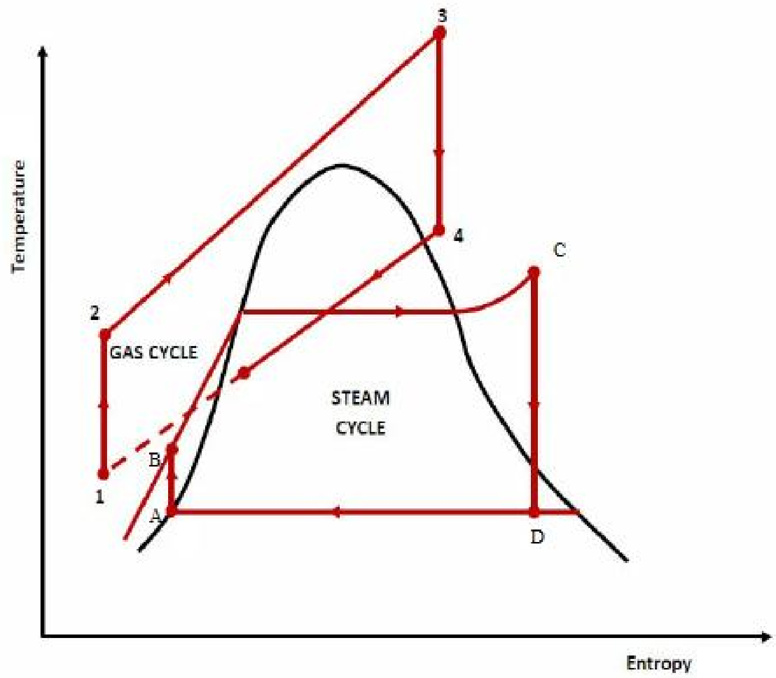

Figure 2. *Combined Cycle T, s diagram (T-E, Charles Mborah 2010)*

In [Figure 2], it is possible to visualize how a high level thermal cycle, numbered 1, 2, 3 and 4, provides energy to a low level, in this case designated by the letters A, B, C and D.

2.1 Brayton Cycle

In [Figure 3], a work scheme of the cycle is shown and includes the relative points and as many of the fundamental elements which compose it:

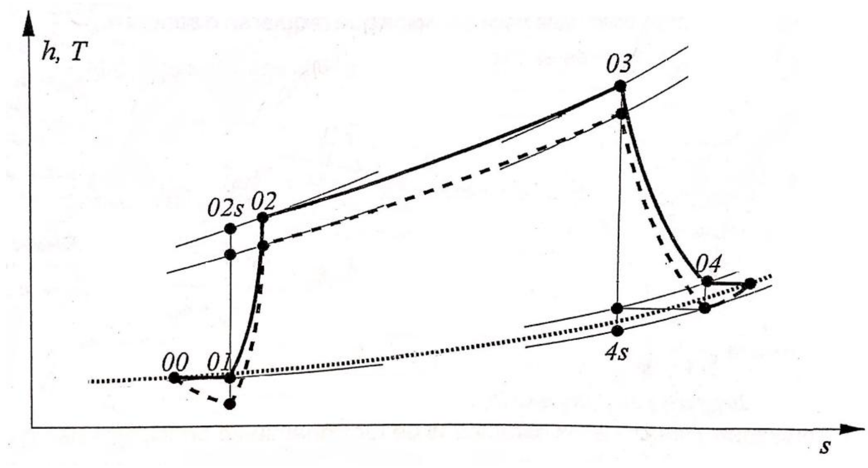

Figure 3. *Brayton simple cycle. T, h – s diagram (LM Antón p. 82).*

The fundamental cycle elements are: air compressor, combustion chamber, gas turbine and electrical generator.

The system is composed of five physical steps:

Admission between points (00 – 01). Air intake from the environmental conditions. Isentropic compressive flow into the compressor points (01 – 02). It produces a pressure increase to achieve an easier combustion process. Heat is added in an exchange heater or combustion chamber which maintains constant pressure points (02 – 03). Isentropic expansion in turbine (03 – 04). In order to produce power from the fluid. It has the same physical stage as the compressor. Exhaust gases (04 – 05) from the thermal engine to the environment or for use in cogeneration. Equations governing the process during the Brayton cycle:

First, based on the general Conservation of Energy equation:

$$\frac{d}{dt}\int_{Vc}\rho\left(e+\frac{v^2}{2}+U\right)dV+\int_{Sc}\rho\left(e+\frac{v^2}{2}+U\right)(\vec{v}-\vec{v}_c)\vec{n}*dS=$$
$$=\int_{Sc}\bar{\bar{\tau}}*\vec{n}*\vec{v}*dS+\int_{Vc}QdV-\int_{Sc}\vec{q}*\vec{n}*dS$$
$$\rightarrow\ w+q=(h_2-h_1)+\frac{1}{2}2({v_2}^2-{v_1}^2)$$

[1]

Implementing assumptions mentioned above to the equation [1], the net power developed by the cycle is the difference between the power generated in the turbine and the power consumed by the compressor:

$$W_{Br}=W_{GT}-W_C=\dot{\mathrm{m}}_g[(h_{03}-h_{04})-(h_{02}-h_{01})$$

[2]

From the last expression it is possible to calculate the power developed by mass unit:

$$\mathrm{w_{Br}} = \mathrm{w_{GT}} - \mathrm{w_C} = (\mathrm{h_{03}} - \mathrm{h_{04}}) - (\mathrm{h_{02}} - \mathrm{h_{01}}) = \mathrm{c_{pg}}(\mathrm{T_{03}} - \mathrm{T_{04}}) - \mathrm{c_p}(\mathrm{T_{02}} - \mathrm{T_{01}}) \quad [3]$$

The heat provided to the cycle by mass unit is:

$$\mathrm{q_{cc}} = \mathrm{q_{02-03}} = \mathrm{c_p}(\mathrm{T_{03}} - \mathrm{T_{02}}) \quad [4]$$

Therefore, it is possible to calculate the basic Brayton cycle efficiency:

$$\eta_{\mathrm{Br}} = \frac{\mathrm{W_{Br}}}{\mathrm{Q_{cc}}} = \frac{w_{Br}}{q_{cc}} = \frac{\mathrm{c_p}(\mathrm{h_{03}} - \mathrm{h_{04}}) - \mathrm{c_p}(\mathrm{h_{02}} - \mathrm{h_{01}})}{\mathrm{c_p}(\mathrm{T_{03}} - \mathrm{T_{02}})} \quad [5]$$

The efficiency of the combustion chamber is:

$$\eta_{\mathrm{cc}} = \frac{q_{Br}}{\mathrm{Q_{CC}}} = \frac{\mathrm{c_p}(\mathrm{h_{03}} - \mathrm{h_{04}}) - \mathrm{c_p}(\mathrm{h_{02}} - \mathrm{h_{01}})}{\mathrm{c_p}(\mathrm{T_{03}} - \mathrm{T_{02}})} = \frac{(\dot{\mathrm{m}}_\mathrm{a} + \dot{\mathrm{m}}_\mathrm{f}) * \mathrm{h_{03}} - \mathrm{h_{02}} * \dot{\mathrm{m}}_\mathrm{a}}{\dot{\mathrm{m}}_\mathrm{f} * \mathrm{H_c}} = \frac{(1 + \mathrm{F}) * \mathrm{h_{03}} - \mathrm{h_{02}}}{\mathrm{F} * \mathrm{H_c}} \quad [6]$$

Being:

$$F = \frac{\dot{\mathrm{m}}_\mathrm{f}}{\dot{\mathrm{m}}_\mathrm{a}} \quad [7]$$

Using the isentropic relation it is possible to know the pressure variations in the pump and in the turbine:

$$\frac{\rho_0}{\rho} = \left(\frac{T_0}{T}\right)^{\frac{\gamma-1}{\gamma}} \rightarrow r = \frac{p_2}{p_1} = \frac{p_3}{p_4} \rightarrow \frac{T_2}{T_1} = r^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4} \quad [8]$$

It is important to mention the following points in order to calculate the results of the Brayton cycle:

- Compression and expansion are adiabatic, so it is necessary to consider the entropy variations.
- The combustion is not ideal so more net power from the combustion chamber is necessary.
- The kinetic energy variation of work flow between intake and output in each element is not considered. For some cases, it is necessary to have to consider due to the high velocities of the turbo-machines

- There are no drops in pressure.
- The mass flow occurs under constant specific heat and a fixed composition.
- It is possible to avoid the fuel contribution of the work-flow because it is similar to the air extraction from the compressor to refrigerate the first turbine blades. The amount of air is approximately 1 – 2 %.

For isentropic variations in the condenser and the gas turbine it is used the following equations:

$$\eta_{SC} = \frac{|T_{02S} - T_{01}|}{|T_{02} - T_{01}|} \qquad \eta_{S\,GT} = \frac{|T_{03} - T_{04}|}{|T_{03} - T_{04S}|} \tag{9}$$

(PPCC, Theoretical and Project, 2006, p. 55) and (Douglas Quattrochi 2006, MIT, Brayton Section)

2.2 Rankine Cycle

This cycle works with condensable work-fluid. Usually, the fluid is demineralised water. The fluid evolution is a close circuit which is composed of the following stages:

First of all, there is the fluid expansion in the steam turbine among points (3 – 4), after that, the condenser works with the fluid energy at constant pressure points (4 – 1). Next will be Stage to achieve higher fluid pressure in the pump with liquid water (1 – 2). Finally, increased temperature process (2 – 3) produces steam from the water, in this case through the HSRG. And at this point, cycle starts again, having the steam at the same beginning conditions. The next picture represents the thermal conditions steps under the saturation steam line.

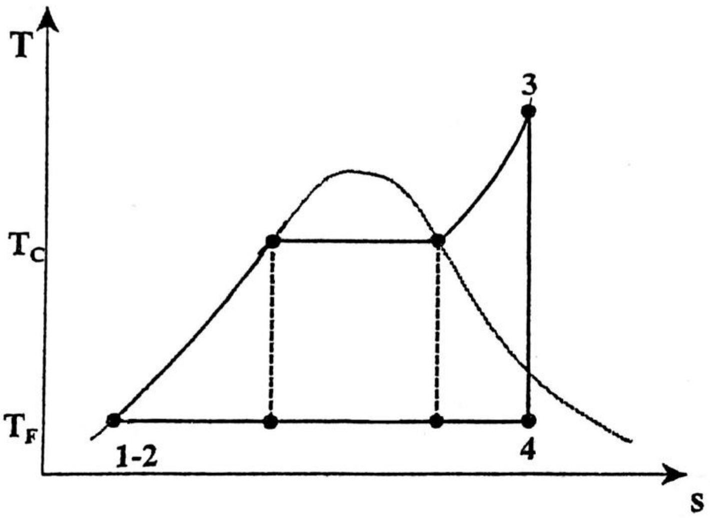

Figure 4. *Rankine Simple Cycle, T – s diagram (LM Antón p.18).*

Following the above process for the steam turbine, the next steps show the calculations for this cycle:

Power generated by the turbine:

$$W_{ST} = \dot{m}_{St}(h_3 - h_4) \qquad [10]$$

Specific power consumption by the pump:

$$W_{ST} = \dot{m}_{St}(h_2 - h_1) \qquad [11]$$

In this way, the specific power generated by the cycle is the difference between the power generated by the turbine and the power consumed by the pump.

$$W_{Rk} = W_{ST} - W_P = \dot{m}_{St}[(h_3 - h_4) - (h_2 - h_1)] \qquad [12]$$

The heat uses by the cycle expressed by the following equation:

$$\mathrm{q_{Rk}} = \mathrm{q_{2-3}} = \dot{\mathrm{m}}_{\mathrm{St}}(\mathrm{h_3} - \mathrm{h_2}) \qquad [13]$$

And the efficiency expression of the Rankine cycle is:

$$\eta_{\mathrm{Rk}} = \frac{\mathrm{W_{Rk}}}{\mathrm{Q_{Rk}}} = \frac{w_{Rk}}{q_{Rk}} = \frac{(\mathrm{h_3} - \mathrm{h_4}) - (\mathrm{h_2} - \mathrm{h_1})}{(\mathrm{h_3} - \mathrm{h_2})} \qquad [14]$$

Basically, in the Rankine cycle, the specific power consumed by the pump (1 – 2) is zero with respect to the specific power generated by the steam turbine (3 – 4).

As in the first cycle, there are the following considerations:

- Compression in the pump and expansion in the turbine are irreversible processes, so it is necessary to consider the entropy variations.
- There are variations in the specific heat and in the adiabatic coefficient during the process which affects the overall efficiency.
- The combustion is not ideal so more net power from the combustion chamber is necessary.

For isentropic variations in the condenser and the gas turbine it is used the following equations:

$$\eta_{sP} = \frac{|h_{2S} - h_1|}{|h_2 - h_1|} \qquad \eta_{s\,ST} = \frac{|h_3 - h_4|}{|h_3 - h_{4S}|} \qquad [15]$$

(PPCC, Theoretical and Project, 2006, p. 66) and (Douglas Quattrochi 2006, MIT, Rankine Section)

2.3 Brief Analysis of Each Element

This part gives a brief description of each single physical component of the combined cycle.

Compressor: It is a thermal machine which increases the pressure through mechanical twist using external power.

Combustion chamber: It is defined as the element where combustion reaction takes place. The fuel is combined with the air from compressor in order to feed the turbine in the Brayton cycle while in Rankine cycle the combustion reaction is external to the flow.

Turbine: It is a thermal machine where power is produced by thermal energy from the flow.

Steam Pump: It transforms mechanical power into pressure energy.

Condenser: In the Rankine cycle, it is where the flow yields heat causing the steam condensation. Conversely, it is not necessary in Brayton cycle (LM Antón p. 21, 72).

2.4 Combined Cycle System

The cogeneration technology through "combined cycle" is fundamentally based on the use of two generation power cycles. One of them, a higher level (usually Brayton cycle, operating between 1000 – 1500ºC) provides energy to another lower cycle level (Rankine cycle, operating between 300 – 600ºC). In this way, GT - ST, it is easier to achieve higher performance from the fuel input on the first higher level cycle.

This cycle combination can have advantages and disadvantages. Usually, the power generation of each combined cycle is two out of three generated by the gas turbine and one out of three by the steam turbine. The next general aspects are characteristic of combined cycle.

The combined cycles GT – ST require a high quality fuel which consequently is more expensive. Although it is possible to use low quality fuel it will significantly increase complications and costs during installation. Also, the start-up time, initial investment and the installation size are greater than GT but significantly lower than ST. Related to the pollution; emissions are similar to the GT. However specific values are lower and significantly lower than the ST due to fuel quality (LM Antón p. 102).

In the following figure, the T-s work scheme of both cycles shows how the energy is provided from the high system to lower.

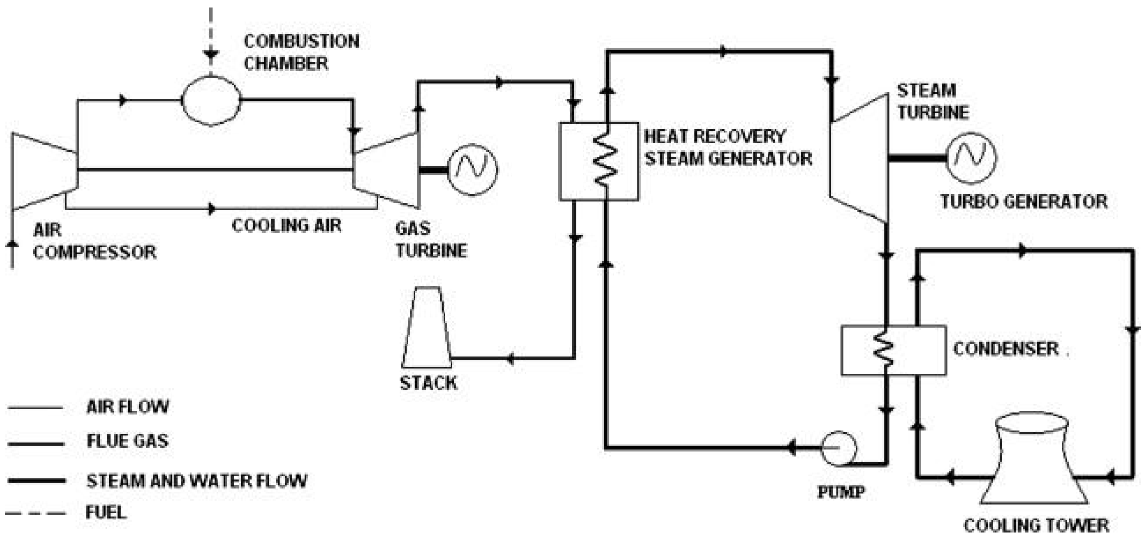

Figure 5. *Combined cycle elements configuration (Researchgate GTA 2014)*

[Figure 5] shows the connexion between the elements of the system to the generator, and the [Figure 6] shows the thermal feature for each point in the system.

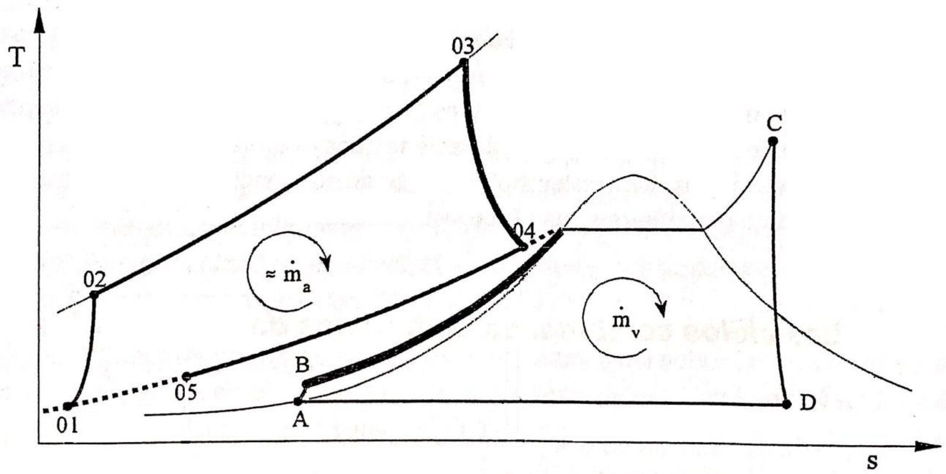

Figure 6. *Combined Cycle T-s diagram (LM Antón p. 104)*

It can be seen how the gas cycle on the left, defined by the points 01, 02, 03, 04 and 05, operates in higher temperatures than the steam cycle on the right, defined by points a, b, c, d. This residual heat is useful to generate power in medium-low pressure. The combining of two cycles drives the general system which increases the global efficiency more than it would achieve by running each cycle separately. Therefore, the global efficiency depends on each cycle´s efficiency and the capability to transfer the energy between both systems (LM Antón p. 103).

The HSRG efficiency is defined by:

$$\eta_{\mathrm{HRSG}} = \frac{\mathrm{Q_{Rk}}}{\mathrm{Q_{Br}}} = \frac{\mathrm{Q_{RBr}} - \mathrm{Q_L}}{\mathrm{Q_{RBr}}} = 1 - \frac{\mathrm{Q_L}}{\mathrm{Q_{RBr}}} \quad [16]$$

Then, using in the combined cycle definition and the individual cycle efficiency:

$$\eta_{\mathrm{CC}} = \frac{\mathrm{W_{Br}} - \mathrm{W_{Rk}}}{\mathrm{Q_{Br}}} \quad [17]$$

It results:

$$\begin{aligned}\eta_{\mathrm{CC}} &= \frac{\mathrm{W_{Br}} - \mathrm{W_{Rk}}}{\mathrm{Q_{Br}}} = \eta_{\mathrm{Br}} + \frac{\mathrm{W_{Rk}}}{\mathrm{Q_{Br}}} = \eta_{\mathrm{Br}} + \frac{\mathrm{W_{Rk}}}{\mathrm{Q_{Br}}} * \frac{\mathrm{Q_{Rk}}}{\mathrm{Q_{Br}}} = \eta_{\mathrm{Br}} + \eta_{\mathrm{Br}} * \frac{\mathrm{Q_{Rk}}}{\mathrm{Q_{Br}}} \\ &= \eta_{\mathrm{Br}} + \eta_{\mathrm{Rk}} * \frac{\mathrm{Q_{Rk}}}{\mathrm{Q_{RBr}}} * \frac{\mathrm{Q_{RBr}}}{\mathrm{Q_{Br}}} = \eta_{\mathrm{Br}} + \eta_{\mathrm{Rk}} * \eta_{\mathrm{Br}} * \frac{\mathrm{Q_{RBr}}}{\mathrm{Q_{Br}}} \\ &= \eta_{\mathrm{Br}} + \eta_{\mathrm{Rk}} * \eta_{\mathrm{HRSG}} * \frac{\mathrm{Q_{Br}} - \mathrm{W_{Br}}}{\mathrm{Q_{Br}}} \\ &= \eta_{\mathrm{Br}} + \eta_{\mathrm{Rk}} * \eta_{\mathrm{HRSG}} * (1 - \eta_{\mathrm{Br}})\end{aligned} \quad [18]$$

Finally, from an economical point of view, it is interesting to know the net power final values. Also, it is important general performance which is affected for both cycles. So to estimate the final power net values and the overall efficiency, it uses the following equations:

Net power generated by gas turbine:

$$N_{GT} = (W_{st} - W_{SP}) * \eta_{\mathrm{mG}} \quad [19]$$

Net power generated by steam turbine:

$$N_{ST} = W_{st} * \eta_{\mathrm{mST}} - \frac{W_{SP}}{\eta_{\mathrm{mTrP}} * \eta_{\mathrm{mP}}} \quad [20]$$

Net power used by combustion chamber:

$$Q_{CC} = \frac{q_{cc}}{\eta_{\mathrm{cc}}} \quad [21]$$

Overall efficiency of the whole cycle:

$$\eta_{cycle} = \frac{N_{GT} + N_{ST}}{Q_{CC}}$$
[22]

In the general combined cycle system, it is possible to verify that increasing the average temperature at the turbine inlet and reducing the average temperature at the combustion chamber and condenser inlet leads to an increase in performance.

Another way to optimize efficiency is by decreasing the temperature differences between exhaust gases at the gas turbine outlet and the steam-cycle inlet in the HSRG, as in the examples of cycles mentioned earlier.

(PPCC, Theoretical and Project, 2006, p. 135) and (Douglas Quattrochi 2006, MIT)

2.5 Heat steam Recovery Generator

HSRG is the fundamental element that joins both cycles. Its function is to recover the heat power provided from the exhaust gases in the gas turbine in order to feed to the steam turbine of at desirable condition.

The main difference between the conventional chambers is the mechanism of heat transfer, because in this case, there are significantly higher temperatures where radiation transfer due to combustion is dominant. But in the HRSG case, the transfer mechanism usually operates in a conductive manner. Therefore, the recovered amount of heat is controlled by the amount of energy available from the gases. It consists of exchange flows where the subcooled water intakes contacts outside exhaust gases (TFG, Joaquin Corredera 2016 p. 16). The basis to classify the HRSG depends on the physical ways that are possible to transfer heat:

Conduction: The heat transfer between two bodies at different temperatures without an exchange of material between them. Or also, through *Convection:* In this heat transfer system, a movement of fluid transports energy between two bodies.

It is possible to find the first classification in function of the number of pressures in the HRSG there are a different number of elements. For each pressure level there is one economizer, one evaporator and one superheater. This classification is the main object of this project. It is because, during the evaporation process, the **steam temperature is constant** and is relative to the steam pressure, which means that the enthalpy variation also depends on the steam pressure and therefore the power of the steam turbine, chamber efficiency and moisture exiting the turbine depends on the steam pressure as well. The different stages and parts for one pressure level are represented in the following figure:

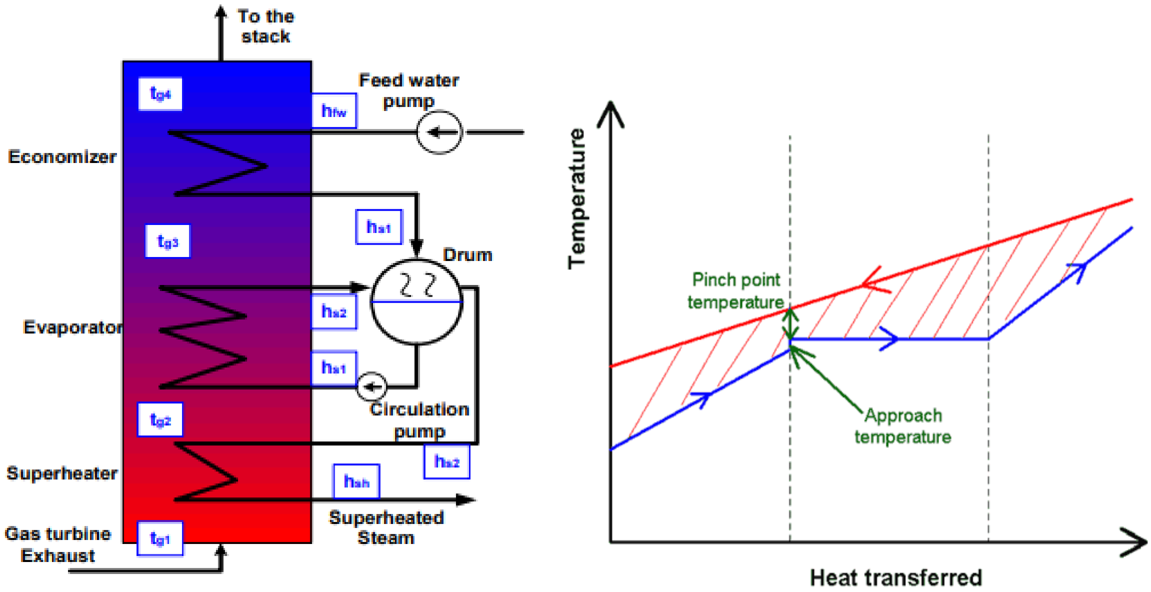

Figure 7. *Stages in HSRG one pressure level and elements (kth portal).*

The application of each element involved is:

The Economizer. It is the first part of the heat exchanger. The cold water enters, propelled by the circulation pumps which increase the flow temperature next to the saturation point without exceeding it. The temperature difference between the saturation of subcooled water and the maximum temperature of the economizer is called the Approach Point or Pinch Point (Δpp), whose optimum values are from between five to ten degrees. The objective of this safety margin is to avoid possible steam formation in the economizer, but only in the case of partial loads on ST.

Steam Drum. It is a cylindrical deposit where the water, near to the saturation situation, comes in both liquid and steam phases. The liquid phase is recirculated to the evaporator.

Evaporator. Inside, the heat transfer process to reach evaporation of the work-fluid takes place. After this stage the fluid is recirculated to the steam drum.

Superheater. The steam derived from the steam drum is again recirculated to the HSRG which is located next to where the hot exhaust gases are released. This is in order to reach the maximum possible temperature before being driven to the ST.

Heaters with 2 or 3 pressure levels are common. In the next figure, for two pressures, it is possible to see how the steam function is closer to the gas line compared to the last model for 1 pressure which results in a thermal improvement (TFG, Joaquin Corredera 2016 p. 13, 15).

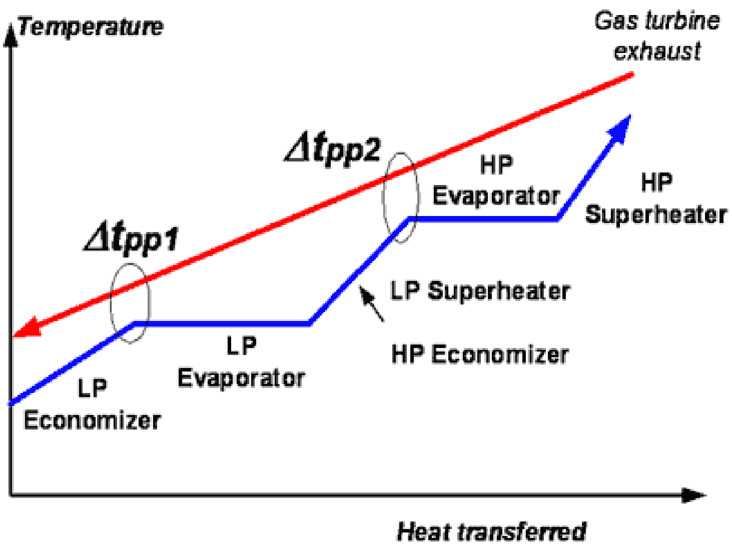

Figure 8. *Heat transfer for two pressure levels (kth portal 2008).*

Dividing the flow in two pressures, there is a first part for the low pressure where saturation point is lower than the second saturation point. It allows the low pressure temperatures to be kept close in the final exhaust gas stage. Simultaneously, as higher pressure has a greater saturation point, the higher temperatures that are reached in the gas turbine will be closer to the superheater HP and with a higher saturation pressure point.

The second main classification of HRSG depends if there is a post combustion chamber or not: An HSRG without a post combustion chamber is the most common case in combined cycles and its use has been increasing during the last few years as a result of the new designs which allow higher exhaust gas temperatures. Essentially, it is a heat exchange where the heat is transferred from the steam cycle exhaust gases by convection. A system with post-combustion includes a heater intake pipe. This utilizes excess O_2 from the exhaust gases depending the building conditions.

Finally the third possibility is regarding to the direction of exhaust gases there are horizontal or vertical HRSGs (2012-2016 Bright Hub Inc).

The horizontal exchange chamber has a horizontal duct for the exhaust gases. The exchange pipes are positioned vertically and the heat transfer takes place by normal circulation. In design conditions, it is important to take care not to exceed the amount of lost gases under 3 kPa.

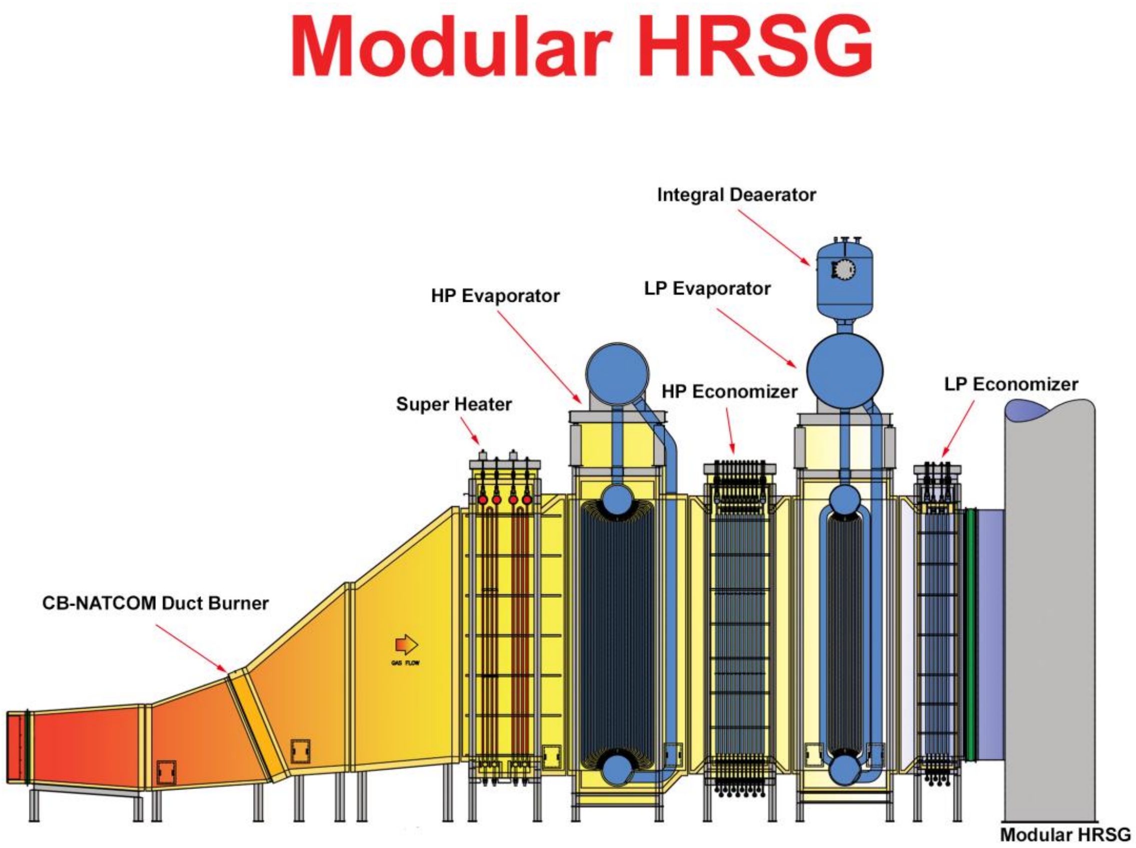

Figure 9. *HSRG horizontal example (Wikipedia HRSG).*

The vertical heat recovery system has a structure composed of containers where water, not vaporized, is collected. In this case, the tubes dilate easily and are not under critical stress and therefore, they are more accessible to maintenance. Here, building design limits are between 2 and 3 kPa.

For Operations, equations governing the heat transfer during the heat exchange process are:

The enthalpy conservation equation for the amount of gases helps to calculate what exact amount of heat is transferred in each stage of the HSRG. The mass flows, gas temperatures and steam enthalpies is possible are the necessary values for operate the heats in the economizer, evaporator and the superheater.

$$\sum \Delta h_i * \dot{m}_{j_{flow\ 1}} = \sum \Delta h_i * \dot{m}_{j_{flow\ 2}} \quad [23]$$

To calculate total heat transfer deducting from the last equation:

$$Q_{eg} = \sum Q_{eco} + \sum Q_{eva} + \sum Q_{sup} \quad [24]$$

To calculate heat from gas:

$$\Delta Q = \dot{m}_{eg} * C_{pg} * \Delta T \qquad [25]$$

To calculate heat from steam:

$$\Delta Q = \dot{m}_{st} * \Delta h \qquad [26]$$

(KTH educational portal 2008), (Douglas Quattrochi 2006, MIT).

2.6 Cycle configurations

Usually, in the gas-steam combined cycles there are several gas turbines feeding a single or double steam turbine through their respective HSRG and therefore, it is possible to find a classification in function of the main configurations.

This **Single – Axis** type has low reliability because it produces power through one single generator.

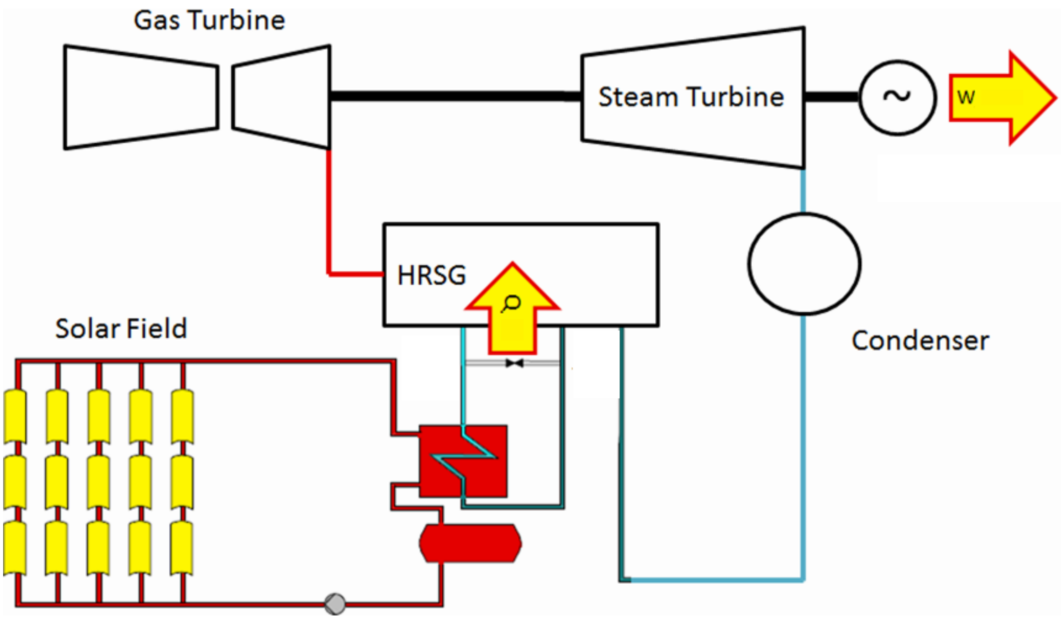

Figure 10. *Analysis of integrated solar combined cycle, single-axis (Renovetec).*

On another hand, using **Multiplex – Axis** configuration, each generator can feed different electrical transport systems with different voltages. Mainly, there are two cases;

Configuration 1x1 (One gas turbine and one steam turbine). There is more gas turbine availability because it can work separately from the steam turbine. The maintenance of turbine generators is easy. But it requires a higher initial investment to buy two different generators and transformers. On another hand, there is **configuration 2x1** (Two gas turbines and one steam turbine). The initial investment is lower, the operational flexibility is higher and it provides greater efficiency. There are also configuration types 3x1 and 4x1. (Renovetec, SL 2009-2018).

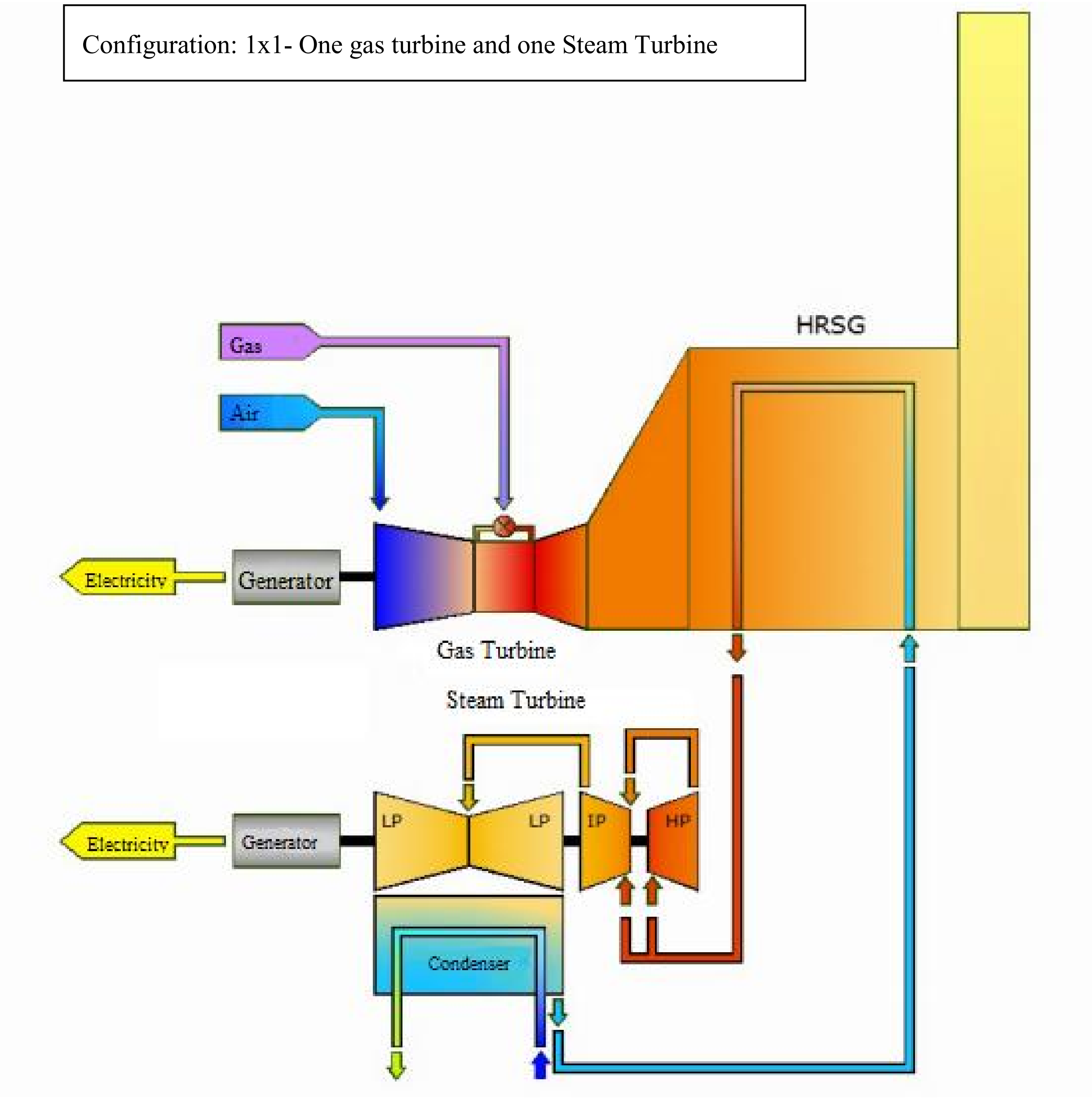

Figure 11. *Multi-axis configuration 1x1*

Configuration: 2x1- 2 gas turbine feeding one Steam Turbine

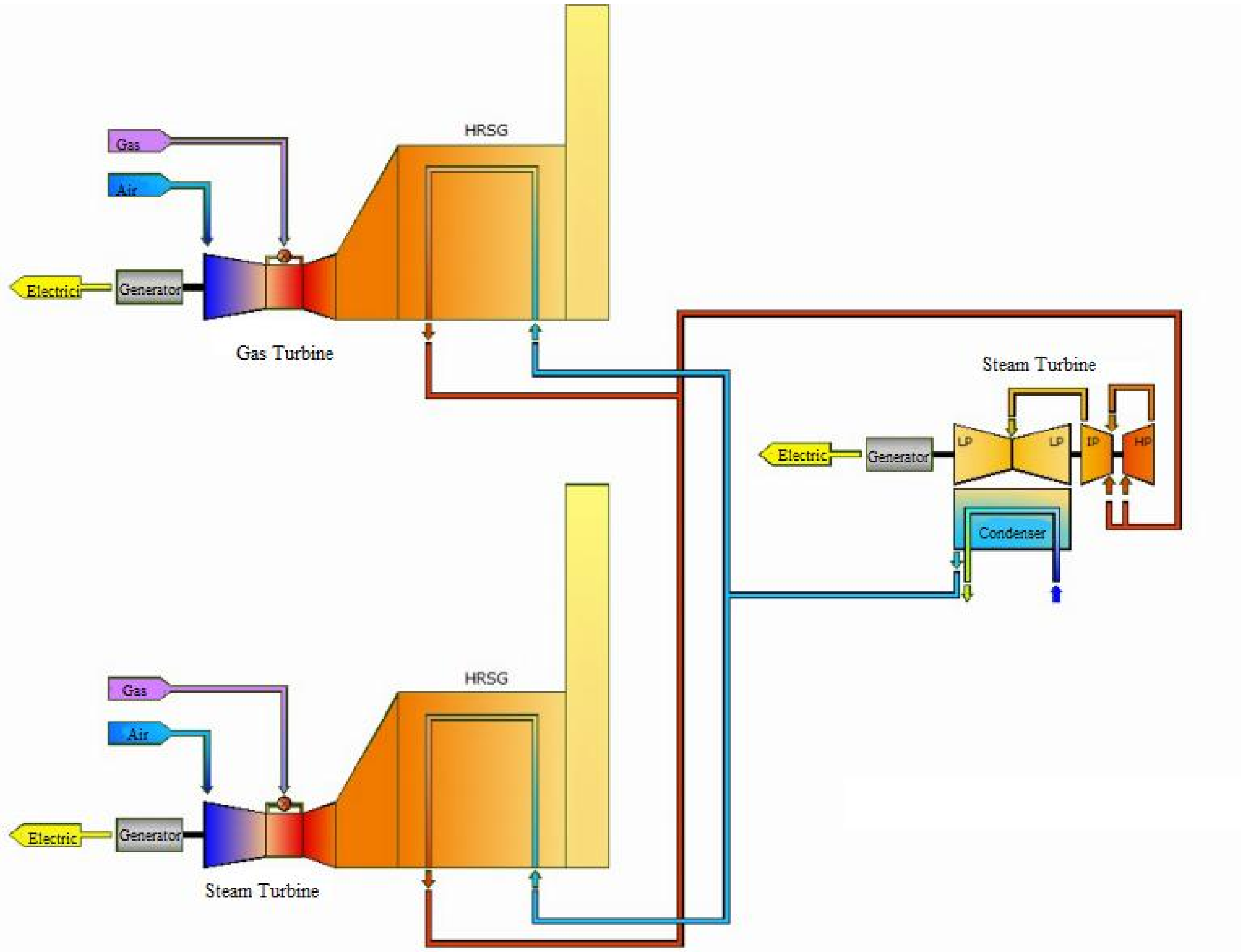

Figure 12. *Multi-axis configuration 2x1*

(Combined power plants, CCGT, J.H. Horlock, Cambridge, figures) and (PFG, Carlos Diaz, p.48).

3. EXAMINED CYCLE CONFIGURATION

This chapter consists in description, design and elements of the combined cycle which is used for the simulation. Based on the power must accomplish by turbine here, there are the aspects to considering.

3.1 Description of studied Model

The model design works in order to get 264 Mw of power in the gas Turbine and 128 Mw in the steam turbine with the appropriate general efficiency of combined cycles. In this case it tries to reach 60% efficiency. It is a medium-small power size adapted for this project (UBA p. 13 2008). The functional plan of the model is:

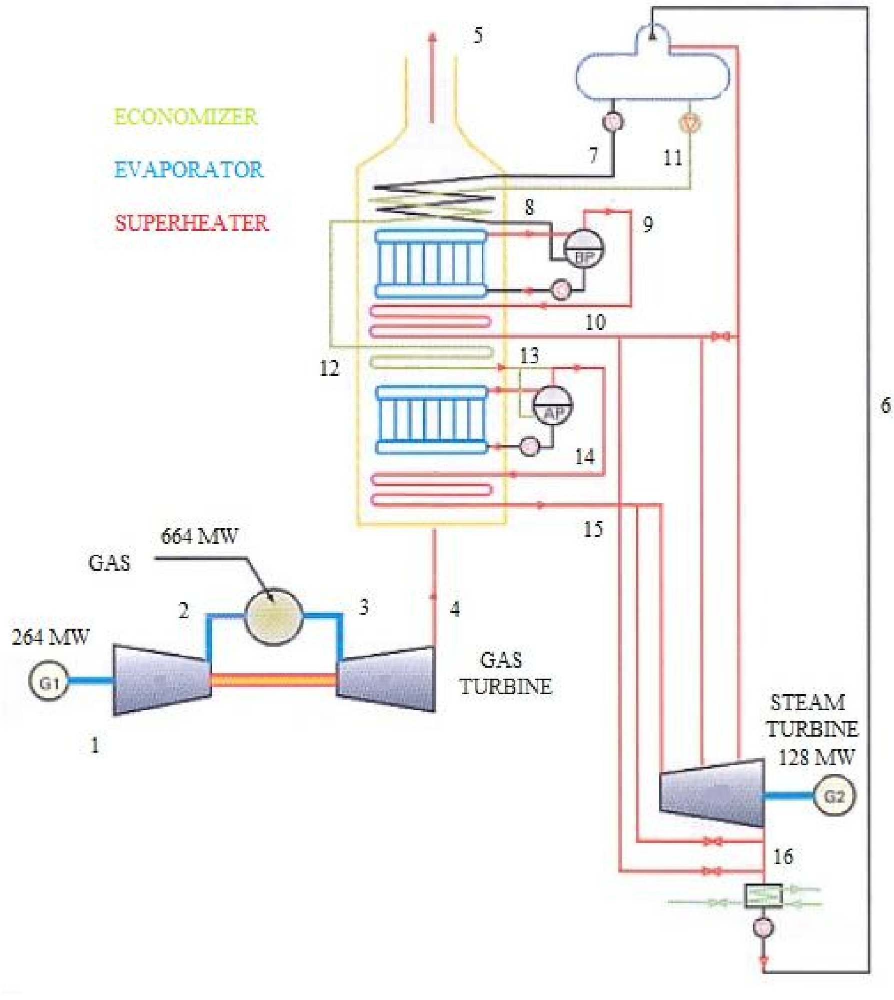

Figure 13. *Singular combined cycle, elements numbered (UBA p.8 modified I)*

The main elements are the compressor (1) and the gas turbine (2) for a gas cycle following the simplest configuration. The HRSG has a horizontal structure and the pressure circuits are represented by the following process:

The exhaust gases that come from the gas turbine go from point 3 and point 7. Therefore, in point 7 the lowest temperatures are found while point 3 takes the maximum temperature the heat can transfer, always respecting the pinch points. Finally, the cold steam flows (14, 15) arrive at point 7.

The low pressure will achieve the saturation temperature and it is driven to the steam drum (9), where work-flow evaporates through the evaporator and afterwards it is sent to the first superheater. At this stage, it is ready to be driven to the steam turbine (10). As the graphic shows, the low pressure steam has to be joined at the medium turbine step in order to have both flows at the same conditions (temperature and pressure).The second circuit is to heat the high pressure flow. In this case, the circuit is the same, but there is a significant increase of the saturation temperature due to higher pressure and this is why this system is positioned closer to the higher exhaust gas temperature. Point (5) shows the second economizer, (8) is the steam drum, (4) is the evaporator and point 3 shows the final temperature arrived at the high pressure circuit.

Steam turbine has several steam intake pipes. For this case, only the two main pipes (one for each pressure) are taken into account. Therefore, the steam turbine has two different entrance of the warm flow, one per each pressure. The pressure valves (16, 17) are to adapt the pressure to the next pipe, and the gas regulator (13) is located in front of to the pumps.

3.2 Fixed Values and Design Conditions

Once decided the net power generated, the second considerations for designing a power plant are the limit conditions of the elements involved. First, it is necessary focus on gas cycle, because, as it is described, it will provide the energy for the steam cycle. Therefore, the limits of the elements to taking in account are:

The compressor imposes two conditions. The capacity to aspirate air indicates the amount of fuel needed in relation to the quality of fuel and the amount of air. Also, the compressor-ratio (ρ) indicates the maximum pressure to design pipes thickness. **Maximum inlet turbine temperature.** The value should not be higher than 1500ºC in order to avoid the maximum NO_x which is damaging to the environment and as well as, the high temperature could damage the first gas turbine parts. This limit is essential in this kind of cycle, because will directly affect the amount of heat from the exhaust gases. **Turbines.** For design this model, turbines have been selected from General Electric supplier. The type chosen is 9F.04 with a maximum possible power of 288 Mw. For steam turbine is chosen model STF-A200 inside part of NON-REHEAT 200 Series which maximum possible output range is 90-220 Mw. And the turbine has representative steam conditions of 140 bar and 565ºC for combined cycle. (GE products website).

3.3 Fuel Natural Gas

Combined cycle power plants often use natural gas as fuel because it causes the lowest environmental impact compared to other fuels. Some of his pros and cons are:

Currently, due to the technology, the flame temperature is low so the Natural gas does not produce sulphur during combustion. Sulphur is one of the most damaging components for the environment as it causes acid rain. However, one disadvantage is transport. Natural gas can only be transported under specific pressure and temperature conditions. That means an increase in price (Fener 2012, Albert Mitja).

3.4 Model Simulation: Boundary Conditions and Starting Values

The objective of this chapter is to describe the design conditions and starting values for numerical calculations of the system model. The cycle is combined and it has two pressure levels in the HSRG and multi-axis turbine configuration (1x1). Each point calculated can be visualized in the following scheme:

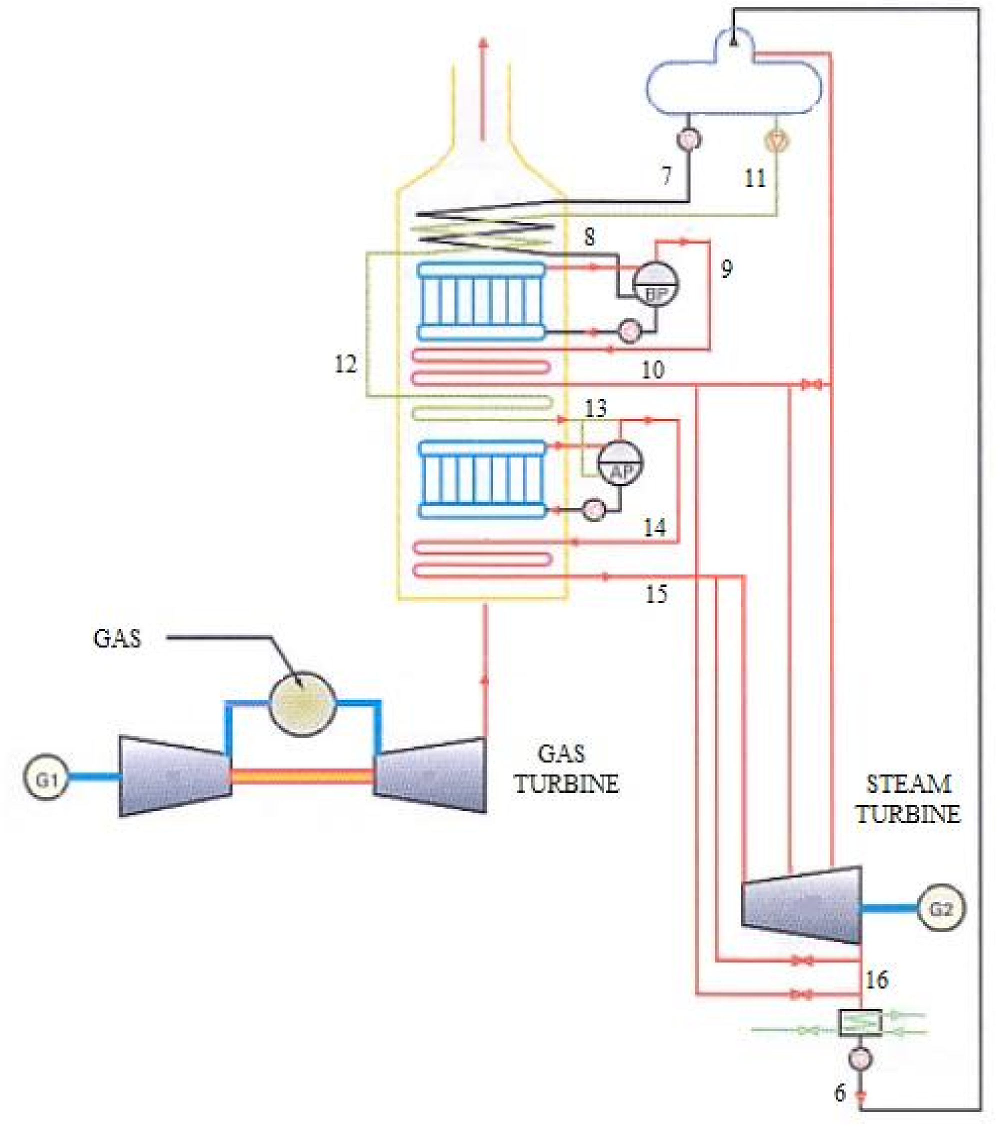

Figure 14. *Combined cycle model defined by points (UBA p.8 modified II)*

As in the theoretical part is mentioned, it is necessary define the boundary physical conditions that there are to having in account in order to resolve the mathematical simulation. These considerations are chosen getting an appropriate approximation to a real power plant case, although those real cases are more complex and they have high number of measure elements in each stage of the system. Then the considerations for each cycle are:

As it in mentioned above, for Brayton cycle:

The kinetic energy variation of work fluid between input and outlet of each element is despicable. There are not load losses during the expansion process and the mass flow remains constant.

On another hand for Rankine cycle:

The Kinetic energy variation of work fluid between input and outlet of each element is despicable. There are not pressure losses in the conducts except in the exchangers and the steam flow remains constant during the different stages in the steam turbine.

In both cycles, the expansion and compression processes are not isentropic and therefore it is necessary to consider a coefficient of performance called isentropic efficiency. It is also necessary to take into consideration the mechanical components which have their own losses which also affect the general performance. Each efficiency value is specified in the next section.

In addition it is also necessary define the starting data table shows the starting data used for the numerical simulation of the cycle. All this is pooled in the following [Figure 15] and [Figure 16].

Constant Values

1	Ambient pressure	$P_1 = 1$ bar
2	Average ambient temperature	$T_1 = 15ºC$
3	Specific heat of air	$C_{pa} = 1.0035$ kJ/kgºC
4	Specific heat of exhaust gases	$C_{pg} = 1.0100$ kJ/kgºC
5	Ideal Monatomic heat gas ratio	$\gamma = 1{,}4$

Gas Cycle Values

6	Compression ratio	$\rho_0/\rho = 14$
7	Temperature outlet of the Turbine	$T_4 = 625$ ºC
8	Temperature outlet of HSRG	$T_5 = 80ºC$
9	Air suctioned by compressor for combustion	$\dot{m}_a = 603$ kg/s

Steam Cycle Values

10	Condenser pressure	$P_6 = 0.01\ bar$
11	Outlet pump low pressure	$P_7 = 6\ bar$
12	Outlet pump high pressure	$P_{11} = 106\ bar$
13	Outside HSRG Temperature (Low Pressure)	$T_{10} = 320$ºC
14	Outside HSRG Temperature (High Pressure)	$T_{15} = 510$ºC
15	Low pressure mass flow	$\dot{m}_{st\ LP} = 15.25$ kg/s
16	High pressure mass flow	$\dot{m}_{st\ HP} = 86.34$ kg/s

Figure 15. *Starting cycle values*

Efficiencies

1	Chamber combustion efficiency	$\eta_{cc} = 97\%$
2	Isentropic gas turbine efficiency	$\eta_{sGT} = 90\%$
3	Isentropic compressor efficiency	$\eta_{sC} = 81\%$
4	Turbine - compressor mechanical efficiency	$\eta_{mG} = 98\%$
5	Steam Generator Efficiency	$\eta_{HRSG} = 98\%$
6	Isentropic Steam Turbine Efficiency	$\eta_{ST\ HP}/\ \eta_{ST\ LP} = 85\%$
7	Mechanical Steam Turbine Efficiency	$\eta_{mST} = 98\%$
8	Pump Mechanical Efficiency	$\eta_{mP} = 95\%$
9	Pump Mechanical Transmission Efficiency	$\eta_{mTrP} = 92\%$
10	Isentropic Pump Efficiency	$\eta_{SP} = 80\%$
11	Isentropic Steam Turbine Efficiency	$\eta_{sT} = 85\%$
12	Mechanical Steam Turbine Efficiency	$\eta_{mST} = 98\%$

Figure 16. *Efficiencies of each component*

3.5 Gas Turbine Calculations

In this chapter the values for the gas turbine are calculated. Therefore, the heat not exploited is used to feed the steam turbine. As mentioned in, the exhaust gas temperature is fixed to achieve the power required in the steam turbine as the model requires.

Firstly, the air is suctioned into the compressor. This will be compressed according to the equation [8]. Taking into account the isentropic compressor efficiency, the temperature after expansion is calculated by equation [9]. Now, it is necessary to reach the inlet turbine temperature required in order to reach the minimum temperature for the exhaust gases. This is possible through the relation between temperatures and pressures before and after turbine stage. It is resolved building a system of equations with equation [8] for the turbine expansion and [9] for the turbine isentropic efficiency.

At this point, it is possible to make an efficiency relation in the combustion chamber in order to obtain the fuel-air relation to reach the temperature necessary while it is taking into account the capacity to suction air from the compressor following equations [6] and [7]. Finally, obtaining the final specific power and efficiency matches the equations [2] and [5].

3.6 Steam Turbine Calculations

In order to obtain the results for the steam cycle, first, it is necessary to obtain the amount of heat that HSRG is going to provide from the exhaust gases. It is calculated by equations [23] and [25].

The operations to calculate the steam transfer follow the next stages:

Using the thermal steam tables (pressure in bar and temperature in ºC) it is possible to know what temperatures the steam must acquire in order to evaporate. These steam values are fixed in function of the pressure. Then, as the steam amount is fixed too, the heat required will not change. As can be supposed, in the end of first economizer, the temperature of the low pressure superheater (T_{10}) has to be the same as the saturation temperature of high pressure saturation liquid (T_{13}). Moreover it is taking in account that pinch points have to be fixed.

Therefore, initially it is calculated the first economizer heat points and it is considered the HP mass flow as any random value. So, solving the equation [24], it is possible to calculate the LP mass flow making a relation between first stage and third in the HRSG parts. After that, it is calculated each heat difference in the LP economizer, LP evaporator, LP superheater, HP economizer and HP evaporator following the equation [25] for each element. Now, there is a value for the second pinch point which is random. To resolve that, it is used the iteration method in order to approximate the second pinch point to 25º (Rice, I.C., 1980) choosing as variable the HP flow mass.

At the final point of HSRG, it is possible to determine the amount of heat there is for the high pressure superheater. Consequently, the amount (in terms of heat) can be determinate by the equation [24]. Therefore, it is possible to calculate the maximum temperature reached by the HP superheater (T_{15}) using equation [26]. The temperatures are checked during whole process through an implemented function called T_HSRG. In this way, the simulation attached in with the project allows change any variable in yellow colour, in order to analyzer the final efficiency

Secondly, once the thermal values from HSRG and the starting values above mentioned in [Figure 15] are known, it is possible to calculate each enthalpy of each point using the thermal steam tables. For this model there are two circuits and therefore, there are two pumps for each circuit.

To determine the points (P_{11}, P_7, P_{16}), it must be noted that enthalpies have to respect isentropic efficiencies. Equations in the pumps and the turbine for both circuits are [15] where is made a relation between points 6 – 7 and 6 – 11 for LP, and the relation between points 10 – 16 and 15 – 16 for HP.

To calculate the power generated by the turbine through both circuits. There are two stages in the turbine. First, the high pressure steam and then during the expansion process the low pressure steam reaches the appropriate turbine stage. According to the equations there are specific power generated by turbine, calculated by equation [10], where it is necessary taking into consideration two specific powers, for LP and HP flows. Secondly there are powers consummated by pumps which are calculated by equation [11].

3.7 Global Values

Finally, from an economical point of view, it is interesting to know the net power final values. So, it uses the equations [19] for net power generated by gas turbine, [20] for net power generated by steam turbine, [21] for net power consumed by combustion chamber and the overall efficiency of the cycle is operated by equation [22].

3.8 Material in use for Simulation

During the development of the project and apart from the sources used, the use of different field software has been required. In order to simulate the different thermal and mechanical processes the following software has been used:

Figure 17. *Excel logo (Wikipedia Excel)*

Microsoft Excel is a spreadsheet developed by Microsoft. There is an extensive variety of uses. There is more specific software for power plant design, but Excel was chosen in order to acquire a complete knowledge and understanding of the mathematical processes, not just simulate the thermal model. As other specific software like Cyclepad or EES. It features calculation, graphic tools, pivot tables, and macro programming language. it has been implemented in a variety of thermodynamic tables.

Another indispensable resource was the thermal data. The source data is collected from **Ascet Engineering**, which is an American Society of Certified Engineering Technicians.

Figure 18. *Ascet Engineering logo (Wikipedia Ascent)*

Tablas Janaf is thermal data source from a website and it has been programmed as a function in Visual Basic in Excel. It provides the necessary thermal values from each single molecular element in combustion (NIST 2013).

Finally, the **Mollier Diagram** was used for manual calculations or in order to check results. This graph-toll is useful to obtain the thermal properties of the steam manually.

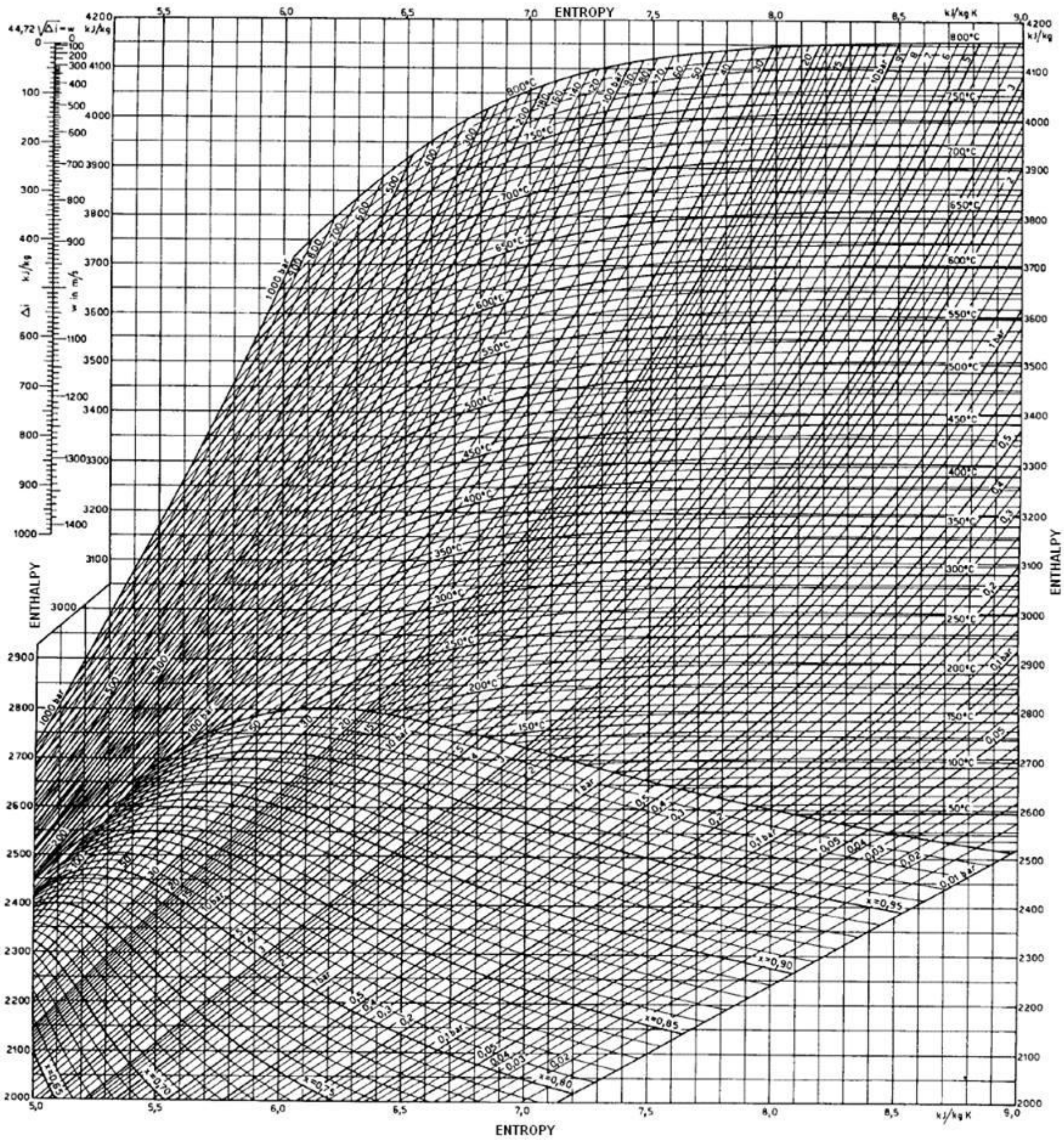

Figure 19. *Mollier Diagram. (ASME, 2010).*

4. RESULTS

4.1 Results of the Model

The following table shows the flow properties in each point in whole cycle calculated by the Excel simulation. From point 1 to 5 is the gas cycle flow and from 6 to 16 represent the steam cycle flow.

Point	P (bar)	T (K)	T (ºc)	h (kJ/kg)	s (kg/sºC)	m (kg/s)
1g	1.00	288.00	15.00	289.01	5.66	603.00
2sg	14.00	612.15	339.15	618.27	5.66	603.00
2g	14.00	688.18	415.18	690.59	6.95	603.00
3g	14.00	1715.62	1442.62	1732.78	7.50	615.07
4sg	1.00	807.15	534.15	815.22	7.50	615.07
4g	1.00	898.00	625.00	906.98	8.02	615.07
5g	1.00	353.00	80.00	356.53	7.03	615.07
6s	0.01	279.97	6.97	29.30	0.11	101.60
7ss	6.00	280.02	7.02	29.90	0.11	15.26
7s	6.00	280.02	7.02	30.05	0.11	15.26
8s	6.00	431.83	158.83	670.50	1.93	15.26
9s	6.00	431.83	158.83	2756.14	6.76	15.26
10s	6.00	588.31	315.31	3093.90	7.43	15.26
11ss	106.00	280.66	7.66	42.55	0.12	86.34
11s	106.00	280.02	7.02	39.90	0.11	86.34
12s	106.00	431.83	158.83	676.45	1.92	86.34
13s	106.00	588.31	315.31	1433.50	3.40	86.34
14s	106.00	588.31	315.31	2714.23	5.58	86.34
15s	106.00	794.13	521.13	3422.41	6.63	86.34
16ss	0.01	279.97	6.97	2092.84	6.63	101.60
16s	0.01	279.97	6.97	1858.21	8.97	101.60

Figure 20. *Results of the combined cycle.*

**Note that points with s are isentropic.*

The following T – s cycle picture shows the different thermal points of each stage in the system. Red line represents gas cycle points; it has higher work temperatures than steam cycle which is represented by blue line for low pressure stages, black line for high pressure stages and red line when the flows get back together.

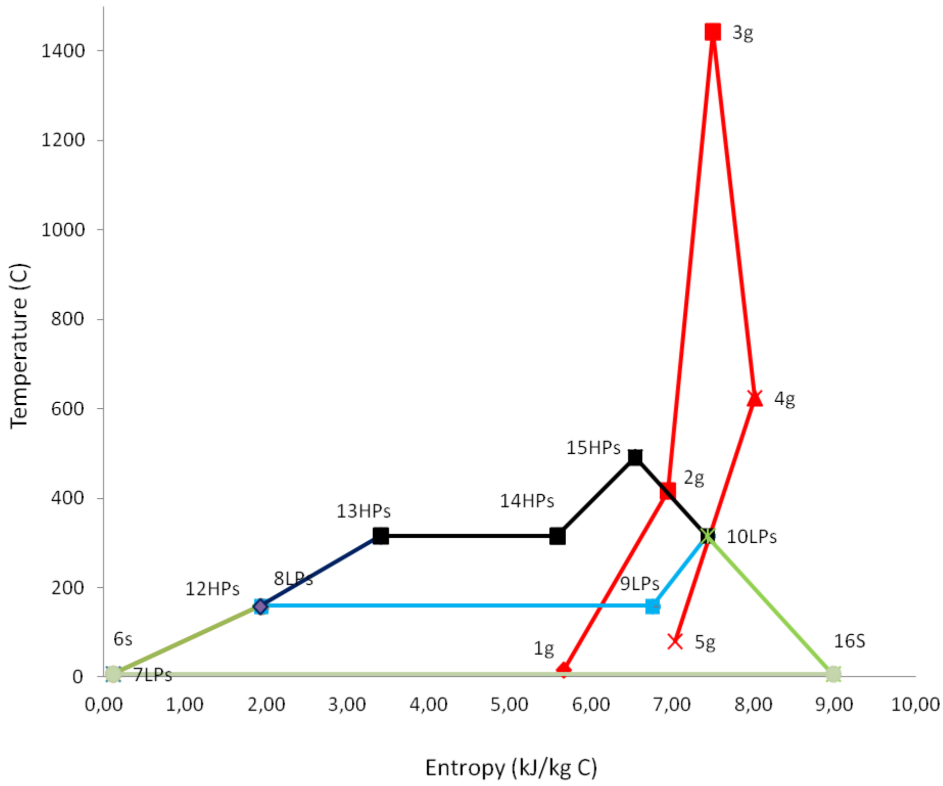

Figure 21. *Diagram T-s cycle.*

The [Figure 22] shows the heat transfer stages inside HSRG. It is possible appreciate the approximation between the flows and the heat consummated by each stage.

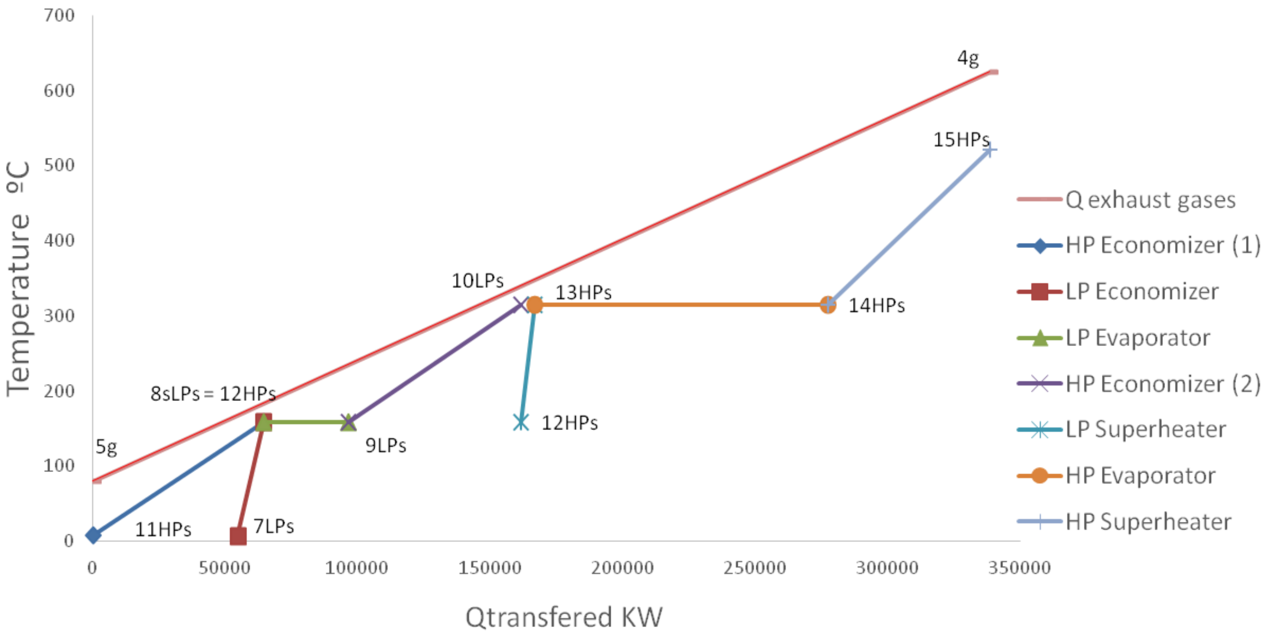

Figure 22. *Heat transfer of each element in the HRSG*

The heat transferred in the HSRG in each stage:

Graphic Points	Element	Heat transferred	Units
Q_{11-12}	HP First Economizer	54.731	Mw
Q_{7-8}	LP Economizer	9.770	Mw
Q_{9-10}	LP Evaporator	31.816	Mw
Q_{12-13}	HP Second Economizer	65.365	Mw
Q_{10-11}	LP Superheater	5.152	Mw
Q_{13-14}	HP Evaporator	110.579	Mw
Q_{14-15}	HP Superheater	61.145	Mw
Q_{TOTAL}		338.562	Mw

Figure 23. *Heat Transferred HRSG.*

As it is possible to see in [Figure 22] and [Figure 23], the evaporator takes the biggest heat transfer stage (110 Mw), it is because the HP pressure flow is significantly higher than LP pressure flow. HP Economizer also takes over 100 Mw but for this system this amount of heat is divided in two stages in the way to increase the performance. It considered that mechanical efficiency of combustion chamber is 100%.

The final values of the overall efficiency and power generated by the cycle are:

Name	Symbol	Value	Units
Net Power Gas Turbine	NeGT	264.0909	MW
Net Power Steam Turbine	NeST	128.7549	MW
Net Power by Combustion Chamber	Ncc	664.5656	MW
Specific Efficiency of the cycle	η_{cycle}	59.1132	%

Figure 24. *Net power consumed and generated and overall efficiency*

4.2 Efficiency Performance

In this chapter, overall efficiency of the system will be discussed and how it changes in function of the two pressures. It is viewed through a simulation represented by a 3-D function. The following parameters are modified and those not mentioned remain constant. The results are presented grouped below:

η(LP, HP)		LP					
		5	7	9	11	13	15
HP	60	58.2938	58.1783	58.0902	58.022	57.9676	57.9232
	75	58.6131	58.558	58.4531	58.3909	58.3392	58.2959
	90	58.8801	58.8198	58.7537	58.6949	58.6442	58.6006
	105	59.1124	59.0717	59.0115	58.9539	58.9025	58.8576
	120	59.3209	59.2953	59.2379	59.1795	59.1261	59.0788
	135	59.5128	59.4976	59.4402	59.379	59.3222	59.2715
	150	59.6937	59.6838	59.6232	59.5571	59.4954	59.4401
	165	59.8686	59.8578	59.7902	59.7167	59.6483	59.5871
	180	60.0433	60.0231	59.9434	59.8592	59.7818	59.713

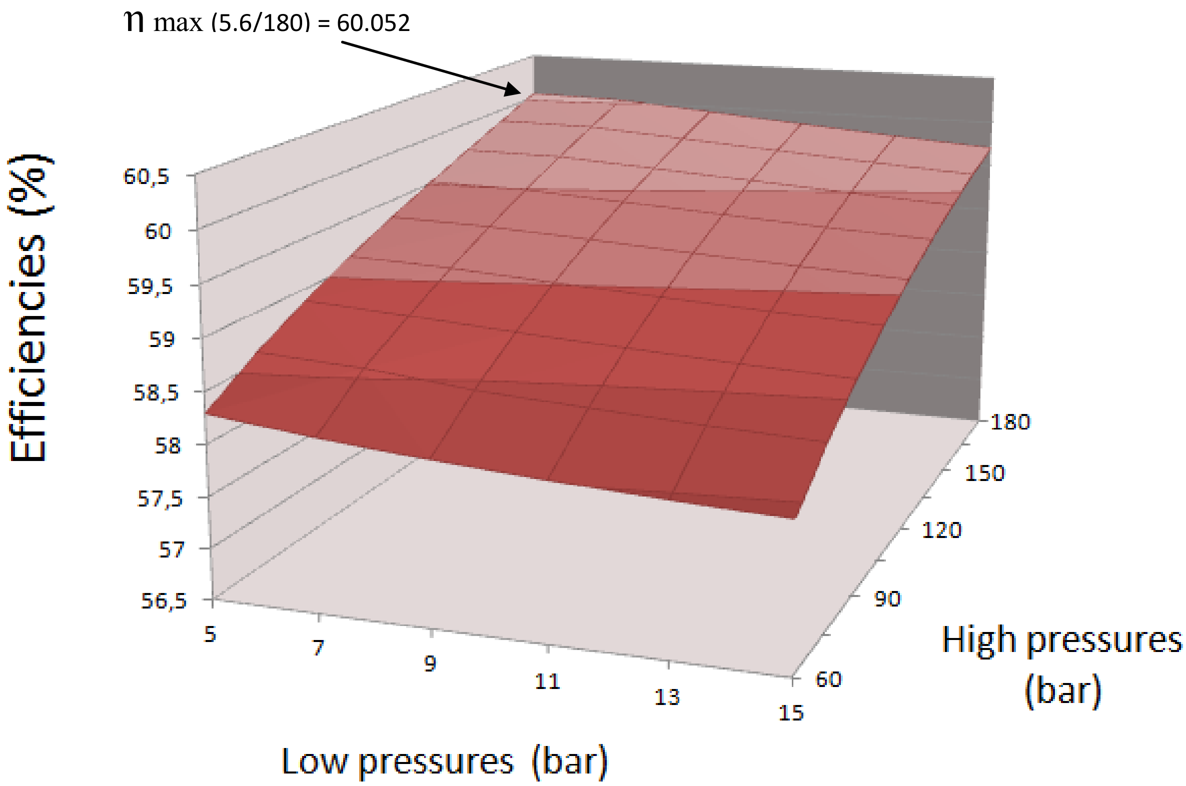

Figure 25. *Data and 3-D chart of General Efficiency.*

4.3 Discussion

This work is focused on the efficiency of the HSRG and by studying of the pressure levels through analysis of the different pressure graphs. It is possible to determine which pressures are the most optimal for this particular cycle as well as it is possible to see the operational limits of the model.

The following are where pressure ranges have been studied:

The maximum for high pressure is 180 bar. Although there are power plants with as high as 300 bar which would imply a higher efficiency, for economic reasons, the limit for this project is established at this value. The economic reasons taken into account are the component design, the thickness of the pipes (thicker pipes for support greater pressures are more expensive) and the higher size of the pump. Also, if the pressure is highly increased, the humidity at the turbine outlet also raises. Consequently, it creates abrasion and ruptures in the pipes and parts of the. To measure this limit, there is a diagram called **Wilson Zone**. It graphs the appropriate work-area composed by 85% of steam which is not harmful for the turbine performance (LM Antón, p. 46). Another important consideration is the fact that this kind of system is use to burn coal as a fuel. Mainly bituminous and lignite which cost of ton is more expensive in order to achieve a better combustion (CIBEN, 2011 Subcritical and Supercritical thermal analysis, p. 1054).

The next situation is more than 15 bar for low pressure measurements have not been considered, because over that value the LP flow tends to be 0, in order for the pinch points to remain constant. Consequently, the results are the same as they would for one pressure level as shown in the [Figure 26]. The final power achieved is **396.3 Mw** which is great power being basically the one pressure case. However, this power is still lower than the best efficiency example.

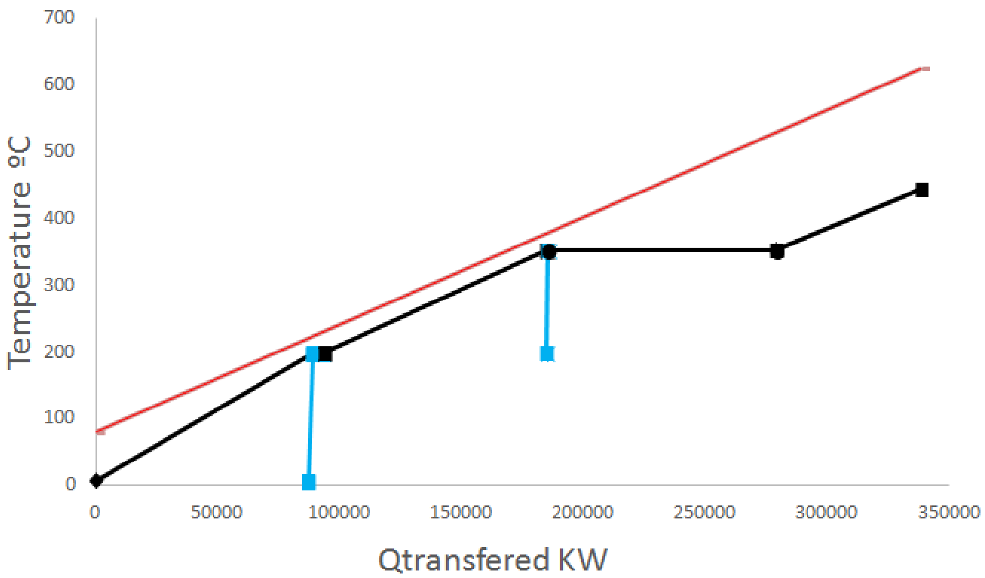

Figure 26. *Example I LP 15 and HP 170 bar*

Then, at less than 5.6 bar for LP circuit, the efficiency starts to decrease again because LP evaporation line increases separation with EG line. As a result, LP flow increases and it consumes a bigger part of the transfer amount than the last case according to the fixed pinch points. Therefore, the power generated is basically the same as before, **396.7 Mw**, and it will continue to decline at a time LP decreases. This case can be compared to the analysis mentioned above (CIBEN, 2011 Subcritical thermal analysis, p. 1054. Figure 1055), but now talking about the subcritical case.

The HP flow works at different pressures, but, nevertheless, it generates the same power because of the variation of the flow amount in each circuit.

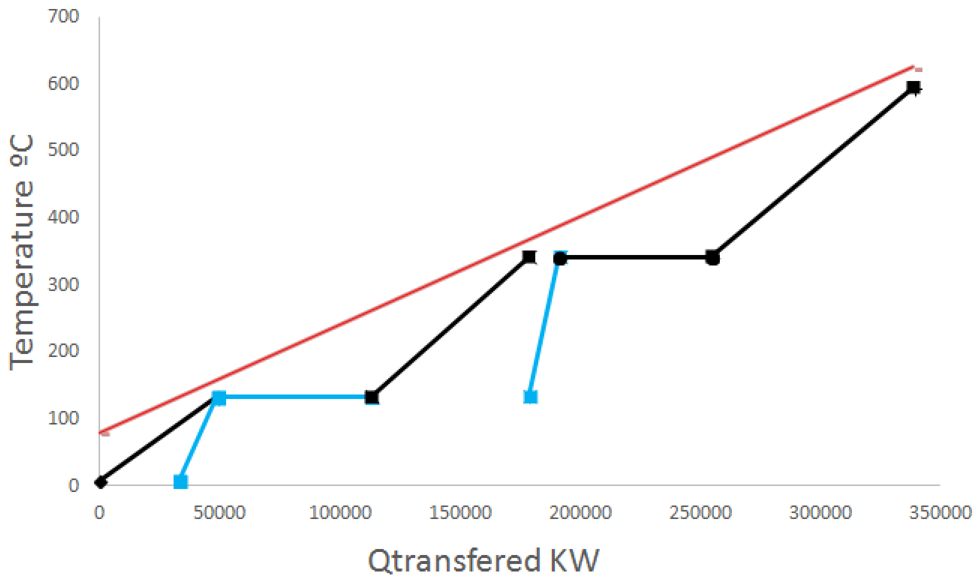

Figure 27. *Example II LP 5 and HP 150 bar.*

The opposite case is showed in [Figure 28]; it does not make sense to continue to reduce the HP because as the HP circuit requires too much heat for the evaporation stage at the time the HP saturation temperature is going down, everything results in a large separation of the warming lines. Consequently, the final power also reduces till **382 Mw**. Moreover, involving a growth in LP and remaining constant HP, it is possible to verify how power continue to decrease and stood at **380.4 Mw**.

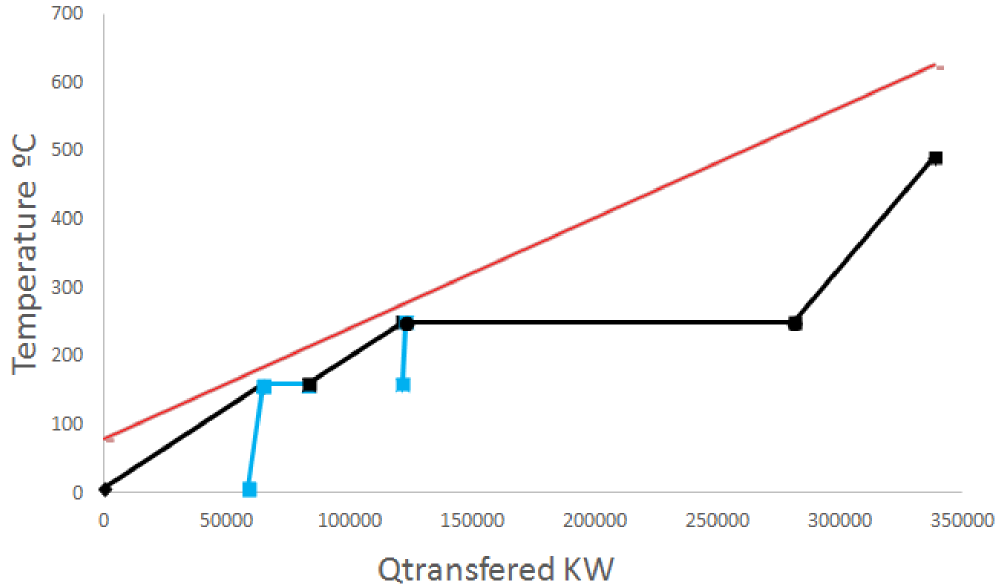

Figure 28. *Example III LP 6 and HP 40 bar.*

The final Example shows the best possible performance of the case. The efficiency can be improved from 59.113% of the study case, to 60.052% of the case graphed in [Figure 29]. The pressure values are LP in 5.6 bar and HP in 180 for this specific model. The power produced is **399Mw**.

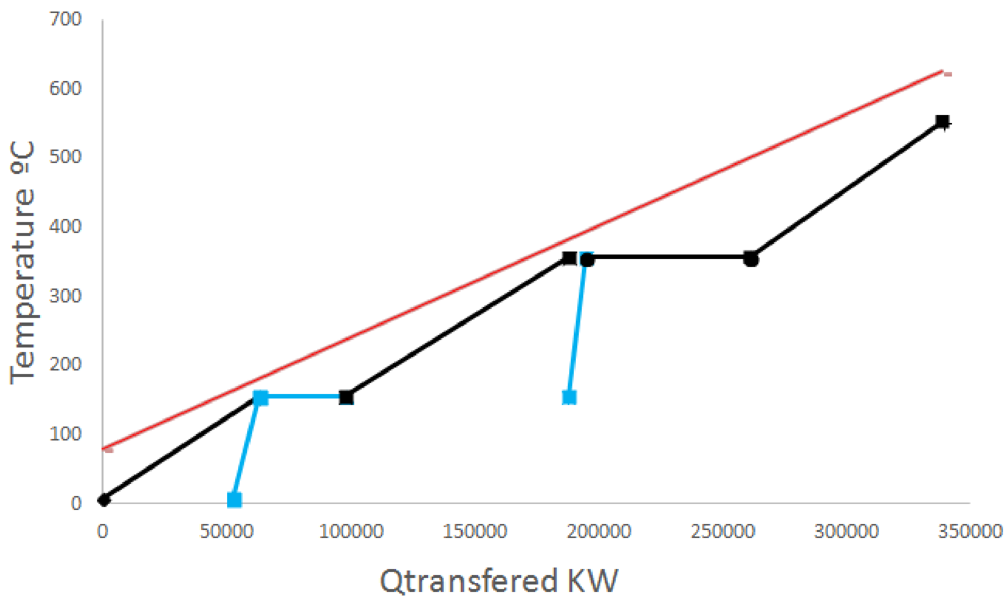

Figure 29. *Example IV LP 5.6 and HP 180 bar.*

Generally speaking, because some approximations were not considered as a real case, it is possible to check that the final performance achieved in this project is quite high. The current record of efficiency in power plant was achieved by Lasward power plant established in Dusseldorf, Germany (EN, Siemens record Jan. 2016). It achieved 61.5% of general performance. The penultimate record was established by Irsching power plant, Germany in 2011 which had 60.75% of efficiency. Both were done by Siemens Company which together with General Electric currently, lead turbines sector (EN, Four biggest turbine manufactures, Ap. 2017). In terms of useful, these great efficiencies are environment friendly and allow an 85% of fuel use as the calculated case.

5. CONCLUSION

Combined cycle technology combines a gas cycle with a steam cycle together with a heat recovery steam generator. This generator uses the energy from exhaust gases from gas turbine to feed the water of the steam cycle. This supports the optimum utilization of the residual energy of the gas cycle. Due to this technology, it is possible to increase efficiency up to 50 or 60 percentages of efficiency which are higher performances than the traditional Brayton cycle performance (25%) and the Rankine cycle performance (40%). (LM Antón p. 102)

The main objective of this work is to create software to represent the operation of a combined cycle system of two pressure levels in order to achieve the maximum efficiency in function and of how different parameters are affected. General efficiency is studied because the steam and gas single efficiencies are not maximum in the case of general efficiency. In fact, the gas performance is lower than expected with regards to the compression ratio. This reduces the power generated by the gas turbine but increases the useful energy in the HRSG.

As it is possible to appreciate, pressure values change a maximum of 1% of the general efficiency for this system. During the case development of the model and considering range of optimal pressures, the general efficiency variation is irrelevant compared to the next parameters:

- Inlet temperature in the gas turbine.
- Environmental temperature suctioned by compressor.
- Flow at different pressures in the HSRG

In relation to the inlet temperature in the gas turbine, it is possible to confirm that a high inlet temperature increases power generated by the gas turbine and it also increases the inlet temperature in the HRSG. This means that reaching high temperatures in the steam turbine result in an increase in the whole plant. For this reason, it is interesting to study materials which are able to support high temperatures or refrigeration systems in gas turbines.

As previously mentioned, power plants function during the whole year so as to meet the energy demand and this is why it is also interesting to study the influence of environment parameters. It is proven that low inlet temperatures in the compressor decrease overall efficiency. One way to increase the general efficiency is by reducing the condensation temperature in the steam cycle, which produces greater expansion in the steam turbine, thereby increasing the amount of power generated. This is why low environmental temperatures in winter create a higher general efficiency. Condensation temperature is also low in relation to exterior temperature.

Finally, steam drums are elements which provide flexibility to the plant. The capacity to utilize the steam which comes from economizer is also a key factor which determines efficiency.

For future work, it is of interest to study three pressure levels of the cycle which reach a higher general efficiency. For this project a cycle of two pressure levels was chosen because it is a useful tool to more extensively analyse a conventional combined cycle rather than a cycle with three pressure levels. Further work on this project could include the design of code which allows the main efficiency parameters to be modified. Some specific alternatives could be:

- Analyses of inlet pressure in steam drums.
- Studying the model by changing the efficiency of the elements.
- Modifying restrictions as pinch points or minimal temperatures by which exhaust gases are vented to the environment.

Figure 30. *Combined Cycle of 800Mw in As Pontes, Spain (Intec Energies).*

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