POLITECNICO DI TORINO

Master's Degree in Mechanical Engineering

Master Thesis

Regeneration of a turbogas plant
Compact heat exchanger's design

Academic supervisor: Prof. Daniela Anna Misul

Co-supervisor: Prof. Mirko Baratta

Company supervisor: Ing. Luca Forno

Candidate: Annapaola Zingarelli

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Abstract

The aim of this thesis is to evaluate the turbogas cycle of the TG16 machine (an Ethos' machine) and to design a compact heat exchanger to turn the cycle into a regenerative one.

Up till now, another student of Politecnico di Torino has already studied the TG16 cycle choosing the shell and tube type of exchanger, which doesn't fit for the size. His analysis is reported at the end of the document (Appendix).

Indice

Chapter 1

Introduction

The social and political contest sensitize the entire community about the environmental pollution: the actual society lives the fight between advantages of progress and drawbacks due to exploitation of non-reusable resources.
In general population growth and modern lifestyle involve an increase in energy demand all around the world; even if the use of fossil fuels tends to be reduced through years.

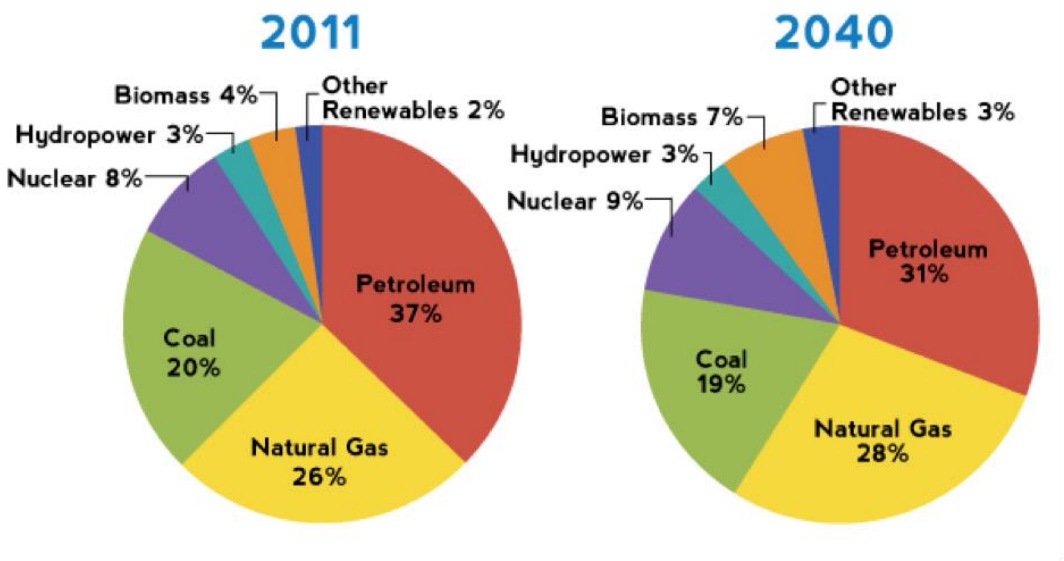

Figure 1. Projection in global energy resources used

During last years lots of industries decide to develop and use different types of 'emission free' energy sources, to reduce as much as possible gas emission, especially CO_2 ones. However, for now, reusable resources cannot be used for everything because of their intermittent power, that makes it really hard to quantify the amount of energy which can be produced, and because of the impossibility to guarantee major powers in a stable way.

It's important to specify that over 65% of the world's electrical energy is produced by fossil fuels, as gas turbines TG do. All these installations have the same aim: to produce electric power; and are commonly called "heavy-duty". Nowadays they are characterized by machines' power which are above 240 MW, and by overall efficiencies even higher than 40% for simple cycles or than 60% for double shafts (both gas turbines and steam turbines are present).
Moreover, gas turbines are advantageous because of their high speed at which are able to work at a full capacity, of their abundant availability, of their low level of emissions and especially because of their flexibility.

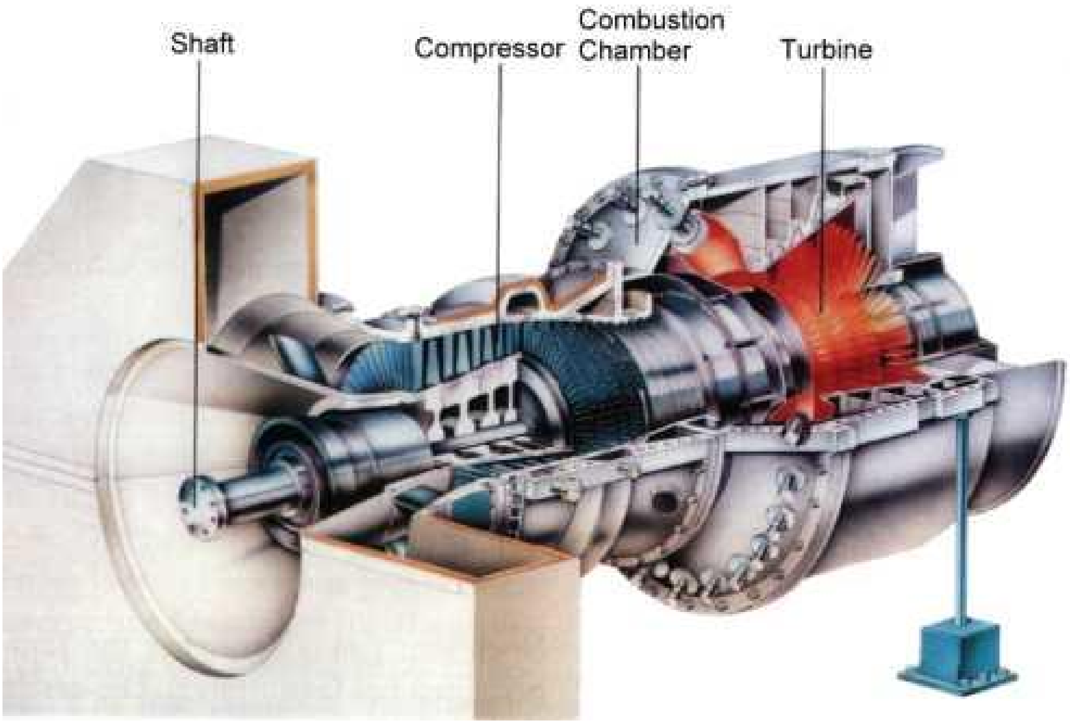

Figure 2. Scheme of a gas turbine

Chapter 2

Heavy-duty turbogas

A gas turbine is an internal combustion engine. The basic scheme of a turbogas is made up of three principal components:

- Gas compressor;
- Combustor;
- Turbo expander (turbine).

The so-called Brayton cycle consists of compressing a gas, dispensing thermal energy by combustion or through a heat exchanger and expanding hot and compressed gas through the turbine that converts thermal energy into power.
The compressor and the expander are located on the same shaft. When the turbogas is used to generate electric power, the turbine is linked to a n electric generator.

Generally, the term “heavy-duty” (or industrial) is used to characterize all the plants which are specifically designed as stationary (non-aircraft unit-derived used). Compared to aircraft turbogas, the heavy-duty ones have lower initial costs, higher power (50 – 500 MW) but lower efficiencies, depending on the maximum temperature of the cycle: for the industrial turbogas the T_{max} is up to 1200° C, instead for the aircrafts it’s about 1400° C.
The characteristic angular speed of the shaft of a turbogas varies between 3000 rpm and 20 - 30000 rpm, depending on the machine size. The velocity of 3000 rpm is the parameter useful to establish the maximum dimensions of the gas turbines: mechanical strength limits of the compressor blades impose a maximum peripheral speed of 500 m/s at the apex, then a maximum section area and so a maximum mass flow rate.

2.1 Main components of turbogas

In the following pages are described and analysed the characteristics of the main components of a turbogas: compressor, combustor, expander, electric generator.

2.1.1 Electric generator

Generally, the electric generator isn't an integral part of a gas turbine; only in the case of turbine shaft is spinning at a speed compatible with the network frequency then the turbine and the generator are mechanically coupled directly, through a joint.
When turbogas require higher speeds two different solutions could be adopted: it's possible to put in a gear mechanical turn reducer, or to insert an inverter that transforms the high frequency of electricity from the rotation of the shaft into a frequency similar to that of the network.
This component causes different types of leaks:

- Mechanical losses, due to the rolling motion of the bearings, the lubrification, etc.;
- Ventilation losses, due to the friction between the rotor and the fluid;
- Electrical losses, due to the resistance of the windings of the circuit when the current flows.

The first two types of losses mainly depend on the angular speed of the shaft, while the last type is affected by electrical power. If there is a reducer it's necessary to consider also losses related to it.
Efficiencies of electric generators are usually pretty high.

2.1.2 Turbo expander

The turbine is a motor machine made up of fixed and rotating blade rows. Usually the number of stages is about 5. The working fluid (gas) expands across the blade passages and transfer work to the walls of the rotating blade vanes. Each stage consists of a set of distributors (fixed), through which the fluid expands, and a set of impellers (rotating), through which the flow produces work. Each blade has a concave side (the pressure surface) and a convex side (the suction surface); the gas pressure is higher on the pressure than the suction surface.
Velocity of the gas reduces ($c_1 > c_2$) as seen in *Figure 3* below, consequently the kinetic energy is converted in mechanical energy, which lets the shaft moving.

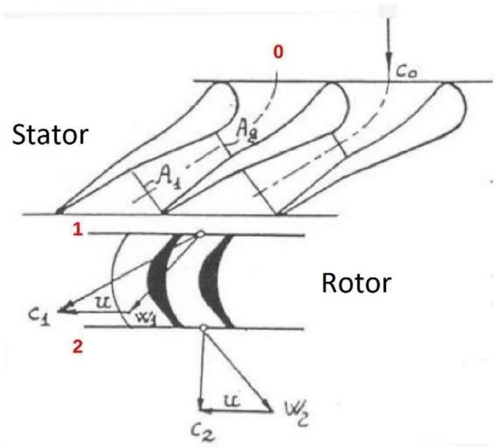

Figure 3. Velocity triangles of a turbine

The turbine is the distinctive element of the turbogas and the most critical one, because of the high temperature to which it's subjected. Turbogas that already exist are able to support about 1800° C, which are temperatures higher than the ones supported by the materials of the expander. This is possible thanks to the powerful cooling systems developed.
However, the goal is to increase more this temperature to raise up also the efficiency.
About the leaks there are both internal and external ones.
The internal could be mainly classified in:

- Losses of profile and incidence, caused by the passage of the fluid on the blades;
- Losses related to secondary flows: vortex in radial direction due to the edges;
- Losses for the development of the boundary layer;
- Losses between the rotating and the fixed blades.

The external could be caused by:

- Organic losses;

- Losses of mass flow rate;
- Heat that flow outward.

There are different types of efficiencies that takes into account all the types of losses in the expander:

- Isentropic efficiency, equal to the ratio between real work and ideal work for an adiabatic reversible expansion

$$\eta_{ad} = \frac{L_i}{(L_i)_{ad}}$$

- Hydraulic efficiency

$$\eta_y = \frac{L_i}{L_i + L_w} = \frac{k}{k-1} \cdot \frac{m-1}{m}$$

- Volumetric efficiency, which compared the different mass flows rate at the suction and at the delivery

$$\eta_v = \frac{\dot{m}}{\dot{m} + \Delta\dot{m}}$$

- Mechanical efficiency

$$\eta_m = \frac{P_u}{P_i}$$

2.1.3 Turbo compressor

The compressor is an operating machine that transfer energy to the fluid (gas), thanks to aerodynamic actions. This energy gets converted into pressure and speed.
Usual, the number of stages in turbogas are between 10 and 20. Each stage is made up of:

- Impeller;
- Diffuser, the fluid is diffused to recover its kinetic energy;
- Distributor (or deflector), an element needed to properly direct the fluid at the impeller inlet.

In turbogas, the compressor is always moved by the expander, thanks to mechanical connection. It's usually used an axial compressor, because it guarantees a higher efficiency and higher mass flow rate compared to a centrifugal compressor.

Figure 4. Gas compressor

Referring to the characteristic curve of gas turbocompressor it' possible to notice that the curve is almost vertical, that means approximately at a constant mass flow rate.

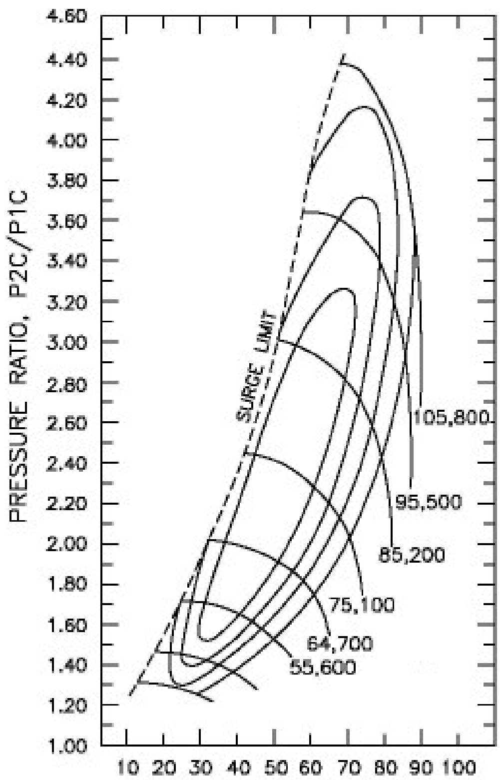

Figure 5. Characteristic curve of an axial turbocompressor

It's common to use statoric blades with variable angle, to regulate the power through the adjustment of the mass flow rate at a constant speed.

The design of the turbocompressor is so much complex, maybe it's considered the most difficult in the design process in a turbogas plant, because of its rigidity: it's not very suitable in working with variable mass flow rate and variable compression ratio.

For what concern the fluid dynamics losses, the study is equivalent to the previous one made for the turbines. Moreover, an important leak linked to the compressor is the one located in the suction air tubes. It is usually about 1 kPa referred to the environmental pressure. It's very important to filter the air, both for decrease the dirtying of the air and for avoid eroding effects of the blades. However, it's necessary to clean frequently the compressor, also during the working.

The work of the compressor on the fluid is evaluated with the 1st Principle of Thermodynamics, with the assumption that the compression is a polytropic transformation with 'm' as exponent:

$$L_c = \frac{k}{k-1} \cdot R \cdot T_1 \cdot \left(\beta^{\frac{m-1}{m}} - 1\right)$$

Where β is the compression ratio of the compressor and k is a constant that depends on the gas (k = 1,4 for air, if it's considered a perfect gas).

Heat exchange could be neglected because of the high speed of the fluid ($Q_e \approx 0$), while the work of the passive resistance (L_w) must be considered:

$$L_c - L_w = \frac{m}{m-1} \cdot R \cdot T_1 \cdot \left(\beta^{\frac{m-1}{m}} - 1\right)$$

To take into account all the losses, there are different efficiencies:

- Isentropic efficiency, equal to the ratio between the minimum work for an adiabatic reversible compression and the real work

$$\eta_{ad} = \frac{(L_i)_{ad}}{L_i}$$

- Polytropic efficiency, which depends on the gas nature and on the exponent 'm' of the polytropic curve

$$\eta_y = \frac{L_i - L_w}{L_i} = \frac{m-1}{m} \cdot \frac{k}{k-1}$$

- Volumetric efficiency, that is the ratio of the amount of gas entering the compressor (suction) versus the amount of gas leaving the compressor (discharge). It considers reflow air in suction environment due to the gap between fixed and moving parts.

$$\eta_v = \frac{\dot{m}}{\dot{m} + \Delta\dot{m}}$$

  Where $\dot{m}$ is sucked mass flow rate and $\Delta\dot{m}$ is re-flowing mass flow rate;

- Mechanical efficiency, equal to the ratio between power given to the fluid and the one available on the shaft

$$\eta_m = \frac{P_i}{P_a}$$

2.1.4 Combustor

The combustor's aim is to raise up the temperature of the fluid, before that the fluid goes in the turbine. To do so it uses the heat released from the fuel oxidation reactions.

It exists a maximum value of the cycle temperature T_3 (in stechiometric combustions T_3 could be also 300 K), so the amount of fuel needed is lower than the one used for a stechiometric combustion of the outgoing air of the compressor; otherwise, the temperature would be too much high for the plant materials.

Turbogas combustors usually work in hard conditions:

- High ingoing temperatures;
- High pression and speed of the ingoing fluid.

Then, the combustor for a gas turbine plant needs:

- Elevated combustion efficiency;
- Reliable ignition system;
- Wide stability range;
- Low load losses;
- Low emissions.

Turbogas are internal combustion engine, this means that combusted gases take part of thermodynamic cycle and that it's important to choose the right fuel to avoid corrosion, erosion and filth.

A stable combustion needs the right amount both of fuel and of oxygen: it's difficult to combust with too much air, so it exists a minimum value below which the flame turns off.

The combustor is divided in three zones, as shown in the figure below:

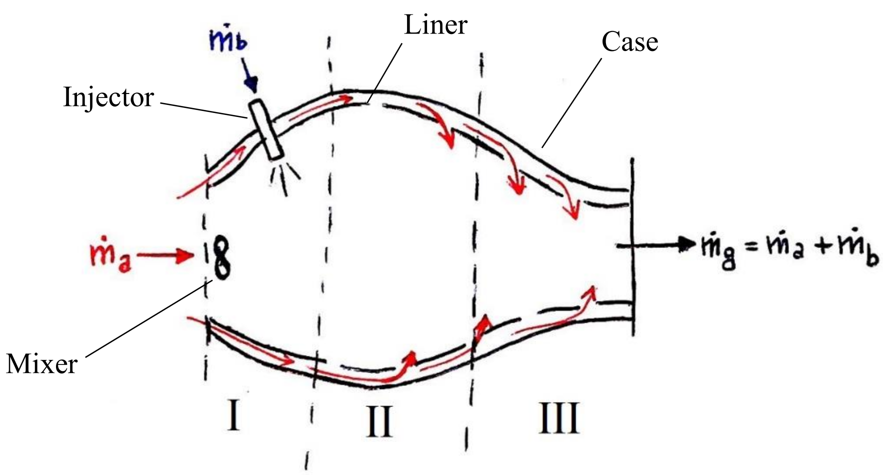

Figure 6. Scheme of a combustor

The gas flows from the left to the right.

The air from the compressor is partially directed in the first stage and partially in the zone between the outer shell of the combustor (case) and the liner. In this way it’s possible the liner introduces the various airflows in the combustion zone.

At the beginning of the I stage there is a mixer (also called “swirler”) that generates turbulence in the flow to rapidly mix the air and the fuel.

The air that is introduced in the I stage is dosed with a stechiometric air-to-fuel ratio (α) to burn almost all the fuel, and reduce pollution, even if the efficiency decrease. While the fuel is introduced in the I stage by the injector.

From the last parts of the primary zone and the beginning of the secondary one the air between the case and the liner starts to flow in the combustion area, thanks to the holes in the liner. This air is added to complete the oxidation: the temperature is still enough high to keep active the chemical reactions. The holes in the liner are throttled ($\eta_{\pi\ b}$) to slow down the air and avoid the flame propagation.

Finally, in the III stage the air is added to cool the mixture and is called “dilution area”.

There are 3 different types of combustors, classifying in base of liner’s form:

- Can, are self-contained cylindrical combustion chambers. Each “can” has its own fuel injector, igniter, liner and casing. The primary air from the compressor is guided into each individual can, where it’s decelerated, mixed with fuel and then ignited. The secondary air also comes from the compressor, where it is fed outside of the liner.
  Referring to *Figure 7* the blue indicates cooling flow path, while the orange indicates the combustion product flow path;

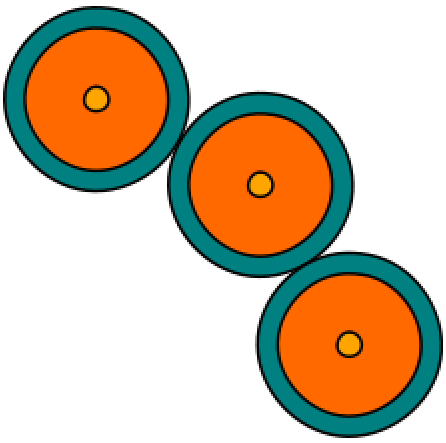

Figure 7. Can combustor, looking axis on, through the exhaust

- Cannular, like the previous type, this combustor has discrete combustion zones contained in separate liners with their own fuel injectors. All the combustion zones share a common ring casing. The combustion zone can also communicate with each other thanks to liner holes. The outgoing flow from the cannular combustor generally has a more uniform temperature profile and eliminates the need for each chamber to have its own igniter;

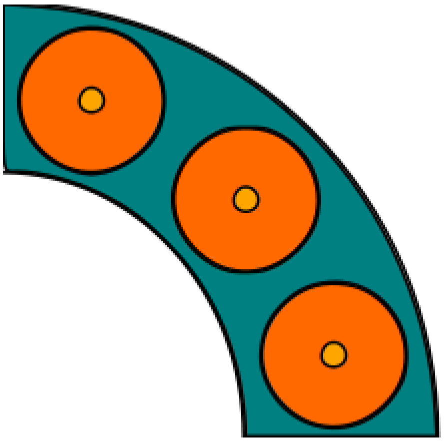

Figure 8. Cannular combustor, viewing axis on, through the exhaust

- Annular, have a continuous liner and casing in a ring. It is the most commonly used type of combustor, because of its numerous advantages, e.g. more uniform combustion, shorter size and less surface area. But the most important advantage is the very uniform exit temperatures.

  Referring to *Figure 9* the orange circles indicates the fuel injection nozzles;

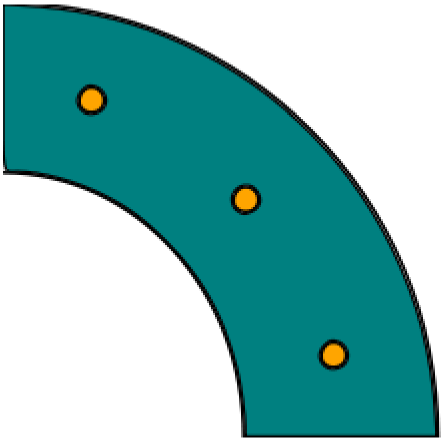

Figure 9. Annular combustor viewing axis on, through the exhaust

2.2 Turbogas cycle

Comparing the different types of cycles from the simplest to the hardest:

- Ideal cycle;
- Real cycle;
- Regenerated cycle.

2.2.1 Ideal cycle

The simplest type of cycle is the ideal one. The Joule-Brayton cycle is the one that fits most for a turbogas with constant pressure in combustion. It consists of four phases: an isentropic compression (1-2), an isobaric adduction of heat (2-3), an isentropic expansion (3-4) and an isobaric heat dissipation (4-1) considering a closed cycle.

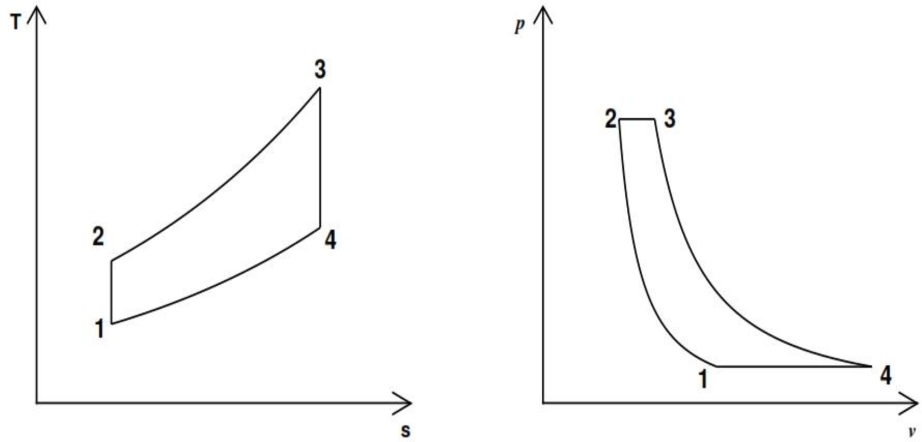

Figure 10. Ideal cycle of a turbogas

1 – 2 The gas is compressed in the first phase. Ideally, this transformation takes place without heat exchange and in a reversible manner (isentropic: adiabatic + reversible). The compression causes an increase in the temperature and a reduction of specific volume;
2 – 3 The gas receives heat from outside at a constant pressure. The supply of heats leads to an increase of the temperature and even an increase of the specific volume.
There is no work exchanged with the outside;
3 – 4 The gas is expanded in this phase. As in the first phase there is not heat exchange and the phase is reversible, so it's isentropic too. The gas expansion carries out work, so temperature decreases, whilst its specific volume increases;
4 – 1 The gas reaches the initial pressure at the end of the previous transformation but needs to be cooled to return to initial conditions.

Using this type of cycle means to make the following assumptions:

- The gas is considered as ideal, this means that the equation state $p \cdot V = m \cdot R \cdot T$ is valid and that c_v c_p and γ are constant;
- Friction, pressure and heat dissipations are assumed negligible;
- Mass variation is considered negligible, so the mass is constant, without any adding of fuel, neither for dissipations;
- Compression and expansion are isentropic transformations.

Obviously with these hypotheses the performances are better than the reality ones.
The useful work L_u corresponds to the area of the cycle in the *Figure 10* and it's equal to the difference between the work of the turbine and the one of the compressor.

$$L_u = L_t - L_c = Q_1 - Q_2$$

Where Q_1 and Q_2 are, respectively, the heat incoming in the combustor and the outcoming heat due to the cooling phase.

$$Q_1 = c_p \cdot (T_3 - T_2) \qquad\qquad Q_2 = c_p \cdot (T_4 - T_1)$$

The efficiency of the cycle is:

$$\eta_{id} = \frac{L_{id}}{Q_1} = 1 - \frac{Q_2}{Q_1}$$

Knowing that:

$$\frac{T_2}{T_1} = \beta^{\frac{k-1}{k}} \qquad \frac{T_2}{T_1} = \beta^{\frac{k-1}{k}} \qquad \text{with} \;\; k = \frac{c_p}{c_v} \;\; \text{and} \;\; \beta = \frac{p_2}{p_1}$$

The efficiency could be written in function of β and k, as:

$$\eta_{id} = 1 - \frac{1}{\beta^{\frac{k-1}{k}}}$$

2.2.2 Real cycle

The ideal cycle allows for understanding the basics of gas turbine plants, but performances can only be defined referring to the real cycle.

The real cycle is an open cycle in which the fluid is to be considered as a real fluid and it's necessary to consider the viscous losses that takes place throughout the compression and the expansion, thus making these latter real evolutions.

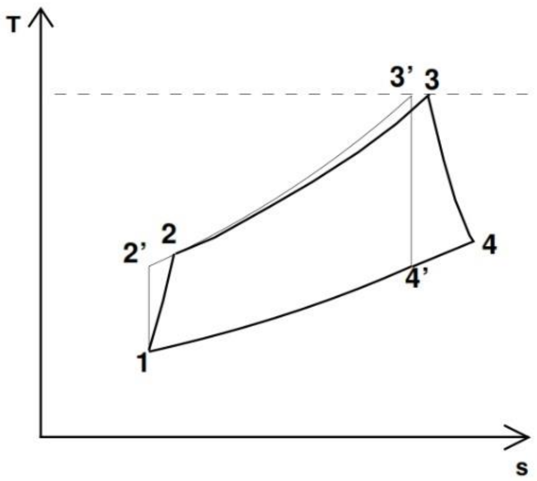

Figure 11. Real cycle of a turbogas

Main differences with the ideal cycle are consequences of:

- The fluid is real, this means that c_p and c_v are not constant when the temperature varies, so the efficiency in this case is lower than the one of the previous case;
- The compression and the expansion are assumed adiabatic, but not isentropic. These losses are considered in the isentropic efficiencies referred to the turbine and the compressor

  $$\eta_c = \frac{L_{c,id}}{L_c} = \frac{h_{2id}-h_1}{h_2-h_1} \qquad \eta_t = \frac{L_{t,id}}{L_t} = \frac{h_{3id}-h_4}{h_3-h_4}$$

  In common gas turbines maximum temperatures out of the combustor are about 1100 – 1500° C, but materials can support around 800 – 900° C. So, it's necessary to cool down first stages of the turbine. To do so it's used the compressed air taken to the compressor, mixed with the fluid that flows in the turbine.;

  Both expansion and compression are modelled on a polytropic curve, then a polytropic coefficient is used.
- The difference between the pressure in and out of the heat exchanger, caused by the load loss and taken into account by the following pneumatic efficiency

  $$\eta_\pi = \frac{p_3}{p_2}$$

The transformation 2-3 is no more isobaric (p_2 = const), but pneumatic efficiency is near to 1 thanks to some technical stuff;

- The imperfect combustion and then energy and power losses. Usually, these losses in turbogas are very low, but it's necessary to considered them by the burner efficiency

$$\eta_b = \frac{\dot{m}_a \cdot Q_1}{\dot{m}_b \cdot H_i}$$

- The mechanical friction in the compressor and in the turbine, respectively considered by mechanical efficiency $\eta_{m,c}$ and $\eta_{m,t}$. Principal causes of these leaks are: friction on the bearings, the lubrification, the ventilation of the rotating parts and auxiliary stuffs.

Finally, the power of the plant must consider all these losses. It is given by the formula:

$$P_u = \eta_m[(\dot{m}_a + \dot{m}_b) \cdot L_t - \dot{m}_a \cdot L_c]$$

When taking into account the specific work, it's necessary to state what mass flow rate it's referring to. In fact, the turbine specific work is referred to the mass of air and fuel ($\dot{m}_a + \dot{m}_b$), whereas the compressor work is referred to the air mass flow rate ($\dot{m}_a$). By convention, the plant specific quantities will be always referred to the air mass flow rate.

The two mass flows rate are connected by the factor α:

$$\alpha = \frac{\dot{m}_a}{\dot{m}_b}$$

It's possible to re-write the power in function of α, as:

$$P_u = \eta_m \cdot \dot{m}_a \left[\left(\frac{1+\alpha}{\alpha}\right) \cdot L_t - L_c\right]$$

So, the overall efficiency formula is:

$$\eta_g = \eta_m \frac{P_u}{\dot{m}_b \cdot H_i} = \eta_m \eta_b \frac{\left(\frac{1+\alpha}{\alpha}\right) L_t - L_c}{Q_1}$$

This formula shows how in the real cycle the efficiency is function of the maximum temperature of the cycle T_3, of the compression ratio and of the gas type.

2.2.3 Regenerative cycle

Regeneration is a typical way to increase efficiency: it's possible to increase the plant efficiency by adopting inter-refrigerated re-compression, by exploiting the re-combustion technique and by taking advantage of regeneration.

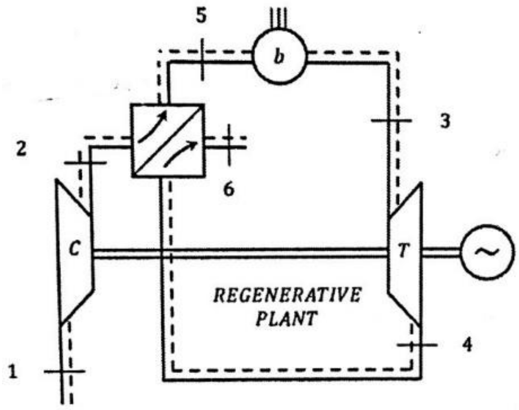

Figure 12. Regenerative plant

In turbogas, the mass flow rate out of the compressor (point 2) is re-heated, before the inlet of the combustor (point 5) using the exhausted gases at the outlet of the turbine (point 4). So, combusted gases pass in the heat exchanger and then dumped into the environment. But regeneration is only possible provided that $T_4 > T_2$ and is meant to pre-heat the air at the burner inlet by exploiting the hot gases exhausted by the turbine.

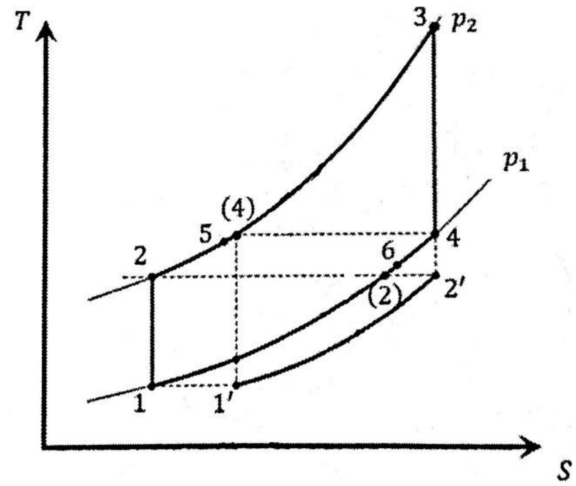

Figure 13. Ideal cycle of a regenerative turbogas

Regeneration takes different advantages:

- Less energy is consumed thanks to the higher efficiency;
- Less emission of pollution in the atmosphere;
- Less risks of interruption of the system.

The *Figure 13* refers to the ideal regenerative cycle, so the combusted gases expand from pressure $p_2 = p_5 = p_3$ to $p_1 = p_4$ in the turbine.
The hypotheses for an ideal regenerative cycle are:

- Consider the gas as perfect;
- Both compression and expansion are isentropic transformations;
- The regeneration is ideal, this means that there are no losses in the heat exchanger.

Characteristic temperatures in the *Figure 13* are:

- T_1 = temperature of compressor suction;
- T_2 = temperature of compressor delivery;
- T_3 = temperature of turbine inlet;
- T_4 = temperature of turbine outlet;
- T_5 = temperature of regenerator outlet air;
- T_6 = temperature of regenerator outlet combusted gases.

If $T_2 = T_6$ and $T_4 = T_5$ compressor's and turbine's work remain unaffected by the base cycle; but the heat leap decreases from $T_2 - T_3$ to $T_5 - T_3$, so the heat entering the cycle decreases:

$$Q_{\text{base cycle}} = c_p \cdot (T_3 - T_2)$$

$$Q_{\text{regenerated cycle}} = c_p \cdot (T_3 - T_5)$$

Recalling that the efficiency formula is:

$$\eta_{id} = \frac{L}{Q}$$

When the work stays the same and the heat decrease, the efficiency increases.

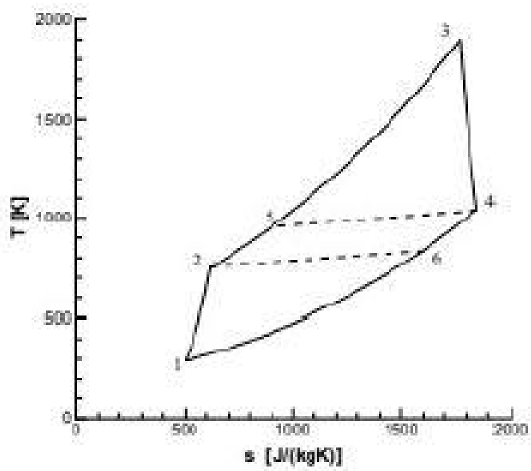

Figure 14. Semi-real cycle of a regenerative turbogas

Unfortunately, is not physically possible to transfer all the heat of combusted gases to the compressed air: $T_2 \neq T_6$ and $T_4 \neq T_5$, in particular $T_2 < T_6$ and $T_4 > T_5$.

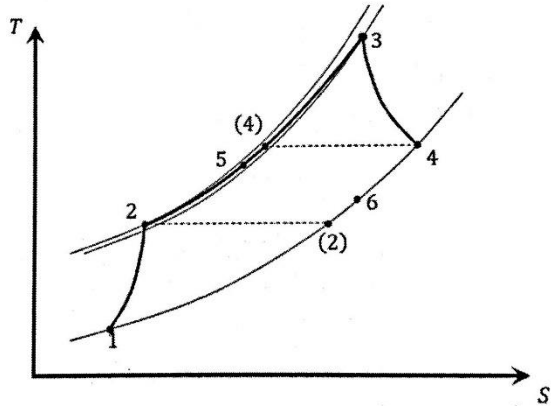

Figure 15. Real cycle of a regenerative turbogas

The ratio between the regenerated heat and the maximum retrievable heat corresponds to the efficiency of the heat exchanger, also called regenerative efficiency R_S:

$$R_S = \frac{T_5 - T_2}{T_4 - T_2} = 0 \div 1$$

It's a value between 0 and 1, where 0 corresponds to a turbogas not regenerated at all, while 1 is the value for a completely regenerated cycle.
The overall efficiency depends on R_S as shown in the figures below:

- For an ideal regenerative cycle

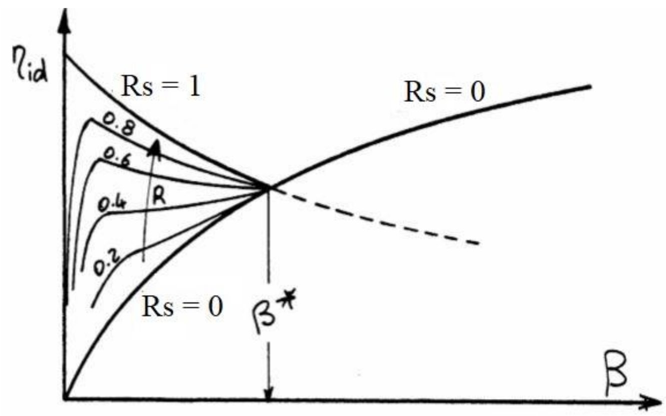

Figure 16. Overall efficiency-pressure ratio plot referring to idea regenerative cycle

- For a real regenerative cycle

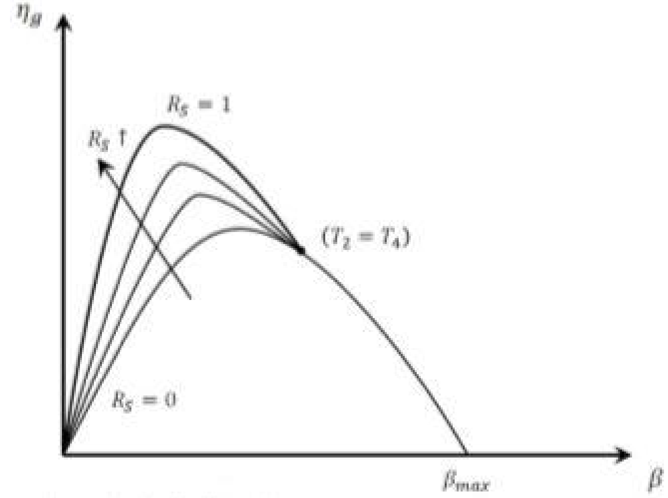

Figure 17. Overall efficiency-pressure ratio plot referring to real regenerative cycle

Chapter 3

Characteristic diagrams

The aim of this thesis is to convert the gas turbine TG16 into a regenerative cycle, using the similar, but smaller, plant TG5BR.
The study is articulated in the following phases:

- Design of the non-regenerative model of the TG16 using CW4145 software;
- Design of the regenerative model TG16R using CW4145 software;
- Choice of the boundary condition of the heat exchanged;
- Design of the compact heat exchanger using the ASPEN EDR software;
- Efficiency analysis of the heat exchanger on regeneration;
- Economic analysis in regenerative e non- case.

First of all, it's important to compare data of TG16 and TG5BR:

- Data of non-regenerative TG5B

[P_n] CV	[P_{max}] CV	[η_g] %	[n] rpm	[T_4] °C	[G_4] kg/s	[n_{max}] rpm	[T_3] °C	[HR] kcal/CVh
8350	9100	18.4	5500	496	49.8	6500	788	3440

- Data of regenerative TG5BR

[P_n] CV	[P_{max}] CV	[η_g] %	[n] rpm	[T_4] °C	[G_4] kg/s	[n_{max}] rpm	[T_3] °C	[HR] kcal/CVh
7250	9100	29.8	5500	-	-	6500	788	2120

Comparing the two data it's possible to evaluate that regeneration increase the efficiency and decrease the power, because of pressure losses in the heat exchanger

$$\Delta P_n\ \% = \frac{P_{n,TG5B}}{P_{n,TG5BR}} = -\ 13.2$$

$$\Delta\eta\ \% = \frac{\eta_{\ TG5B}}{\eta_{\ TG5BR}} = +\ 62$$

- Data of non-regenerative TG16

[P_n] kW	**[P_{max}] kW**	**[η_g] %**	**[n] rpm**	**[T_4] °C**	**[G_4] kg/s**	**[η] % R/A**	**[T_3] °C**	**[HR] kcal/kWh**
18200	19500	25.9	4850	409	122.7	98.5/98	788	3320

Description of the data present in the tables:

- [P_n] CV, power measured at the alternator;
- [P_n] kW, power measured at the turbine;
- [η] %, overall efficiency;
- [n] rpm, rotational speed;
- [T_4] °C, temperature at the turbine delivery;
- [G_4] kg/s, mass flow rate at the turbine delivery;
- [η] % R/A, reducer/alternator efficiency;
- [T_3] °C, temperature at the turbine intake;
- [HR] kcal/CVh, specific consumption;
- [HR] kcal/kWh, specific consumption.

Moreover, it must be considered that there a dimensional constrain: the heat exchanger will be positioned under the turbine, using an elevated base, then the maximum dimensions are

- Length: 7.1 m;
- Wideness: 2.9 m;
- Height: 2.9 m.

The CW4145 software is used to verify the expected performances of an industrial single shaft gas turbine and to obtain the characteristic diagram both of the TG16 and of the TG16R. Then it's necessary to compare the output data with the one previous reported to validate that the model fits with the reality.

The process of the program CW4145 is articulated in different steps:

- Compute the right number of rounds of the compressor, knowing the environmental temperature and the effective speed of the shaft;
- Define the right constant round curve, using interpolation between the right number of rounds of the compressor and the input one on the compressor's map;
- Determine the conditions of the flux at the inlet of the combustor, thanks to the right number of rounds of the compressor, the mass flow rate the compression ratio and the compressor efficiency;
- Calculate the turbine inlet temperature (TIT) for each point of the curve, using a first random Stodola's factor;
- Compute the fuel mass flow rate, so also the inlet turbine mass flow rate;
- Estimate the turbine expansion with an iterative method, which fixes the TIT and changes the inlet turbine mass flow rate to always obtain the correct expansion ratio. These methods continue until the difference between the obtained updated Stodola's factor and the one imposed is under a certain value.

The characteristic curve of the compressor is represented on a diagram with pressure ratio on the horizontal axis and corrected flow on the vertical axis, as shown in the figure below.

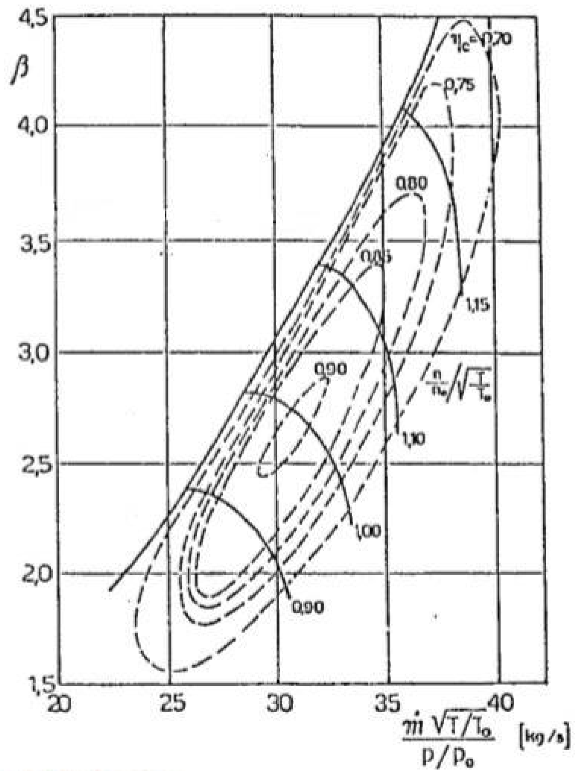

Figure 18. Characteristic diagram of an axial turbocompressor

Figure 18 shows how the compressor diagram depends on two reference conditions: p_0 and T_0, which are usually the standard environmental one (p_0 = 1 atm; T_0 = 297.14 K).
The compressor considered is axial because this type of compressors is able to support high mass flow rate with elevated efficiency and limited dimensions. It is characterized by almost vertical curve for constant velocity of the shaft: very small variation of the mass flow rate. This difference gets smaller and smaller as the speed increases.
There are un upper limit and a lower limit: the upper one is the surge line, this line intersects the characteristic curves before of their maximum point; while the lower one is the choke, which consists of achieving sonic conditions in one or more stages of the compressor and involves shocks and a significant efficiency decrease.
However, it's possible to increase the variation mass flow rate zone using variable geometry compressor (VGC) with inlet guide vanes (IGV).
The characteristic curve of the turbine is represented on a diagram with the pressure ratio of the turbine on the horizontal axis and a mass flow rate parameter on the vertical axis.

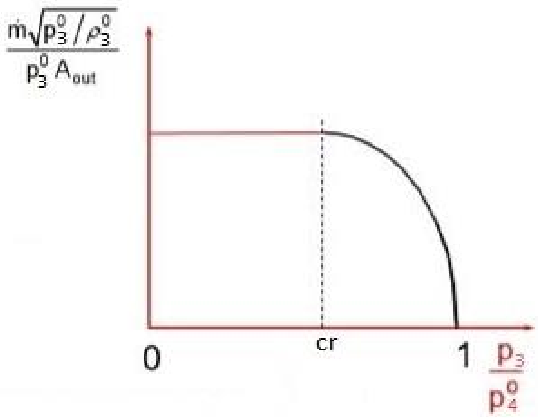

Figure 19.Characteristic diagram of an axial turbine

Where, in the figure before, the point 3 refers to the suction condition of the turbine, while the point 4 refers to the delivery.
The maximum value of mass flow rate is in the left region, which is called "chocked" region and it is reached when the pressure ratio overcome the critical value β_{cr}. The curve is defined with the following equations:

$$\begin{cases} \dfrac{\dot{m}\sqrt{\dfrac{p_3}{\rho_3}}}{p_3} = \text{const} \quad \text{if } \dfrac{p_3}{p_4} < \left(\dfrac{p_3}{p_4}\right)_{cr} \\ \left(\dfrac{\dfrac{\dot{m}\sqrt{\dfrac{p_3}{\rho_3}}}{p_3}}{\Gamma}\right)^2 + \left[\dfrac{\left(\dfrac{p_3}{p^*_4}\right) - \left(\dfrac{p_3}{p_4}\right)_{cr}}{1 - \left(\dfrac{p_3}{p_4}\right)_{cr}}\right]^2 = 1 \quad \text{if } \dfrac{p_3}{p_4} < \left(\dfrac{p_3}{p_4}\right)_{cr} \end{cases}$$

It’s simple to observe that it’s necessary to change the reference axes of one element to adapt it to the other one’s diagram. It’s simpler to change the turbine’s reference axes because it doesn’t depend on the rotational speed.

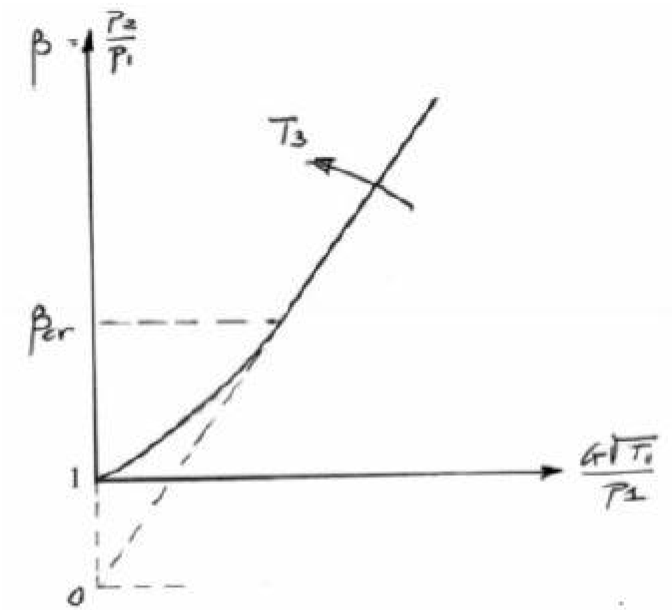

Figure 20. Characteristic diagram of turbine adapted to compressor's axes

Drawing both the curve of the turbine and the one of the compressor in the same diagram with commo reference axes, it’s possible to obtain the working point.

Regenerating the plant, it’s important to consider also the pressure losses in the regenerator using the efficiency $\eta_{\pi R}$: $p_3 = \eta_{\pi R} \cdot \eta_{\pi} \cdot p_2$ for which there is a slip of the working point because:

$$\beta = \frac{k}{\eta_{\pi R}} \cdot \sqrt{T_3} \cdot G_1 \cdot \frac{\sqrt{T_1}}{p_1}$$

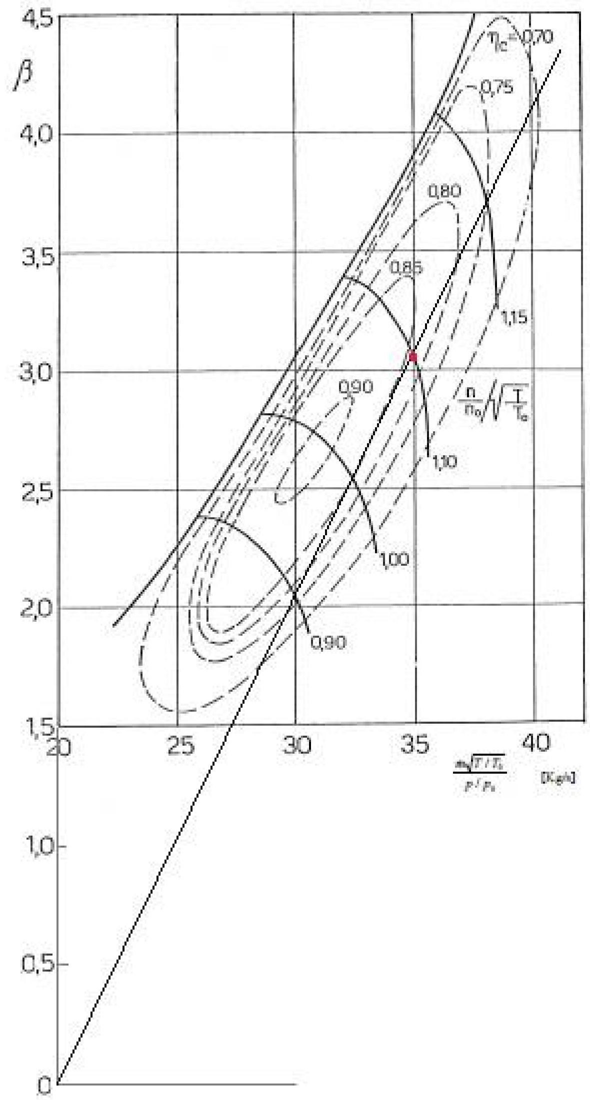

Figure 21. Characteristic diagram of turbine and compressor in the same diagram

3.1 Design of the TG16 and TG16R models

Using the software CW4145, it's necessary to analyse the performances of both the models (the TG16 with base cycle and the TG16R with the regenerative cycle). Then, comparing the computerized model and the experimental data of the two plants, is possible to know if they fit.

3.1.1 TG16 model

To use the CW4145 some assumptions are needed:

- Air composition is divided as follow.
  Molar percentage of the elements in dry air;
  - N_2 = 78.09 %;
  - O_2 = 20.95 %;
  - Ar = 0.93 %;
  - CO_2 = 0.03 %;
- Combustion reaction is modelled as:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

  So, the stechiometric ratio is 0.5. While in this case the quantities are:
  - Air = 123 kg/s;
  - CH_4 = 1.6 kg/s.

  Transformed with the molar masses they become:
  - Air = $4.25 \cdot 10^3$ mol (M = 28.96 g/mol);
  - CH_4 = $0.1 \cdot 10^3$ mol (M = 16 g/mol).

  Excess air is $= \frac{42.5}{9.52} \cdot 100 \cong 400\%$.
- Both losses on compressor and on turbine use the isentropic efficiencies ($\eta_{is,c}$ = 0.84 and $\eta is,t$ = 0.91);
- Pneumatic losses in the combustor are about 3.2% ($\eta_{\pi,b}$ = 0.968);
- Mechanical losses are $\eta_{m,t} = \eta_{m,c} = 0.97$;
- The combustion is not perfect: it has 2.85% of losses (η_b = 0.972).

Using all the previous assumptions in the CW4145 program, the analytical model of the turbogas TG16 is obtained. Now, it's possible to compare data of the physical and the analytical models.

	[P_n] kW	**[T_4] °C**	**[G_4] kg/s**	**[T_3] °C**	**[HR] kcal/kWh**
Physical model	18200	409	123	788	3320
Analytical model	18176	409	122	788	3316
Variation %	-0.13 %	-0.12 %	-0.98 %	0.00 %	-0.12 %

It’s easy to see that the analytical model perfectly fits with the physical one, so it’s considered reliable.

3.1.2 TG16R model

For the regenerated model is not possible to use the same method of the TG16 because the physical model doesn't exist yet. It is used the Westinghouse Canada's study of the 1976 as reference.
Reference data are:

- Fuel: natural gas;
- Power = 17300 kW;
- Heat Rate = 10100 BTU/kWh;
- Speed = 4650 rpm;
- T_{amb} = 60 °F;
- T_3 = 1450 °F.

Two assumptions are needed, to evaluate the model:

- Regenerated efficiency is: $\varepsilon = \frac{T_5 - T_2}{T_4 - T_2} \cdot 100 = 87\ \%$;
- Pressure losses in the regenerative heat exchanger are:
  - 2% of the air, $\eta_{\pi A} = \frac{p_5}{p_4} = 0.98$;
  - 2% of the combusted gases, $\eta_{exc} = \frac{p_6}{p_4} = 0.98$;

Both hypotheses are made from the turbogas TG5 BR data.
Using all the previous assumptions in the CW4145 program, the analytical model of the turbogas TG16R is obtained. Now, it's possible to compare data of the model obtained by the Westinghouse's study and the analytical model.

	[P_n] kW	**[HR] kcal/kWh**
Westinghouse model	17300	2545
Analytical model	17296	2547
Variation %	-0.02 %	+0.08 %

It's easy to see that the analytical model perfectly fits with the Westinghouse's one, so it's considered reliable.

3.2 Comparison of the non- and the regenerated models

	[P_n] kW	[T_4] °C	[G_4] kg/s	[T_3] °C	[HR] kcal/kWh	[η_g] %	[Speed] rpm
Analytical model of TG16	18807	409	122	788	3207	26.8	4850
Analytical model of TG16R	17823	410	121.2	788	2651	32.4	4850
Variation %	-5 %	-0.2 %	-0.6 %	0 %	-17 %	21 %	0 %

From the difference between the two model it's easy to say that regeneration involves two significant data:

- A reduction of power of 5%, due to the pressure at the turbine delivery and at the suction of the axial turbocompressor;
- An increase in the efficiency of 21%, thanks to combusted gases that heat air delivered by the compressor.

Chapter 4

Heat exchangers: state of art

Heat exchangers are used to exchange heat among fluids with different temperatures, directly or indirectly. These stuffs are used in the 90% of the production of the energy nowadays. Their applications are numerous, e.g. in power plants, food or chemistry industries, aerospace or nuclear industries and lots of others.

Heat transfer is made up by convection between fluids or conduction through the surface (almost never the irradiance is considered: just for special cases e.g. in aerospace because there is no air).

There are lots of classifications:

- According to heat transfer manner:
  - Direct contact heat exchangers, when the fluids contact directly or are mixed together to exchange heat;
  - Recuperative heat exchangers, when the fluids flow simultaneously through the exchanger in separate paths and exchange heat across the walls separating the fluids;
  - Regenerative heat exchangers, when only a single set of flow channels through a relatively massive solid matrix exist and the hot and cold fluids pass through the matrix alternately.
- According to ratio surface on volume: heat exchangers are considered compact if the surface area density is at least 300 m^2/m^3; a good manner to have the right compactness, when at least one of the fluids is a gas, is to add a flap on separation wall.
- According to different type of constructions:
  - Shell and tube;
  - Bayonet;

  - Double pipe;
  - Spiral;
  - Air cooled;
  - Plate.
- According to flow pattern the heat exchangers:
  - Co-current, when the two fluids flow in the same direction (also known as) parallel flow arrangement);
  - Counter-current, when the two fluids flow in opposite directions;
  - Cross flow, when the fluids move at right angles to each other.
- According to thermal exchange mechanism:
  - Natural or forced convection;
  - Radiation;
  - Phase change, condensation or boiling.

4.1 Compact heat exchangers

It's important to distinguish between a compact heat exchanger and a small one: even an exchanger with both compact surfaces is not necessarily small. For a given thermal and pressure drop specification, the size of an exchanger is a function of both geometrical compactness of the surfaces and of the prescribed performance parameters independently of the surfaces.

4.1.1 Compactness

To describe compactness there are some geometrical parameters, as:

- The hydraulic diameter, $d_h = \frac{4 \cdot A_c \cdot L}{A_s}$, where A_c is the flow area and A_s is the surface area within that low length. Alternatively, $d_h = \frac{4 \cdot V_s}{A_s}$, when the area varies with flow length and V_s is the wetted volume;
- The surface porosity, $\sigma = \frac{V_s}{V}$;
- The surface area density, $\beta = \frac{A_s}{V} = \frac{4 \cdot \sigma}{d_h}$, which should be higher, or at least equal, to 300 m^2/m^3;
- The heat transfer coefficient, $\alpha = \frac{Nu \cdot \lambda}{d_h}$, where Nu is the Nusselt number and λ is the thermal diffusivity.

The porosity affects the actual value of surface density. A typical value high performance plate-fin surface is 0.8, but if the hydraulic diameter increases it's hard to maintain such an high value because of high temperature and high pressure containment, that means a significant higher fin thickness to contain pressure, and because of the low material conductivity for higher thickness to maintain fin efficiency and surface effectiveness.
While the performance parameters are described by:

- The ratio j/f for flow or face area;
- The ratio j^3/f for volume.

Both depends on the Reynolds number, which is proportional to the hydraulic diameter, as described in the following equation:

$$Re = \frac{\rho \cdot v^2 \cdot {d_h}^2}{\eta \cdot v \cdot d_h} = \frac{\rho \cdot v \cdot d_h}{\eta} = \frac{v \cdot d_h}{\nu}$$

It's possible to see from this equation that Reynolds number indicates the relative importance of inertial to viscous forces in a fluid process.

The corresponding volume parameter that gives a direct measure of overall compactness is Pv, which considers both the geometry component (hydraulic diameter and porosity) and the porosity (f/j^3) component:

$$P_v = \frac{d_h}{\sigma} \cdot \left(\frac{f}{j^3}\right)^{1/2} = \frac{V}{\dfrac{\dot{m} \cdot Pr \cdot N^{3/2}}{4 \cdot (2 \cdot \rho \cdot \Delta p)^{1/2}}}$$

4.1.2 Compact surfaces

4.1.2.1 Plate surfaces

Plate Heat Exchangers (PHEs) are exchangers which uses metal plates to transfer heat between two fluids. The flux of the two fluids (hot and cold) are separated, as shown in *Figure 22*, and exposed to a large surface area because they are spread out over the plates: heat's transfer is facilitated and the speed of the temperature change is increased.

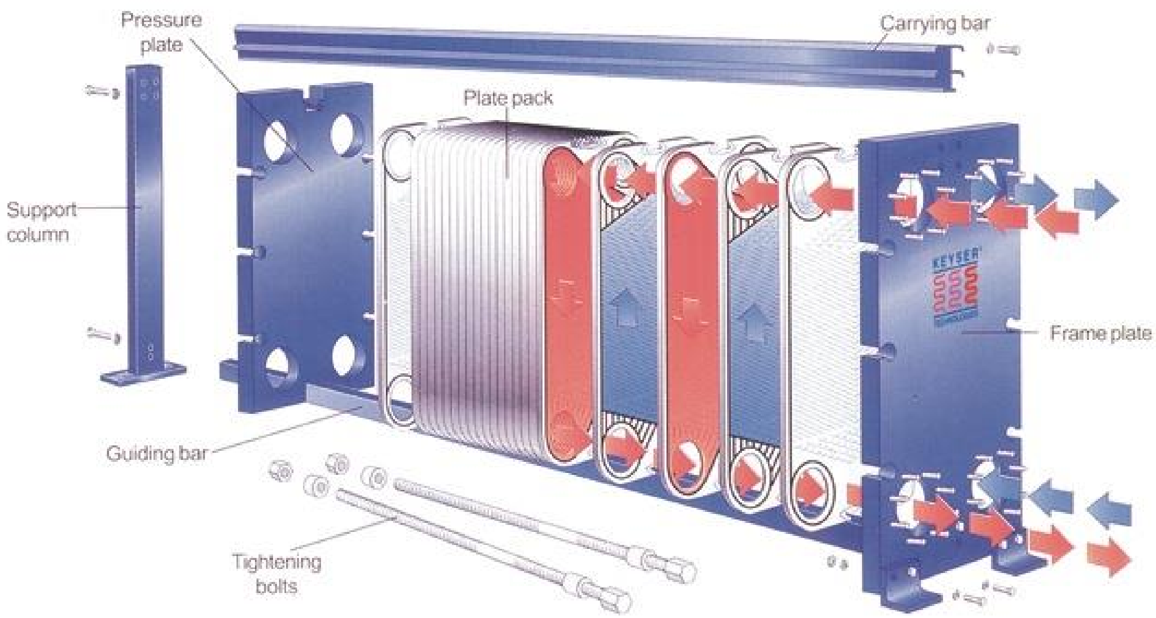

Figure 22. Plate fin surface

The thickness of the plate is very small (~ 1 mm) and they are usually spaced by rubber sealing gaskets, cemented into a section around the edge of the plates; so, the PHEs are usually quite thin: the plates are compressed together in a rigid frame to form an arrangement of parallel flow channels with alternating hot and cold fluids. Making the chamber thin ensures the majority of the volume of the liquid contacts the plate; moreover, the flow is turbulent thanks to troughs, and this maximize heat transfer.

The plate heat exchanger is usually used to transfer heat between medium- and lo-pressure fluids.

4.1.2.2 Plate fin surfaces

The plate-fin heat exchangers are made of layers of corrugation sheets (fins) separated by an equal number of flat metal layers (plates), as shown in *Figure 23*.

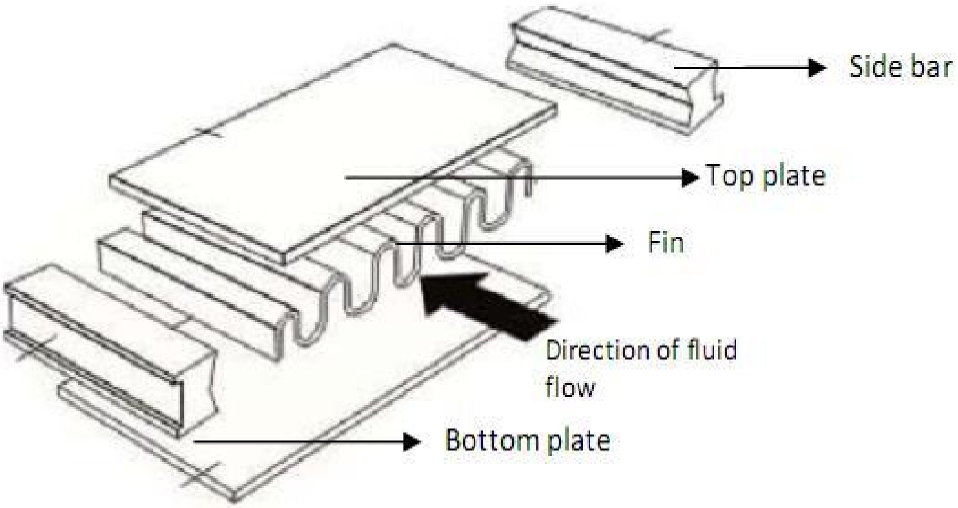

Figure 23. Components of plate-fin surfaces

The parting made by plates is needed to separate the flux of the hot fluid and the cold one in alternate layers, as in *Figure 24*. Fins are used for two main reasons: increase the structural integrity of the heat exchanger and to transfer heat from one fluid and the adjacent fluid.

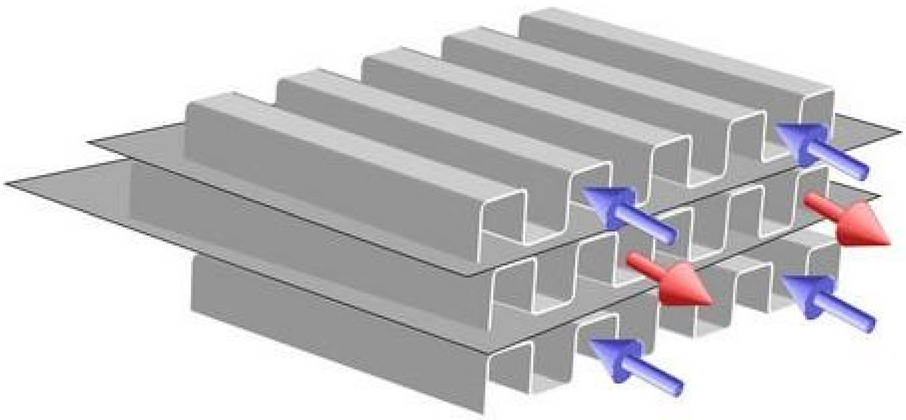

Figure 24. Separated layers of fluids

Plate-fin heat exchangers have a high flexibility because they can operate with any combination of gas, liquid and two-phase fluids. In fact, it is the commonest surface of all compact types.

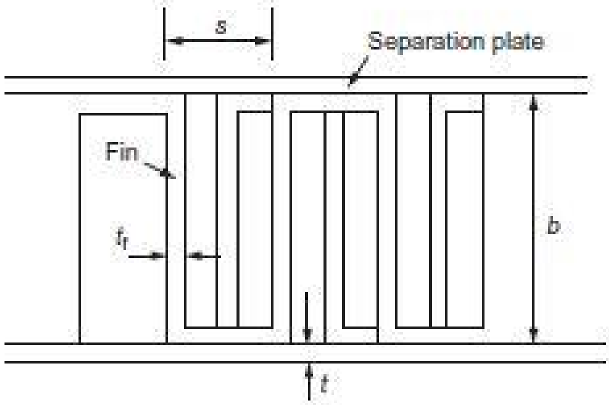

Figure 25. Basic geometry of rectangular plate-fin surface

In which:

- Fin thickness: t_f;
- Separation thickness: t;
- Plate-gap (distance between plate surfaces): b;
- Fin spacing: s.

The six basic fin configurations are:

- Plain rectangular (A);
- Plane triangular (B);
- Wavy (C);
- Offset strip (D);
- Perforated (E);
- Louvred (F).

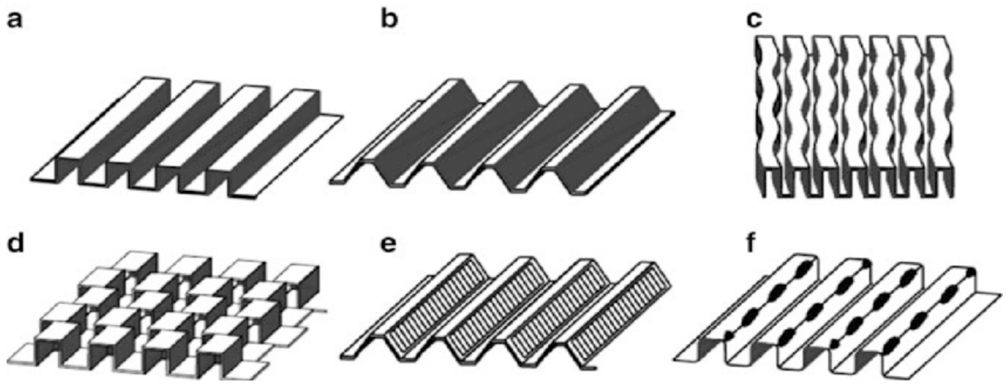

Figure 26. Plate fin surface types

For the plain type data are similar: they are straight fins that are continuous in the fluid flow direction; triangular and rectangular cross section are more common, although the fins can have any desired shape. Triangular arrangement can be manufactured at high speeds and less expensive than rectangular fins, but they are structural weaker for the same passage size and fin thickness and they also have a lower heat transfer performance than the rectangular one.
The offset-strip fin is the highest performer and the most widely used fin geometry, but these data should be used with some caution because most experimental surface were of 2 rows of fins, and only three were of 3 or 4 rows.
Concerning the wavy fin, it is competitive with the offset-strip; it form an be either the one of a folded fin strip between flat separating plates, with the corrugations being in the planes of the plates, or that of a corrugated fin interlaced by flat or round tubes.
For perforated fins, the perforation allows for lateral migration of flow, without the pressure drop penalty of the offset-strip. The interruptions give a slight performance improvement but are offset by loss of surface area and the friction factor is generally higher.
The louvred fin surface is formed by a rolling process (instead of a reciprocating press) which let it be much cheaper to produce, hence the form is widely used for mass production applications. The finning can be bonded to either separating plates or to tubes, even if the tubes themselves may not come under the strict definition of “compact”.

4.1.3 Maintenance

The maintenance of compact heat exchanger is a big aspect to be considered, because of fouling and corrosion: in many cases the reduced volume of the compact core has some advantages, as it can be readily shipped offsite for maintenance and cleaning.

An important aspect of maintenance in associated to gaskets used, for example, between plates: uniform gasket compression should be ensured.

Also checking all the equipment for the monitoring of the heat exchanger is part of the maintenance, to ensure that such equipment is used during the heat exchanger running, e.g. the cleaning of filters, or the pressure drop across the heat exchanger.

Fouling is a major barrier for wide use of Compact Heat Exchangers (CHEs), it consists in the accumulation of unwanted material on solid surfaces.

The main types of fouling are:

- Chemical fouling, it arises when constituent's reaction causes a viscous or solid layer on the heat surface. This sort of phenomenon is outside the control of the heat exchanger designer but can be minimized by careful control of the tube wall temperature, which is in contact with the fluid. When this type of fouling occurs, it must be removed by either chemical treatment or mechanical de-sealing processes and if the deposit turns from a tar to a hard coke, removal is very difficult;
- Biological fouling, it is caused by deposition and growth of organisms (like bacteria) within the fluid, which deposit on to the heat exchanger's surfaces. This effect can't be directly controlled by heat exchanger, but it depends on the choice of material, i.e. non-ferrous brasses kill certain organism. This type of fouling can be removed either by brushing process or by chemical process, e.g. using biocides which kill bacteria but not remove the biofilm;
- Crystallisation or precipitation fouling, it happens when a solute in the fluid is precipitated and crystals are formed, either directly on the heat transfer surface or in the fluid which subsequently deposits the crystals on the surface. This effect is generally avoided by pre-treatment of the fluid stream and by the continuous addition of chemicals to reduce deposit formation;
- Silting, it consists of the deposition of solid particles on a surface. The particles are quite easy to be removed, especially when they are large, but it becomes more difficult when particles are combined with other fouling mechanism. This phenomenon can be also controlled through the critical velocity of the fluid particle combination, or mounting

the heat exchanger vertically, when it's possible, to be helped by gravity to pull out the particles;

- Corrosion fouling, it occurs when chemical reactions involving the heat transfer surface are left on the heat transfer surface or when corrosion products from other parts of the system are transported on the heat transfer surface. The corrosion fouling can be minimized by the selection of corrosion resistant materials, such as stainless steels or nickel-based alloys;
- Freezing fouling, it's caused by the solidification of the fluid at the heat transfer surface, due the temperature of the fluid which becomes too low. It's quite easy to control and solve, especially for plate heat exchangers because of the low metal mass and low quantity of fluid involved: a raise in temperature is required;

There are also a few heat exchangers types, not all of them CHEs, which are designed specifically to handle fouled process streams.

Chapter 5

Design of the compact heat exchanger

Environmental issues impose increasingly stringent constraints and force the energy sector to develop new innovative technologies with reduced emissions.

This revolution could be applied both to new projects and to pre-existing installations; obviously for the first case the design process is easier, but most of the time also the redesign of an old plant is possible: sometimes not all the constraints could be satisfied (e.g. size or location of the exchanger, etc.).

Two years ago, a student tried to solve the regeneration of the TG16 turbogas plant using the Shell and Tube exchanger with 2 different regenerative efficiencies ($\varepsilon = 0.87$ and $\varepsilon = 0.75$), but both the solutions were too big for company needs.

The same process has been used for two different geometries of compact heat exchangers: Plate (PHE) and Plate-Fin (P-FHE).

5.1 Preliminary design of the heat exchanger with LMTD method

To evaluate the range of the efficiency of regeneration (ε) to consider and the effect that have to the dimensions of the exchanger, it's useful to design the heat exchanger with the LMTD method, which is the most common.
Referring to the scheme below shown in *Figure 27* it's possible to extrapolate temperatures and the thermal power.

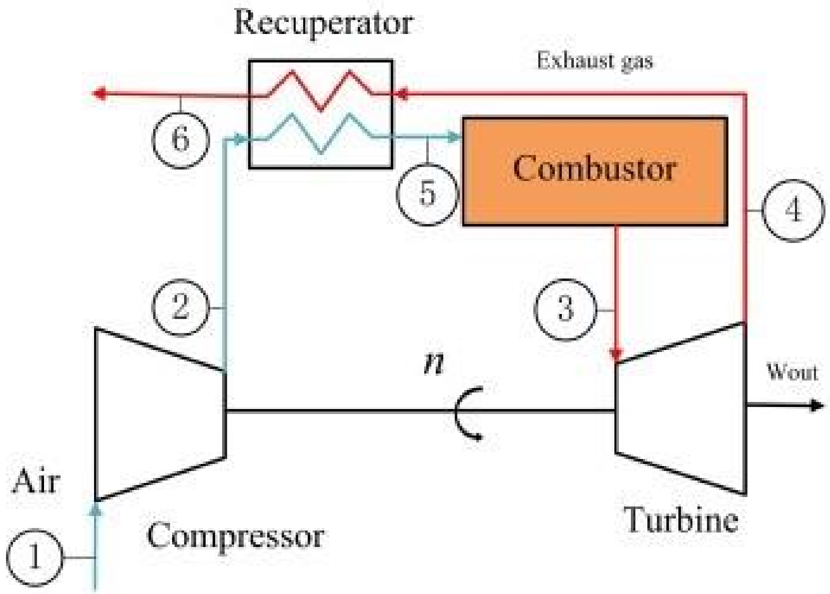

Figure 27. Referring scheme of a regenerative turbogas plant

Main references of the scheme are:

- T_2 = temperature of the compressor delivery;
- T_5 = temperature of the air at exchanger delivery;
- T_4 = temperature of exhausted gases;
- T_6 = temperature of exhausted gases at exchanger delivery;
- G_{air} = mass flow rate of air at the compressor delivery;
- $G_{exhaust}$ = mass flow rate of exhausted gases.

The Logarithmic Mean Temperature Difference (LMTD) method is an indicator of the average temperature difference between the hot and cold fluids in a heat exchanger. The procedure is articulated as following:

- Thermal power exchanged:

$$[\dot{Q}]\ \mathrm{MW} = G_{exch} \cdot c_{p,exch} \cdot (T_4 - T_6) = G_{air} \cdot c_{p,air} \cdot (T_5 - T_2)$$

- Mean temperature difference:

$$[\mathrm{LMTD}]°\,\mathrm{C} = \frac{(\mathrm{T}_4 - \mathrm{T}_5) - (\mathrm{T}_6 - \mathrm{T}_2)}{\ln\left(\frac{\mathrm{T}_4 - \mathrm{T}_5}{\mathrm{T}_6 - \mathrm{T}_2}\right)}$$

- Heat transfer coefficient:

$$[\mathrm{U}]\,\frac{\mathrm{W}}{\mathrm{m}^2\mathrm{K}}$$

- Configuration correction factor:

$$\mathrm{F} = \mathrm{f(geometry)}$$

- Corrected mean temperature difference:

$$[\mathrm{CMTD}]°\,\mathrm{C} = \mathrm{F} \cdot \mathrm{LMTD}$$

- Heat transfer surface area:

$$[\mathrm{A}]\,\mathrm{m}^2 = \frac{\dot{\mathrm{Q}}}{\mathrm{U} \cdot \mathrm{CMTD}}$$

- Temperature difference ratios:

$$\mathrm{R} = \frac{\mathrm{T}_2 - \mathrm{T}_5}{\mathrm{T}_6 - \mathrm{T}_4}$$

$$\mathrm{S} = \frac{\mathrm{T}_6 - \mathrm{T}_4}{\mathrm{T}_5 - \mathrm{T}_2}$$

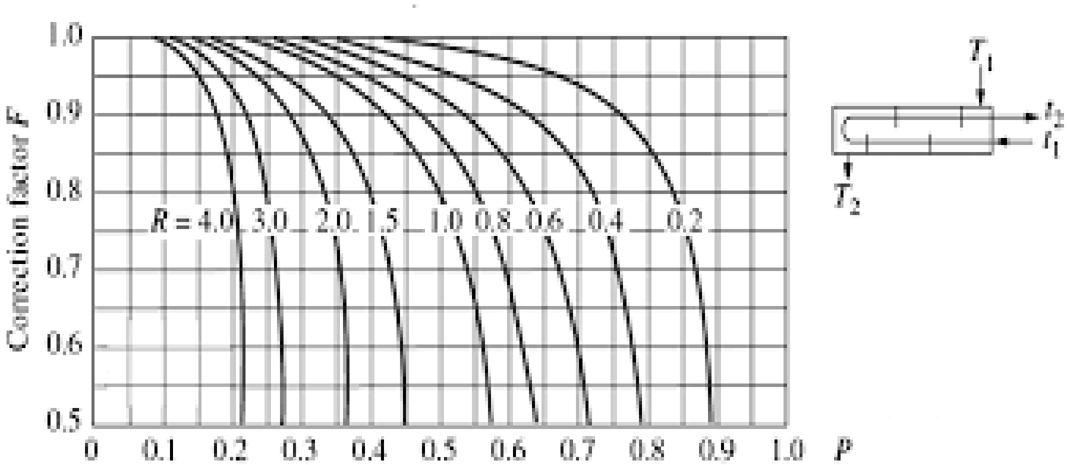

Figure 28. LMTD correction factor

This preliminary study is important to show the effect of the regeneration efficiency on:

- The size of the exchanger;
- The increase of the global efficiency;
- The decrease of the nominal power.

Results obtained with the LMTD preliminary design method are represented in *Figure 29*: percentage deviation is referred to the non-regenerated turbogas plant TG16.

ε	Pn [kW]	Scostamento (%)	ηg	Scostamento (%)
0,87	17823	-5,23	0,324	20,90
0,85	17818	-5,26	0,322	20,15
0,83	17821	-5,24	0,32	19,40
0,81	17836	-5,16	0,319	19,03
0,79	17822	-5,24	0,316	17,91
0,77	17833	-5,18	0,315	17,54
0,75	17836	-5,16	0,313	16,79
0,73	17841	-5,14	0,311	16,04
0,71	17842	-5,13	0,309	15,30
0,69	17847	-5,10	0,308	14,93
0,64	17856	-5,06	0,303	13,06
0,59	17877	-4,94	0,299	11,57
0,5	17898	-4,83	0,292	8,96
0,4	17909	-4,77	0,284	5,97
0,3	17939	-4,62	0,277	3,36

Figure 29. Performances of the exchanger as the efficiency of regeneration changes

From the data in the table, it's easy to say that the nominal power P_n is only slightly affected (≈ 5%) by the change of the regenerative efficiency ε, while the global efficiency of the plant η_g is mostly influenced by the efficiency of regeneration (3.6 ÷ 20.9%).

Obviously, when mass flow rates and temperatures T_2 and T_4 are constant and ε increases, it is expected that the global efficiency raises up, and so also the heat transfer area.

5.2 Design of the Plate heat exchanger with Aspen EDR software

The design activity of the Plate compact geometry is carried out using Aspen EDR (Exchanger Design and Rating) software.
Starting from the known data of the hot and cold fluxes of the TG16R, the size, the heat surface area and the cost of the heat exchanger has been obtained.
This study is articulated in 3 different phases:

1. Choice of the exchanger geometry, assuming regeneration efficiency $\varepsilon = 0.87$;
2. Choice of the exchanger geometry, assuming regeneration efficiency $\varepsilon = 0.75$;
3. Cost analysis and estimation of the payback period.

The choice of to use the two regenerative efficiency mentioned above is dedicated by the comparison with the results found in the previous study on the Shell and Tube heat exchanger. The evaluation must be done on several parameters: with the same input data it's necessary to consider the dimensions, the cost, the annual profit and the payback period.

5.2.1 Plate Heat Exchanger (PHE) with regeneration efficiency equal to 87%

Input data for the sizing of this exchanger are represented in the tables below:

- Fluxes data:

INPUT DATA	
$G_{exhaust}$	122.4 kg/s
G_{air}	121.2 kg/s
T_2	278° C
T_5	393° C
T_4	410° C
T_6	302° C
p_4	1.07 bar
p_2	7.64 bar
Max pressure drop	2 %

- Air composition:

Volume % of elements in dry air	
N_2	78.09
O_2	20.95
Ar	0.93
CO_2	0.03

- Exhaust gas composition:

Volume % of elements in exhaust gas	
N_2	75.73
O_2	16.12
Ar	0.9
CO_2	2.14
H_2O	5.11
NO_X	0.006

Plate Heat Exchanger Specification Sheet

1	Company: Annapaola Zingarelli					
2	Location:					
3	Service of Unit:	Our Reference:				
4	Item No.:	Your Reference:				
5	Date: Rev No.:	Job No.:				
6	CASE		HOT SIDE		COLD SIDE	
7	Fluid		Exhaust gas		Air	
8	Total flow	kg/s	122,39		121,19	
9	Flow per PHE	kg/s	6,4416		6,3784	
10	Pressure drop allow. / calc.	bar	0,105 /	0,10481	0,75 /	0,01379
11	Velocity between plates	m/s	11,91		1,77	
12	Wall shear stress	N/m^2	6,63		0,87	
13	Fouling margin	%	0		0	
14	OPERATING DATA		INLET	OUTLET	INLET	OUTLET
15	Liquid flow	kg/s	0	0	0	0
16	Vapor flow	kg/s	122,39	122,39	121,19	121,19
17	Operating temperature	°C	410,25	302,05	277,85	393,05
18	Operating pressure	bar	1,05	0,94519	7,5	7,48621
19	LIQUID PROPERTIES					
20	Density	kg/m^3				
21	Specific heat	kJ/(kg-K)				
22	Viscosity	mPa-s				
23	Thermal conductivity	W/(m-K)				
24	Surface tension	N/m				
25	VAPOR PROPERTIES					
26	Density	kg/m^3	0,53	0,57	4,74	3,91
27	Specific heat	kJ/(kg-K)	1,103	1,074	1,043	1,068
28	Viscosity	mPa-s	0,0317	0,0281	0,0275	0,0314
29	Thermal conductivity	W/(m-K)	0,0595	0,0521	0,0498	0,058
30	Relative molecular mass					
31	Dew point / bubble point	°C	/		/	
32	Latent heat	kJ/kg				
33	Critical pressure	bar	37,70329		37,70329	
34	Critical temperature	°C	-140,5834		-140,5834	
35	Total heat exchanged	kW	14573,8			
36	U^*	W/(m^2-K)	Clean condition:	59,4	Service:	59,4
37	LMTD / Effective MTD	°C	20,5		/	20,5
38	Heat transfer area	m^2	21542,3			
39	Stream heat transfer coeff.	W/(m^2-K)	121,3		116,7	
40	CONFIGURATION FOR EXCHANGER AND PLATE DETAILS					
41	Number of PHE in parallel	19	Heat transfer area/PHE		m^2	21542,3
42	Number of passes, hot side	1	Heat transfer area/plate		m^2	2,786
43	Number of passes, cold side	1	Plate chevron angles(s)		Degrees	60
44	Number of plates per PHE	409	Nominal plate thickness		mm	0,6
45			Nominal plate gap		mm	4,68
46	Mass empty / full of water	kg	5432,1		/	51072,5

Figure 30. API sheet for the PHE with regenerative efficiency equal to 0.87

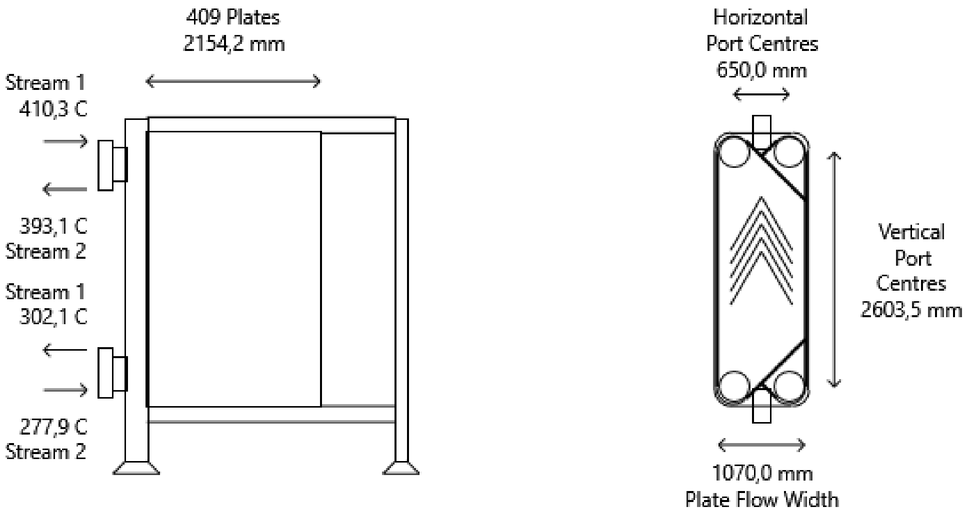

		Plate thickness	0,6 mm
Effective surface area	1133,805 m2	Compressed plate pitch	5,3 mm
Number of passes Stream 1 / 2	1 / 1	Area of each plate	2,786 m2
Effective channels Stream 1 / 2	204 / 204	Chevron angle (to horizontal)	60
Number of exchangers	19	Material type	SS-304
		Port diameter	400,0 mm

Figure 31. Geometry and configuration of the exchanger

It can be seen that:

- For this geometry, the software foresees 19 exchangers in parallel 1.07 meters long per each, for a total length of 20.33 m. Even if plate geometry is characterized by a big heat transfer area, to exchange 14,5 MW the number of exchangers in parallel (19) is too much;
- The pressure drop is close to 2%, therefore they are acceptable in these conditions, but if the number of exchangers were to decrease, pressure drop would increase considerably, which wouldn't be convenient.

5.2.2 Plate Heat Exchanger (PHE) with regeneration efficiency equal to 75%

Input data for the sizing of this exchanger are represented in the tables below:

- Fluxes data:

INPUT DATA	
$G_{exhaust}$	122.4 kg/s
G_{air}	121.2 kg/s
T_2	278° C
T_5	377° C
T_4	410° C
T_6	318° C
p_4	1.07 bar
p_2	7.64 bar
Max pressure drop	2 %

- Air composition:

Volume % of elements in dry air	
N_2	78.09
O_2	20.95
Ar	0.93
CO_2	0.03

- Exhaust gas composition:

Volume % of elements in exhaust gas	
N_2	75.73
O_2	16.12
Ar	0.9
CO_2	2.14
H_2O	5.11
NO_X	0.006

Plate Heat Exchanger Specification Sheet

1	Company: Annapaola Zingarelli					
2	Location:					
3	Service of Unit:	Our Reference:				
4	Item No.:	Your Reference:				
5	Date: Rev No.:	Job No.:				
6	CASE		HOT SIDE		COLD SIDE	
7	Fluid		Exhaust gas		Air	
8	Total flow	kg/s	122,39		121,19	
9	Flow per PHE	kg/s	6,4416		6,3784	
10	Pressure drop allow. / calc.	bar	0,105 /	0,10466	0,75 /	0,01338
11	Velocity between plates	m/s	11,74		1,7	
12	Wall shear stress	N/m²	6,55		0,83	
13	Fouling margin	%	0		0	
14	OPERATING DATA		INLET	OUTLET	INLET	OUTLET
15	Liquid flow	kg/s	0	0	0	0
16	Vapor flow	kg/s	122,39	122,39	121,19	121,19
17	Operating temperature	°C	410,25	318	277,85	377
18	Operating pressure	bar	1,05	0,94534	7,5	7,48662
19	LIQUID PROPERTIES					
20	Density	kg/m³				
21	Specific heat	kJ/(kg-K)				
22	Viscosity	mPa-s				
23	Thermal conductivity	W/(m-K)				
24	Surface tension	N/m				
25	VAPOR PROPERTIES					
26	Density	kg/m³	0,53	0,55	4,74	4,01
27	Specific heat	kJ/(kg-K)	1,103	1,078	1,043	1,065
28	Viscosity	mPa-s	0,0317	0,0286	0,0275	0,0309
29	Thermal conductivity	W/(m-K)	0,0595	0,0534	0,0498	0,057
30	Relative molecular mass					
31	Dew point / bubble point	°C	/		/	
32	Latent heat	kJ/kg				
33	Critical pressure	bar	37,70329		37,70329	
34	Critical temperature	°C	-140,5834		-140,5834	
35	Total heat exchanged	kW	12486,6			
36	U*	W/(m²-K)	Clean condition:	58,8	Service:	58,8
37	LMTD / Effective MTD	°C	36,59		/	36,59
38	Heat transfer area	m²	21859,9			
39	Stream heat transfer coeff.	W/(m²-K)	120,7		114,9	
40	CONFIGURATION FOR EXCHANGER AND PLATE DETAILS					
41	Number of PHE in parallel	19	Heat transfer area/PHE		m²	21859,9
42	Number of passes, hot side	1	Heat transfer area/plate		m²	2,786
43	Number of passes, cold side	1	Plate chevron angles(s)		Degrees	60
44	Number of plates per PHE	415	Nominal plate thickness		mm	0,6
45			Nominal plate gap		mm	4,68
46	Mass empty / full of water	kg	5511,8		/	51821,9

Figure 32. API sheet for the PHE with regenerative efficiency equal to 0.75

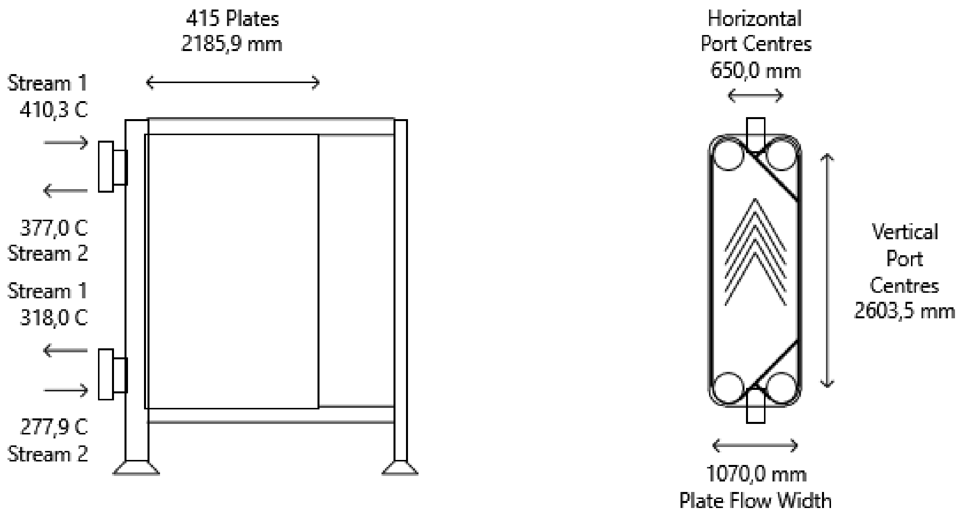

		Plate thickness	0,6 mm
Effective surface area	1150,520 m2	Compressed plate pitch	5,3 mm
Number of passes Stream 1 / 2	1 / 1	Area of each plate	2,786 m2
Effective channels Stream 1 / 2	207 / 207	Chevron angle (to horizontal)	60
Number of exchangers	19	Material type	SS-304
		Port diameter	400,0 mm

Figure 33.Geometry and configuration of the exchanger

It can be seen that:

- For this geometry, the software foresees 19 exchangers in parallel 1.07 meters long per each, for a total length of 20.33 m. The regeneration efficiency 0.75 is lower than those of the previous case 0.87, hence also the thermal power exchanged and the heat transfer area A_0 are reduced.

5.3 Comparison between all heat exchangers with regeneration efficiency equal to 87%

The Plate results are compared with those of the Shell and tube heat exchanger with the same regeneration efficiency $\varepsilon = 0.87$, obtained two year ago.

THERMAL POWER EXCHANGED = 14.6 MW		
	PLATE	**SHELL&TUBE**
A_0	21542 m^2	16673 m^2
Number of exchangers	19	7
Length	1.03 m	9.1 m
Max pressure drop	2%	3.5%
Total weight	5432 kg	43729 kg
Cost	1’571’148 €	2’077’709 €

From the comparison in the table it’s evident that, with the same thermal power exchanged, the Plate exchanger is better in terms of all the parameters considered. Especially, the total length of the exchangers is one third of the Shell and tube exchangers’ size.

5.4 Comparison between all heat exchangers with regeneration efficiency equal to 75%

The Plate results are compared with those of the Shell and tube heat exchanger with the same regeneration efficiency ε = 0.75, obtained two year ago.

THERMAL POWER EXCHANGED = 12.4 MW		
	PLATE	**SHELL&TUBE**
A_0	21860 m^2	13782 m^2
Number of exchangers	19	6
Length	1.03 m	8.4 m
Max pressure drop	2%	3.5%
Total weight	5512 kg	39268 kg
Cost	1’593’755 €	1’721’200 €

From the comparison in the table it’s evident that, even in this situation, with the same thermal power exchanged, the Plate exchanger is better in terms of all the parameters considered. Especially, the total length of the exchangers is half of the Shell and tube exchangers’ size.

5.5 Design of the Plate-Fin heat exchanger with Aspen EDR software

The design activity of the Plate-Fin compact geometry is carried out using Aspen EDR (Exchanger Design and Rating) software.

In this case input data are both those of the hot and cold fluxes and those on the geometry of the exchanger; unfortunately it's not possible to satisfy all the inputs, hence the software keep the size of the exchanger unchanged and modify the outlet temperature to reach convergence.

INPUT DATA	
$G_{exhaust}$	122.4 kg/s
G_{air}	121.2 kg/s
T_2	278° C
T_5	377° C
T_4	410° C
T_6	318° C
p_4	1.07 bar
p_2	7.64 bar
Max pressure drop	2 %
Number of exchangers	7
Length	1 m

The data in red are the one modified by the software to find an appropriate solution.

Other input elements used are the compositions of the two fluxes, listed below:

- Air composition:

Volume % of elements in dry air	
N_2	78.09
O_2	20.95
Ar	0.93
CO_2	0.03

- Exhaust gas composition:

Volume % of elements in exhaust gas	
N_2	75.73
O_2	16.12
Ar	0.9
CO_2	2.14
H_2O	5.11
NO_X	0.006

PLATE-FIN Heat Exchanger Specification Sheet

2	Company:									
3	Location:									
4	Service of Unit:	Our Reference:								
5	Item No.:	Your Reference:								
6	Date:	Rev No.:	Job No.:							
7	Stream i.d./fluid name				1/ Exhaust gas		2/ Air		3/	
8	Flow rate	Total		kg/s	122,39		121,19			
9		Vap./liq.	In	kg/s	122,39	0	121,19	0		
10		Vap./liq.	Out	kg/s	122,39	0	121,19	0		
11	Molecular weight	Vap.	In/Out		28,65644	28,65644	28,95999	28,95999		
12		Liq.	In/Out							
13	Density	Vap.	In/Out	kg/m³	0,54	0,56	4,83	4,3		
14		Liq.	In/Out							
15	Viscosity	Vap.	In/Out	mPa-s	0,0317	0,0296	0,0275	0,0298		
16		Liq.	In/Out							
17	Specific heat	Vap.	In/Out	kJ/(kg-K)	1,103	1,086	1,043	1,058		
18		Liq.	In/Out							
19	Thermal cond.	Vap.	In/Out	W/(m-K)	0,0595	0,0554	0,0498	0,0549		
20		Liq.	In/Out							
21	Temperature	In/Out		°C	410	345,68	278	345,68		
22	Operating pressure	In		bar	1,07		7,64			
23	Maximum allowable pressure drop			bar	0,22		0,5			
24	Heat load			kW	-8614,2		8614,2			
25	Calculated MTD			°C	7,33					
26	Fouling resistance			m²-K/W	0		0			
27	Core size			mm	Width	2000	Height	2132	Length	1000
28	Number of layers				94		94			
29	Fin code: Heat transfer fin				1		2			
30	Fin code: Distributor fin				1		2			
31	Heat transfer surface/core			m²	17676,2		17676,2			
32	Core opening size	In/Out		mm	40		40			
33	Nozzle size	In/Out		mm						
34	Calculated frictional pressure drop			bar	0,06157		0,00795			

Figure 34. ALPEMA sheet for the PFHE with regenerative efficiency equal to 0.51

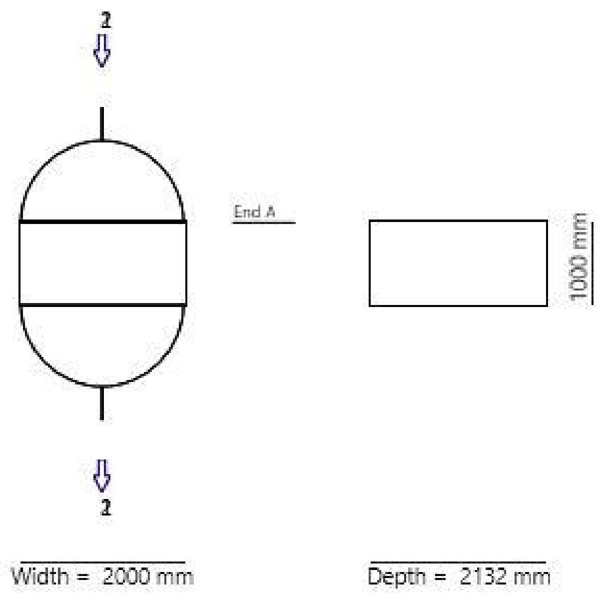

Figure 35. Geometry and configuration of the exchanger

It can be seen that:

- For this geometry, 7 exchangers in parallel are used. Each one is 1 meter long, for a total length of 7 m;
- To respect the dimensional constraints the regeneration efficiency becomes very low (ε = 51%), therefore the global efficiency increases slightly;
- The pressure drop is close to 2%, therefore they are acceptable in these conditions.

Chapter 6

Estimation of the payback period

It's important to evaluate the feasibility of the investment and the annual profit to compare all the exchangers designed.

The payback period is the period of time in which the profits reach the value of investment expenses, from than moment on, all the revenues are earnings. To estimate the payback period of the initial investment, two different aspects are considered:

- Annual fuel savings, thanks to regeneration (positive effect);
- Power reduction (≈ 5%) in terms of kW per year.

For each exchanger it's necessary to consider the annual monetary flows with the following hypothesis:

- Operating hours per year are 8000;
- Fuel cost is 0.27 €/Sm^3;
- Energy cost is 0.138 €/kWh;
- Different additional cost needs to be added to the estimated cost of the exchanger:
  - Engineering/Administration;
  - Shipping;
  - Site preparation;
  - Plumbing;
  - Mechanical works;
  - Commissioning.

With these assumptions is simple evaluate monetary fluxes: for each type of geometry, yearly fuel and energy cost must be calculated.

6.1 Evaluation of the annual monetary fluxes

The study is articulated in the following steps:

- Fuel volume flow rate: $[\dot{V}_{fuel}]\frac{Sm^3}{h} = \frac{\dot{m}_{fuel}}{\rho_{fuel}}$;
- $[\text{Annual fuel consumption}]\frac{Sm^3}{year} = \dot{V}_{fuel} \cdot h_{year}$;
- [Annual fuel cost] €/year = annual fuel consumption · fuel cost;
- [Annual power produced] kWh/year = $P_n \cdot h_{year}$;
- [Annual energy produced] €/year = annual power produced · energy cost.

The summary quantities referring to the TG16 turbogas plant are shown below:

Base load operation of TG16	
Power	18807 kW
Fuel mass flow rate	1.52 kg/s
Fuel specific gravity	0.72 kg/Sm^3
Fuel volumetric flow	7620 Sm$_3$/h
Operating hours per year	8000 h/year

Annual fuel cost of TG16	
Cost of fuel	0.27 €/Sm^3
Fuel consumption	59316375 Sm^3/year
Annual fuel cost	16015421 €/year

Annual energy cost of TG16	
Cost of energy	0.138 €/kWh
Energy consumption	150456000 kWh/year
Annual energy cost	20762928 €/year

The same calculations must be done for the regenerative plant with all the 3 options:

TG16R with PHE and ε = 0.87	
Annual fuel cost of TG16R	
Cost of fuel	0.27 €/Sm^3
Fuel consumption	40565021 Sm^3/year
Annual fuel cost	10952556 €/year
Annual energy cost of TG16	
Cost of energy	0.138 €/kWh
Energy consumption	142584000 kWh/y
Annual energy cost	19676592 €/year

TG16R with PHE and ε = 0.75	
Annual fuel cost of TG16R	
Cost of fuel	0.27 €/Sm^3
Fuel consumption	59316375 Sm^3/year
Annual fuel cost	16015421 €/year
Annual energy cost of TG16	
Cost of energy	0.138 €/kWh
Energy consumption	142688000 kWh/year
Annual energy cost	19690944 €/year

TG16R with P-FHE and ε = 0.51	
Annual fuel cost of TG16R	
Cost of fuel	0.27 €/Sm^3
Fuel consumption	50917428 Sm^3/year
Annual fuel cost	16015421 €/year
Annual energy cost of TG16	
Cost of energy	0.138 €/kWh
Energy consumption	143016000 kWh/year
Annual energy cost	19736208 €/year

6.2 Calculation of annual cash flow

To calculate the annual cash flow and the payback period it's important to consider the annual fuel saving and the annual power loss. The difference between these two effects allows to calculate the annual profit.

To analyse the potential payback period of the investment, the annual cash flow is compared to the total cost of the exchanger. In particular, the following formulas are used:

- [Annual fuel saving] €/year = annual fuel cost of TG16R - annual fuel cost of TG16;
- [Annual power loss] €/year = annual energy cost of TG16R – annual energy cost of TG16;
- [Annual profit] €/year = annual fuel saving – annual power loss.

6.2.1 Case 1: Plate exchangers with regeneration efficiency 87%

Previous formulas are used for the calculations of the Plate heat exchanger with regeneration efficiency equal to 87%:

Annual fuel saving	+5062865 €/year
Annual power loss	-1086336 €/year
Annual profit	3976529 €/year

PLATE HEAT EXCHANGER	
Total cost of exchanger	5547677 €
Payback period	1 year and 5 months

These results are compared with those of the Shell & Tube heat exchanger with the same regeneration efficiency obtained two years ago:

SHELL & TUBE HEAT EXCHANGER	
Total cost of exchanger	4155418 €
Payback period	2 years and 1 month

6.2.2 Case 2: Plate exchanger with regeneration efficiency 75%

The same method of Case 1 is used for the calculations of the Plate heat exchanger with regeneration efficiency equal to 75%:

Annual fuel saving	+3543454 €/year
Annual power loss	-1071948 €/year
Annual profit	2471506 €/year

PLATE HEAT EXCHANGER	
Total cost of exchanger	4065261 €
Payback period	1 year and 8 months

These results are compared with those of the Shell & Tube heat exchanger with the same regeneration efficiency obtained two years ago:

SHELL & TUBE HEAT EXCHANGER	
Total cost of exchanger	3442400 €
Payback period	2 years and 2 months

6.2.3 Case 3: Plate-Fin exchanger with regeneration efficiency 51%

Previous formulas are used for the calculations of the Plate heat exchanger with regeneration efficiency equal to 87%.

Annual fuel saving	+2267701 €/year
Annual power loss	-1026720 €/year
Annual profit	124 0981 €/year

PLATE HEAT EXCHANGER	
Total cost of exchanger	3840921 €
Payback period	3 years and 2 months

For this case of study there isn't any other heat exchanger with the same regeneration efficiency with which is it possible to compare.

Chapter 7

Conclusions

The aim of Ethos Energy S.p.A., and of this thesis too, is to regenerate the TG16 turbogas pant, designing a heat exchanger to have an amount of fuel saved: combusted gases pass in the heat exchanger and then dumped into the environment, in this way the air out of the compressor is re-heated, before the inlet of the combustor. The plate data of the TG5B (a similar turbogas plant of smaller dimensions) and Westinghouse' studies were used as guideline of this script.
The choice to use compact heat exchangers is due to the failure of the attempt with shell and tube exchangers of two years ago.
The design activity was carried out with Aspen EDR software. This program has two different compact geometries: Plate and Plate-Fin. Both geometries have been developed.
For the first type of geometry, analysing the results emerged to obtain a model that replicate the Westinghouse study, therefore with a regeneration efficiency of 87%, the Plate heat exchanger is not sufficient: it fits much better than the Shell and Tube model, but the dimensional constraints are not respected. Even if the plate geometry is characterized by a considerable heat transfer area, the mass flow rates are big (about 120 kg/s) and the temperature differences ΔT are too small to allow a high exchange efficiency.
Another attempt with the same geometry was therefore made to reduce the regeneration efficiency by bringing it to 75%, in order to reduce the dimension of the exchanger and construction costs. Again, the study did not lead to a convenient result.
With the other geometry the model optimization consists of an iterative process which aimed at minimizing the size of the exchanger to correctly fit in Ethos' dimensional constraints: for the plate fin geometry it's possible to check different geometrical parameters as input data (e.g. number of exchangers in parallel, dimensions of each exchanger, layer types, fins, etc.), but it's not possible to satisfy both input data of the fluxes to have a regeneration efficiency of 87% or 75% and those of the geometry of the exchanger.

The only way to satisfy the dimensional constraints given by Ethos is to modify the outlet temperature of the exchanger to reach convergence. In this way the regenerative efficiency falls down to 51% and the global efficiency increases too little to justify the choice to use this heat exchanger.

In conclusion, the plate heat exchanger with $\varepsilon = 87\%$ is the best compromise between size and costs: even if it doesn't fit exactly in the size request, the PHE significantly reduces the length respect to the Shell and tube solution; moreover it's cheaper and with an higher annual profit than the other exchangers.

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Chapter 8

Appendix

Two years ago, Ethos Energy and Politecnico had already started the study of the regeneration of the TG16 turbogas. Their first try was to use a shell and tube heat exchanger, but after the design and the numerical results, it was evaluated that that type of heat exchanger doesn't fits good or the researched solution.
Let's evaluate that search and analyse the results.

Shell and tube heat exchanger
The first attempt for the regeneration of the TG16 was a shell and tube heat exchanger. This type of exchanger is the most common one and it's characterized by a shell with a bundle of tubes inside it. One fluid flows through the tubes while the other flows over the tubes (in the shell) to transfer heat to the first fluid using convection.

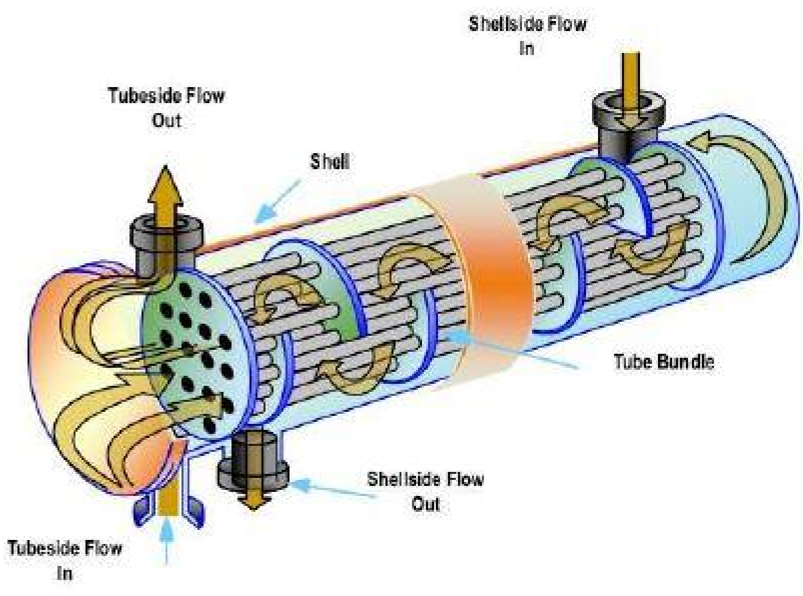

Figure 36. Shell and tube heat exchanger

Design of this type of heat exchanger depends in American association Tubular Exchanger Manufacturers' Association (TEMA), based on:

- Interaction between fluxes (Shell types);
- Thermodynamic conditions and flux's category;
- Thermal stress between tubes and shell.

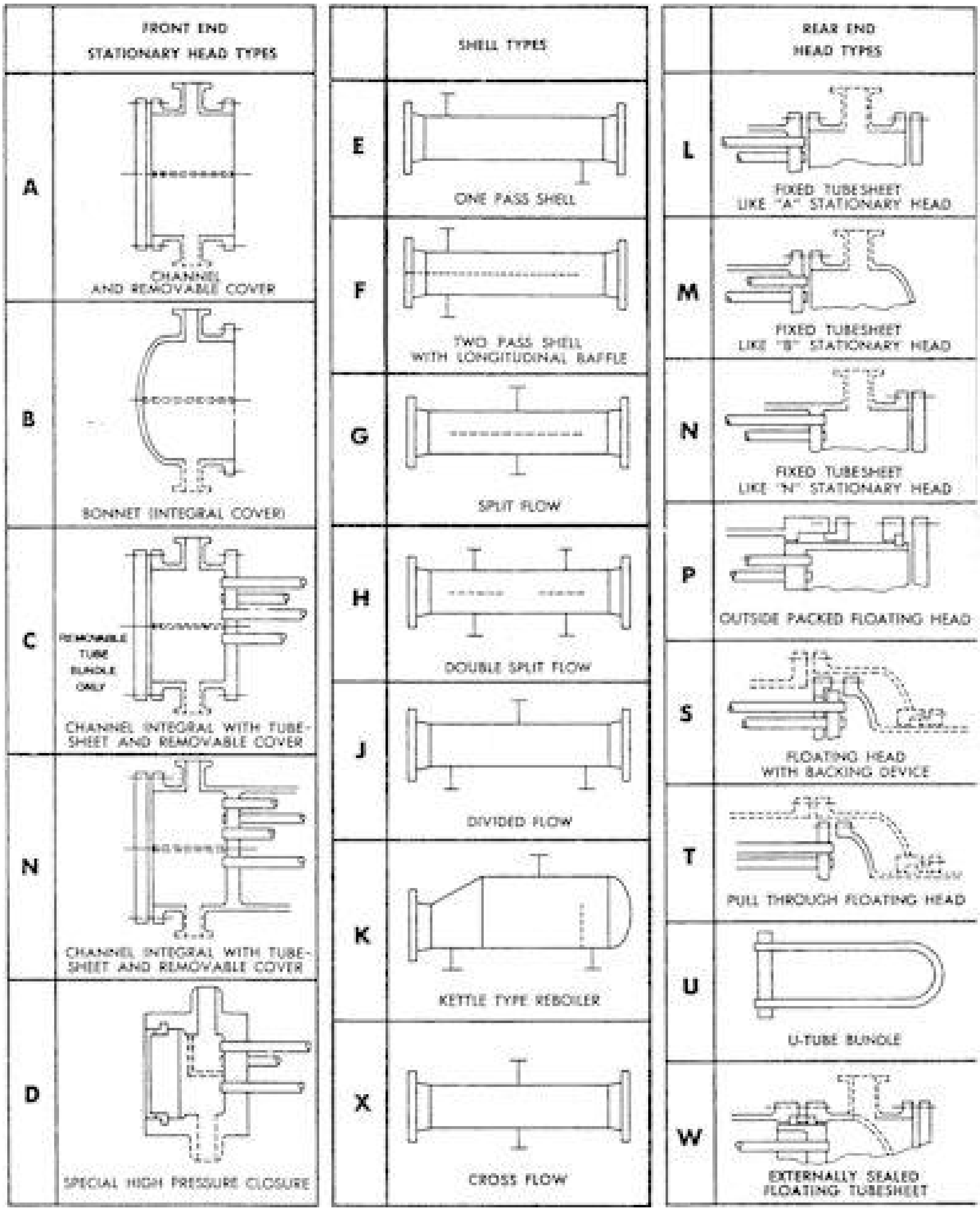

Figure 37. TEMA classification of shell and tube heat exchanger

The exchanger is made up three different components:

- Baffles;
- Tubes bundle;
- Tubesheets.

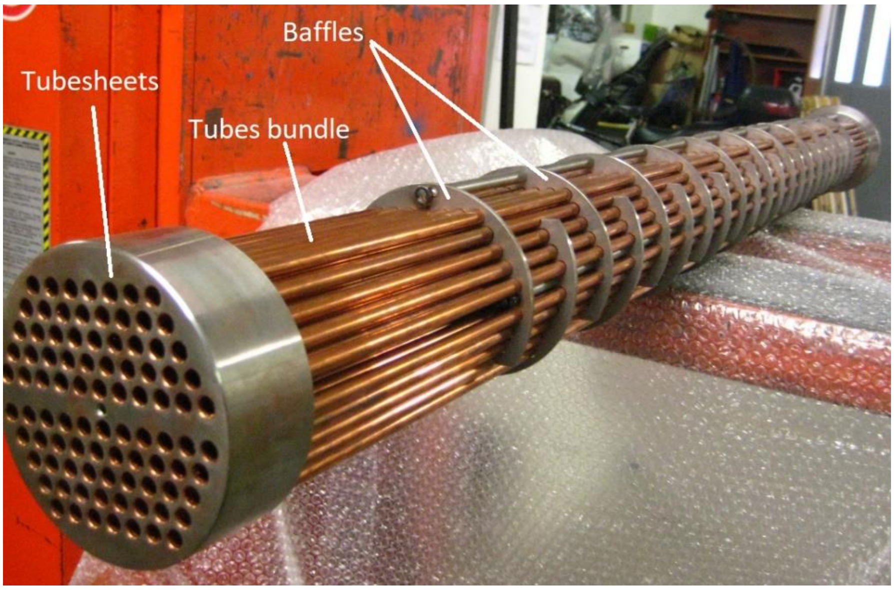

Figure 38. Shell and tube components

Tubes could have smooth or rough surface, with different properties:

- External diameter varies from 6.35 to 50.8 mm;
- Length is up to 30 m;
- Materials are steel, plastic or ceramics;
- Processes are forming, drawing, extrusion or welding;

Pay attention to keep $\frac{1}{5} < \frac{\text{shell diameter}}{\text{length}} < \frac{1}{15}$.

Most common configurations are the triangular (a) and the square (b) ones. The first one increases the tube density, while the second one gets easier the mechanical cleaning.

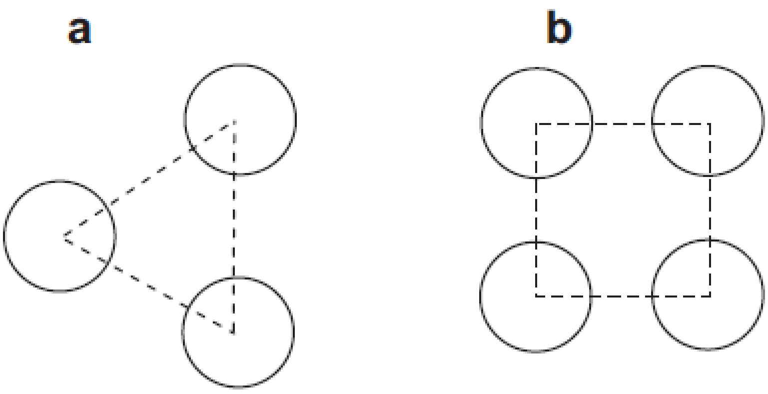

Figure 39.Triangular and square configurations of tubes

Nevertheless, shell and tube exchangers have some malfunctions due to fouling, vibrations and/or non-uniform flux. Especially:

- Fouling consists in a layer of solid materials, that flows inside the tubes thanks to the fluids. This phenomenon leads to a consistent reduction of the surface, so the fouling materials prevent efficient heat transfer and reduce the efficiency of the heat exchanger.

Figure 40. Fouling phenomena

  There are different types of fouling:
  - Corrosion fouling;
  - Particulate fouling;
  - Precipitation fouling;
  - Chemical reactions fouling;
  - Bio fouling;
  - Freezing fouling.

  To avoid fouling mechanical cleaning could be used, e.g. deposits which are weakly attached to the exchanger wall can be washed off by operating the exchanger at higher fluid velocity;
- Vibrations, that could be in the tubes and/or acoustic (even higher than 160 dB), caused by the flows of the fluid. The probability that vibrations occurs is less than 1%, but the effects if this happens are too expensive, so vibrations must be considered during the design process of the heat exchanger;
- Non-uniform distribution of the flux leads to a decrease in the exchanger performances and its damage. It could be caused by many factors, i.e. device tolerances.

Design of the shell and tube heat exchanger

Using the software Aspen EDR, the model has been designed.

The design process was divided into three phases:

- First phase: choice of the geometry of the heat exchanger with efficiency of regeneration $\varepsilon = 0.87$;
- Second phase: choice of the geometry of the heat exchanger with efficiency of regeneration $\varepsilon = 0.75$;
- Third phase: economy analysis.

Both the case with $\varepsilon = 0.87$ and with $\varepsilon = 0.75$ had the following composition of air and of exhausted gases:

% of element in dry air		**% of element in combusted gases**	
N_2	78.09	**N_2**	75.73
O_2	20.95	**O_2**	16.12
Ar	0.93	**Ar**	0.9
CO_2	0.03	**CO_2**	2.14
		H_2O	5.11
		NOx	0.006

First phase

Input data for the TG16R with $\varepsilon = 0.87$ were:

[G_{exh}] kg/s	122.4
[G_{air}] kg/s	121.2
[T_2] °C	278
[T_5] °C	393
[T_4] °C	410
[T_6] °C	302
[p_4] bar	1.07
[p_2] bar	7.64
Pressure losses	2 %
Fouling coefficient [m^2K/W]	0.0001

Two of the different geometries proposed by TEMA were analysed:

- BEM, a heat exchanger with tube bundle fixed by mechanical rolling to two tube sheets for easy mounting on skid. This configuration of the tube bundle allows the cleaning of the inside of the tubes with mechanical systems, while the cleaning of the outer side can only be done using chemical means;

Figure 42. BEM geometry of the shell and tube heat exchanger

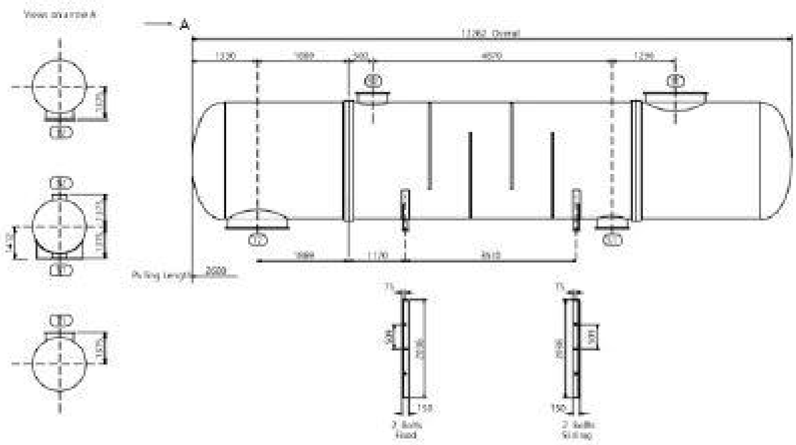

Figure 41. Longitudinal view of a BEM shell and tube

This geometry has a flux interaction of 1/1, so area which exchange heat is small and it would be necessary to use 16 exchangers of 12 m long for each. It would have been a too big area;

- CFN, a heat exchanger with flat head bolted to the tubes bundle. It has a flux interaction of 2/2, but to exchange a thermal power of 14.5 MW 7 exchanger of 9 m long per each are needed.

  Even if this configuration is better than the BEM one, the area needed to cover the total amount of heat to exchange is always too large.

Below are the results which were obtained for both the geometries:

	BEM	CFN
[A_0] m^2	32140	16673
Number of heat exchanger	16	7
[Length of each heat exchanger] m	12.3	9.1
[Maximum pressure losses] %	2	3.5
[Total weight] kg	35435	43729
[Costs analysis] €	4178873	2077709

Second phase

Input data for the TG16R with ε = 0.75 were:

[G_{exh}] kg/s	122.4
[G_{air}] kg/s	121.2
[T_2] °C	278
[T_5] °C	377
[T_4] °C	410
[T_6] °C	318
[p_4] bar	1.07
[p_2] bar	7.64
Pressure losses	2 %
Fouling coefficient [m^2K/W]	0.0001

The different efficiency of regeneration led to a reduction in the thermal power exchanged, then to a decrease of global area of heat exchanged.

As for the first phase, the same geometries proposed by TEMA were analysed: BEM, CFN.

	BEM	CFN
[A_0] m^2	6264	13782
Number of heat exchanger	9	6
[Length of each heat exchanger] m	9.9	8.4
[Maximum pressure losses] %	2	3.5
[Total weight] kg	30079	39268
[Costs analysis] €	1104798	1721200

Comparing the two cases, it’s possible to see that for the second phase there was a consistent reduction of A_0 and of costs.
However, the choice of the Shell and Tube type of heat exchanger was not good because of the huge dimensions needed. This is the reason why Ethos Energy decided to pass to compact heat exchanger for the regeneration of the turbogas TG16.