Fossil Fuel Power Plants
Intermediate CO2 removal

CONTENT

1_Introduction

Carbon dioxide concentration into the atmosphere is a major issue. Unfortunately, this gas is not the only one responsible for atmosphere pollution, other gases produced during fossil fuel combustion present similar danger like greenhouse effect, toxicity and acid rains. These gases are mainly water vapour, nitrogen oxides, carbon monoxide, hydrocarbons and fine particles. As such, all efforts to limit their emission should be undertaken. In this document, all these gases are often included in the wording “Carbon dioxide” for simplification.

Page “Power plants – EXTERNAL CO2 removal” of this web site, describes how carbon dioxide produced in the combustion chamber of a gas turbine may be absorbed at a relatively high pressure and low temperature in a contactor by a physical solvent. This high pressure is achieved by a **compression - expansion loop** activated by the energy of the hot gas exiting the power turbine. **This gas process is carried out externally to the gas turbine** to the burnt gas exhausting the gas turbine at roughly atmospheric pressure. https://yvcharron.com/index.php/fossil-fuel-power-plants/

This mode of operation is transferrable to a steam boiler burning carbonated fuels (including coals) discharging to the atmosphere a polluted gas containing carbon dioxide at a relatively high temperature. The higher the exhaust temperature, the higher is the pressure level reached by the compression – expansion loop.

The present document describes a significantly different system where the carbon dioxide is capted upstream the combustion chamber. In that instance, the compression – expansion loop is operated from the turbine air compressor exit pressure level and not from the atmospheric pressure level. **This gas process occurs internally** (or more precisely at an intermediate position) **to the gas turbine.**

External and internal treatments of the polluted air (mainly, carbon dioxide and NOx) presents some similarities described below (section 4) also relative advantages and disadvantages (section 6).

To determine the applicability of this intermediate air process, it is important to look at the characteristics of the different types of gas turbines (mainly aero derivative and heavy duty gas turbines - section 2) and the other cases of internal air process (mainly regeneration and intercooling - section 3).

2_Gas turbine types and some characteristics

The mode of operation of gas turbines is presented in page “Power plants – EXTERNAL CO2 removal” https://yvcharron.com/index.php/fossil-fuel-power-plants/ and resumed below.

There are two main types of gas turbines: the aero derivative and the heavy duty (or industrial). They differ mainly by their application, their performance and their mode of construction.

As suggested by their name, **aero derivative gas turbines** are based on a central core engine derived from the aircraft industry (jet engine) discharging hot gases with a high momentum (in an aircraft, the backward engine thrust displaces forward the aircraft) towards a last turbine stage usually named **Power turbine** or **Free turbine**. Aircraft engines often use several concentric shafts connecting separately compression to expansion sections with similar **specific velocity**. This aerodynamic adaptation associated to relatively high pressure (30 to 40 Bar abs) and temperature at the combustion chamber (above 2200 °C) provides a relatively high thermal efficiency, of the order of 40%. As a result, less heat is rejected to the atmosphere due to an exhaust gas temperature usually lower than 500°C (an average of 450°C).

GT Reference	Supplied power MW	Thermal efficiency per cent	Mass flow out kg/s	Exhaust temper. °C	Fuel flow MT/year	CO2 (1) mass per cent	Air (2) dilution factor
T11	25.7	38.9	84	443	44	4.6	3.47
T12	28.3	29.2	126	516	65	4.5	3.54
T13	27.6	37.3	83	511	50	5.2	3.05
T14	29.5	38.0	94	490	52	4.8	3.31
T15	39.7	32.5	137	503	82	5.2	3.05
T16	41.9	39.5	126	447	70	4.9	3.26
T17	52.5	42.4	159	428	83	4.6	3.49
T18	80.7	32.6	280	528	166	5.2	3.08
T19	123	33.8	404	538	245	5.3	3.01
T20	226	35.6	602	589	427	6.2	2.58

Figure 2.1 – Characteristics of some aero derivative (in blue) and industrial (in red) gas turbines. Note (1) CO2 mass fraction, in per cent, refers to the mass flow exiting the turbine. Note (2) air dilution is the ratio between the total inlet air flow and the stoichiometric part.

Industrial gas turbines differ from aeronautical design in that frames, bearings and blading are of **heavier construction**. They include only one driving shaft from the extreme compression end (air entrance) to the extreme expansion end (air release to atmosphere) preventing aerodynamic speed adaptation between turbine elements. As such, the thermal efficiency is smaller (30 to 35 %). As a result more heat is rejected to the atmosphere due to an exhaust gas temperature often larger than 500°C (an average of 550°C). For this reason, these turbines are often used in combined cycles (Combined Heat and Power) to significantly improve the efficiency of the whole system. At the combustion chamber exit, the gas pressure ranges between 15 and 25 Bar abs while the combustion gas temperature ranges from 1500 to 2000 °C.

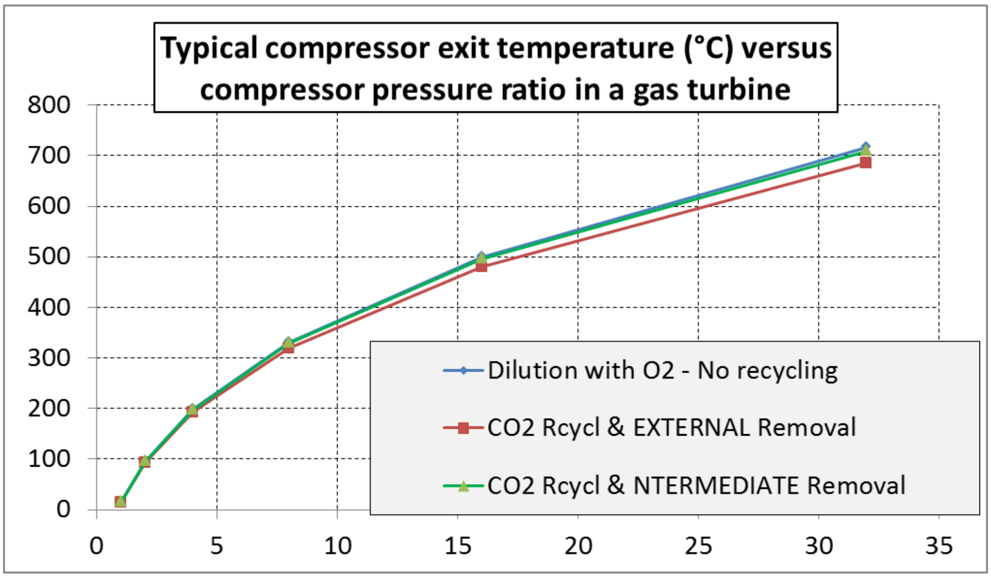

Figure 2.2 – Typical air compressor exit temperature versus pressure ratio in a gas turbine.

The relatively large compression pressure ratio of an aero derivative gas turbine (30 to 35) **induces a relatively high temperature at the compressor exit** (approaching 700 °C). In contrast, the lower pressure ratio of a heavy / industrial gas turbine (15 to 20) induces a lower temperature at the compressor exit (of the order of 500 °C° - figure 2.2).

The temperature and the pressure levels at the GT compressor exit are two important parameters for the design of CO2 removal at an intermediate position in a gas turbine as it will be seen in this document.

3_Examples of gas turbine intermediate air process

The normal air process in a gas turbine is as follows: ambient air is suctioned by a compressor, pushed towards a combustion chamber then released in an expander where the hot gas expansion provides sufficient energy to drive both the air compressor and an external load (an electric generator or a mechanical brake).

It is meant here by **external and intermediate air process** (or treatment for CO2 removal) the following:

- **In an external treatment**, the air flow circulates straight through the gas turbine with the gas treatment occurring downstream the gas turbine.
- **In an intermediate treatment**, the air flow exits the compressor, enters in a process unit then returns to the combustion chamber before being released by the power turbine. The gas treatment occurs inside the normal course of the gas turbine air flux.

Below are two examples of intermediate treatment of the gas turbine air flux.

3.1_Gas turbines with air regeneration

The hot air exhausting the gas turbine (following hot air expansion) is directed towards a heat exchanger (H.E.R) transmitting heat to a flux of air discharged from the compressor. Following heat exchange, the compressed air is directed towards the combustion chamber (C.C.) following which it follows the normal air flux. This additional heat transmitted to the compressed air reduces the fuel flow rate for a constant supplied power. This process increases the gas turbine efficiency. See figure 3.1.

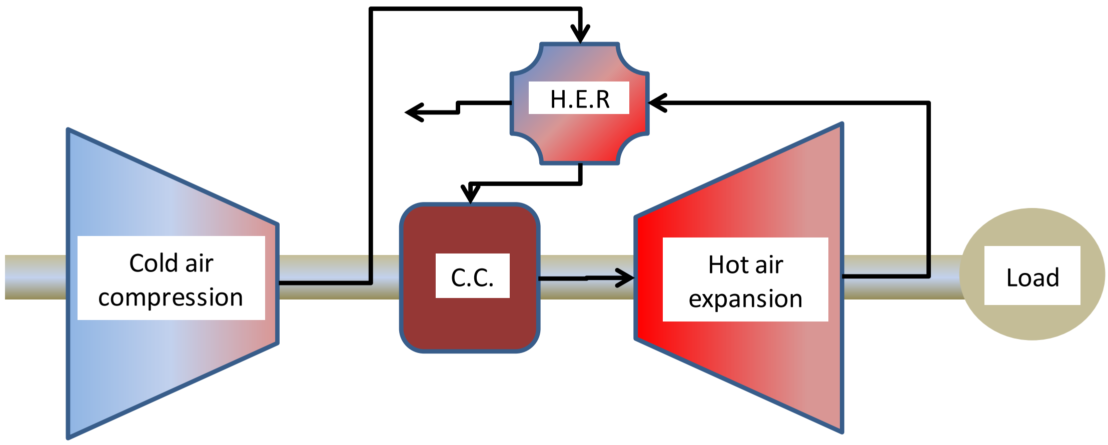

Figure 3.1 – Air regeneration in a gas turbine enhancing the temperature of the compressed air before entering the combustion chamber.

This air regeneration is generally applied to heavy turbines where the intermediate pressure and temperature are not very high (less than 20 Bar and 400 °C) while the exhaust temperature is relatively high (above 500 °C) and the construction of heavy turbines is suitable for such air rerouting. For the same reasons, this is less suitable to aero derivative gas turbines.

3.2_Gas turbines with compressed air intercooling

Gas compression is facilitated when the suctioned gas is at low temperature. Consequently, cooling the air during compression permits to approach an isothermal process and to reduce the absorbed power. This is sometimes applied to some gas turbines where the compressed air is extracted after the first compression section (Sect 1 - figure 3.2), cooled down in a heat exchanger (H.E.I) before being redirected towards the second compression section (Sect 2). This permits to reduce the compression energy providing more energy to the driven load.

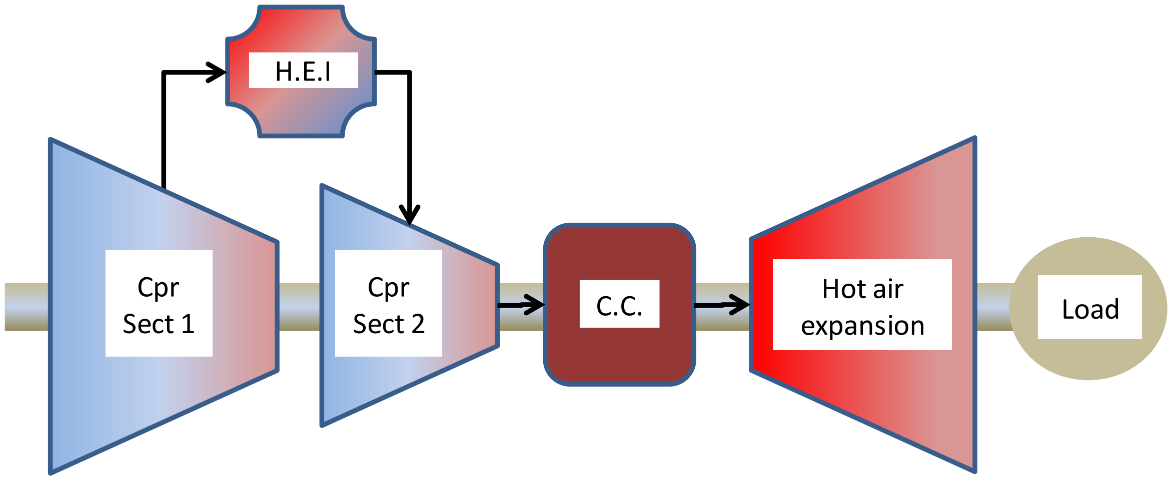

Figure 3.2 – Air intercooling in the compression section of a gas turbine to reduce the power absorbed by the compression sections.

4_Description of an intermediate CO2 removal system

4.1_Stoichiometric combustion

In a gas turbine, the air flows, successively, in the compression section, the combustion chamber and the expansion section before being discharged into the atmosphere. In some cases, additional equipment's are inserted between the expansion section and the exhaust stack for energy recovery purposes.

In a reciprocating engine, the combustion occurs in **a stoichiometric condition** meaning that the number of hydrocarbon molecules match exactly the number of oxygen molecules to provide a complete combustion without any molecule of oxygen in excess.

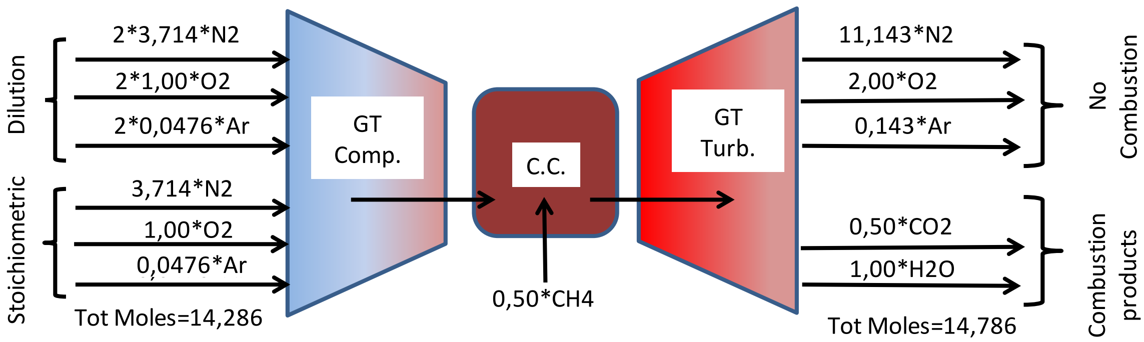

Figure 4.1 – Gas turbine operating with an excess of oxygen. In the present case the air dilution is equal to 3.

Contrary to a reciprocating engine, **the combustion in a gas turbine operates with an excess of air** therefore of oxygen. This is required to prevent the temperature entering into the expansion section to reach a too high level and also to permit a sufficiently high mass flow rate across the gas turbine i.e. high power. In this mode of operation, it is said that the air entering the gas turbine is diluted.

In this document, we define the air dilution factor by the ratio of the number of oxygen molecules entering the gas turbine and those satisfying a stoichiometric condition. In the following, the air dilution factor is equal to 3 (figure 2.1).

4.2_Combusted air recycling with EXTERNAL air treatment

This mode of operation represents the case treated in page "Power plants – EXTERNAL CO2 removal" of this web site.

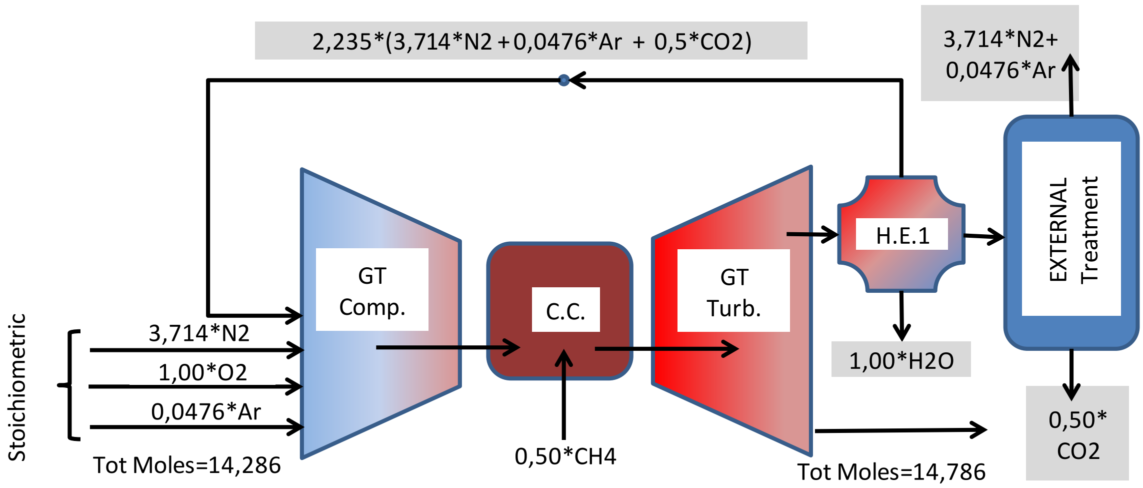

Figure 4.2 – External treatment of rejected gases, gas turbine operating in stoichiometric condition. Air dilution provided by cooling and recycling burnt gases.

In this system, the hot burned gases are cooled down by heat exchanger H.E.1 (figure 4.2) before being partially recycled in the gas turbine while the other part is directed towards the gas treatment system. As a reminder, this system includes a compression – expansion loop and a contactor operating at the highest pressure of that loop allowing dissolution of acid gas molecules (mostly carbon dioxide but also NOx) into a physical solvent. The gas recycling is required to provide the largest CO2 fraction into the gas turbine flux to ease its dissolution in the contactor. As a consequence, **the gas turbine operates in a stoichiometric condition.**

4.3_Combusted air recycling with INTERMEDIATE air treatment

Like in the previous paragraph, combusted air is recycled in order to provide a stoichiometric combustion in the gas turbine. However, in the present case, the air treatment occurs in between the air compressor exit and the combustion chamber inlet (figure 4.3). This permits to elevate the bottom (lowest) pressure of the compression – expansion loop (CpR - XpR – figure 4.4) requiring a lower pressure rise in the CpR-XpR loop (compared to an external treatment) to reach a suitable pressure in the contactor.

This method presents however a significant disadvantage. Carbon dioxide molecules being generated in the combustion chamber, the fraction of the burnt gas which is not recycled is rejected into the atmosphere. As such, in the case of a dilution factor of 3, the fraction of CO_2 rejected into the atmosphere is equal to 40 % of the carbon dioxide produced in the combustion chamber (see figure 5.1).

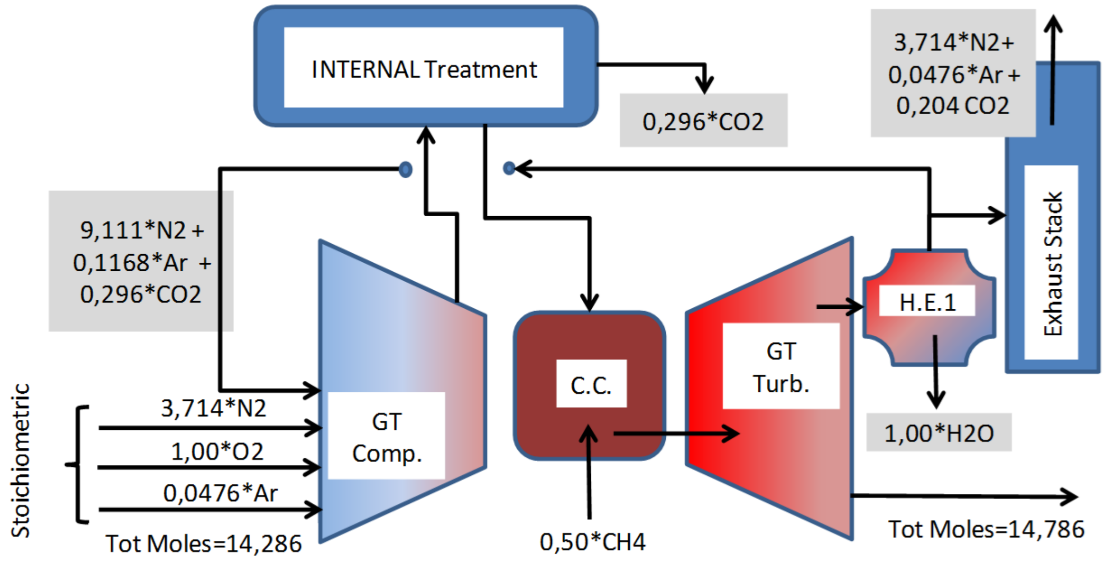

Figure 4.3 – Intermediate treatment of compressed gases, gas turbine operating in stoichiometric condition. Air dilution provided by cooling and recycling burnt gases.

At the outlet of the gas turbine compression section the air flow exits the gas turbine casing (Figure 4.4). It is successively cooled in heat exchanger HC1, compressed by CpR, cooled down again in HC2 to enhance sour gas dissolution before entering the contactor where liquid solvent is injected in extremely fine droplets (large liquid contact surface) to facilitate the absorption of the sour gas into the physical solvent. Downstream the contactor, the sweet gas is heated in heat exchanger HC3 before entering the expander XpR then heated again to recover all residual heat available from heat exchanger HC1. Heat exchanger HC3 receives the total heat available from HC2 and also a fraction of heat at high temperature from HC1 to get the maximum power delivered by expander XpR.

The maximum pressure level at the contactor is reached when the XpR delivered power equals to the CpR absorbed power representing the equilibrium status of the compression – expansion loop. A regulating system is required both to start the operation of the loop and to control the speed variation of the mechanically coupled machines (XpR and CpR – note that the mechanical link is not represented on figure 4.4).

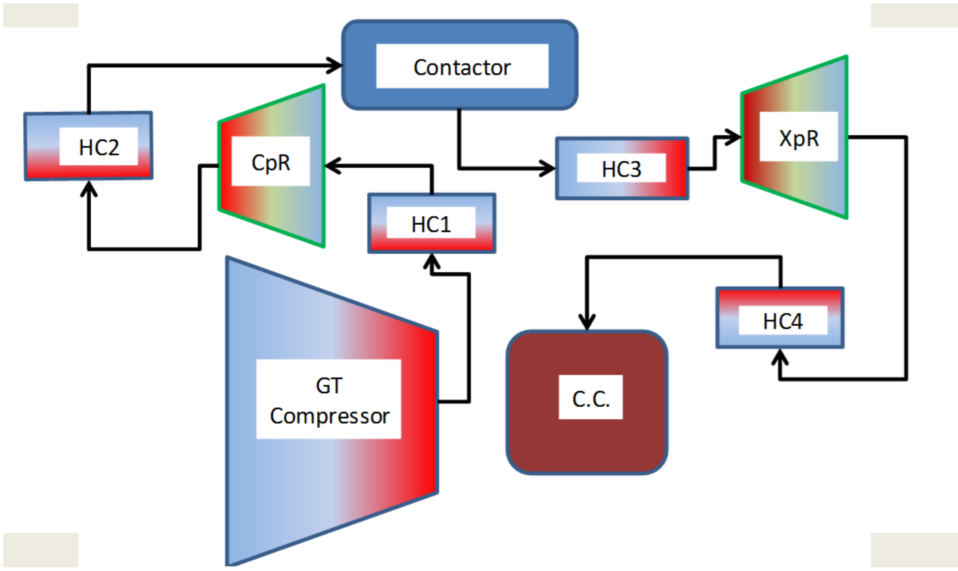

Figure 4.4 – Intermediate treatment of compressed gases between main compressor exit and combustion chamber inlet including compression CpR and expansion XpR sections, a contactor and four heat exchangers HC1 to HC4 (In reality, two vessels)

5_System performance

5.1_Fractions of sour gases absorbed and rejected to the atmosphere

According to gas turbine design, the degree of air dilution is adjusted (or fuel mass flow rate) to provide an acceptable temperature at the exit of the combustion chamber preventing damages to the first stages (fixed vanes and rotating blades) of the high pressure turbine section.

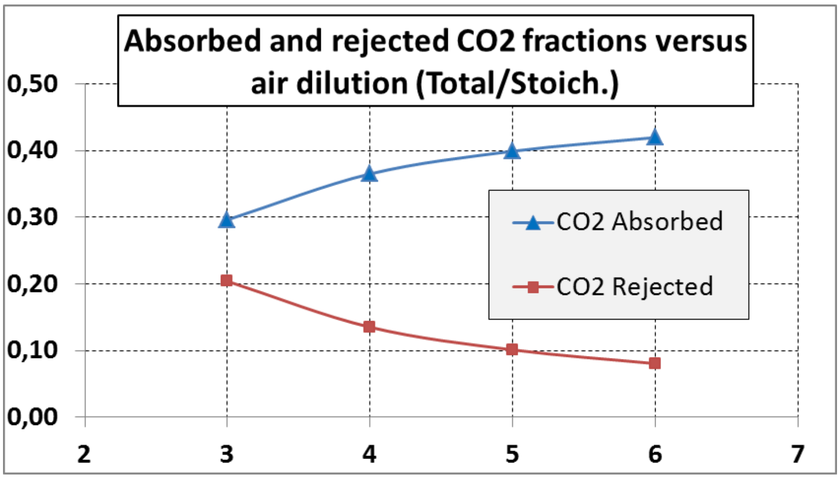

Figure 5.1 – Fractions of carbon dioxide (or sour gases) absorbed and rejected to the atmosphere versus gas turbine air dilution at gas turbine entrance.

A large fraction of air dilution corresponds to a large fraction of recycled burnt gas and therefore to a large fraction of capted sour gases, therefore, **to a low**

fraction of rejected sour gases. To the contrary when the air dilution factor is small, the fraction of recycle burnt gas is small and the fraction of rejected sour gases is relatively large. The variation of absorbed and rejected sour gases versus air dilution is represented on figure 5.1.

5.2_CpR compression, XpR expansion and heat exchangers duties

This section 5.2 has been modified in April 2021.

Calculations have been performed on the following basis:

- An air compressor with 16/1 pressure ratio. This value was considered as it is the **more difficult case for reaching the contactor pressure**. This point is developed in annexe 1.
- The contactor pressure is equal to 80 Bar abs.
- Following a 16/1 pressure ratio, the compressor exit pressure is 16 Bar abs and the exit temperature is 496 °C
- One mole of di-oxygen per second that is 32 g/s or 411 g/s total inlet flow.

	CASE	16	Bar a Air Cmp out		Tin XpR	450	°C		
	°C	°C		kJ/kg/°C	g/mole		kg/s	kJ/s	
HC1	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	
	495	216	14,29	1,021	28,75	1,395	0,411	117	
CpR	Pd/Ps	Ps_B.A	Mcmp	Pd_B.A.	Pw_kW				
	5,00	16,00	0,354	80,00	94,4				
CpR	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	
	20	245	14,29	1,021	28,75	1,395	0,411	-94	
HC2	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	HC2+HC3
	245	20	14,29	1,021	28,75	1,395	0,411	94	Marg Kj/s
HC3	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	2,6
	20	245	13,99	1,025	28,43	1,399	0,398	-92	2,8%
HC3	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	HC1+HC2
	245	450	13,99	1,025	28,43	1,399	0,398	-84	+HC3 Marg
XpR	Pd/Ps	Ps_B.A	Mexp	Pd_B.A.	Pw_kW	XpR/CpR margin			Marg Kj/s
	0,200	80,00	0,243	16,00	-95,3	0,95	Percent		1,4
XpR	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	1,2%
	450	216	13,99	1,025	28,43	1,399	0,398	95	
HC4	Tin	Tout	Moles	Cp	M W	K	Flow	Heat	
	216	295	13,99	1,025	28,43	1,399	0,398	-32	
		200	°C : Temperature reduction in compression - expansion loop						

Figure 5.2 – CpR, XpR and heat exchangers operation based on 495°C inlet temperature in the compression – expansion loop. Data based on one mole of oxygen at GT entrance.

Calculation results are presented on figure 5.2. Heat exchanger HC2 provides 94 kJ/s to HC3 which receives 92 kJ/s (2.8% margin) to rise the temperature from 20 to

245 °C at XpR inlet. Heat exchanger HC1 provides 84 kJ/s to HC3 to rise the temperature from 245 °C to 450 °C at XpR inlet.

Expansion in XpR reduces the temperature from 450°C to 216°C. Heat exchanger HC1 provides 32 kJ/s to HC4 to rise the temperature from 216 °C to 295 °C before entering the combustion chamber. Heat balance between HC1, HC3 and HC4 indicates 1.2 % margin.

The compressor absorbed power is 94 kW (based on 80 % efficiency) while the expander supplied power is 95 KW (based on 85 % efficiency) i.e.1% margin.

As a result, the compression – expansion loop permits to rise the air pressure from 16 to 80 Bar abs at which pressure level, acid molecules (carbon dioxide and NOx) are removed from the polluted gas in the contactor. The successive polytropic compression and expansion and heat exchanger temperature limitation induce a reduction in the gas temperature of approximately 200 °C (from 495 to 295 °C).

5.3_CpR and XpR polytropic efficiencies

The compression – expansion loop performance has been evaluated for several values of polytropic efficiencies. In these calculations it is admitted that the expander efficiency is 5 points above the compressor efficiency, that is, when the compression efficiency is equal to 80 %, the expander efficiency is assumed equal to 85 %. The result is presented on figure 5.3.

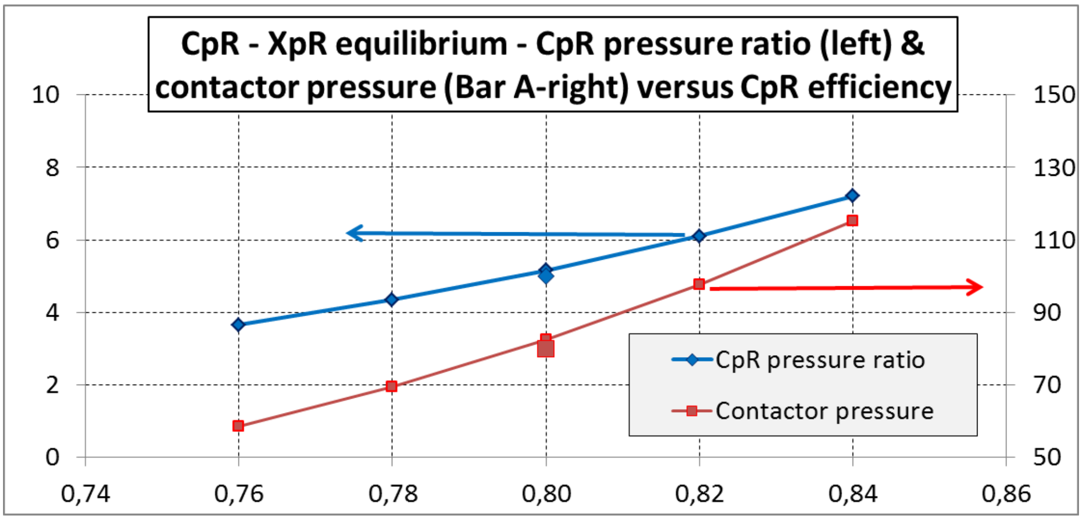

Figure 5.3 – Equilibrium pressure at contactor for several values of CpR compressor efficiency. XpR efficiency is 5 points above CpR efficiency.

When the CpR efficiency reaches 84 % the pressure ratio is 7.2 and the contactor pressure is 115 Bar abs. When this efficiency is reduced to 76 %, these values become, respectively, 3.65 and 68.4 Bar abs.

The polytropic efficiency of centrifugal and axial machines varies considerably with numerous parameters, particularly, flow and pressure coefficients and also the

Reynolds number. In his document, efficiency values are for indication only as the study was performed for a unit mass (One mole of di-oxygen).

Several documents indicate that the efficiency of axial turbines may reach 95 % at optimum conditions. See https://core.ac.uk/download/pdf/76985234.pdf from OPTIMIZATION of SMALL-SCALE AXIAL TURBINE FOR DISTRIBUTED COMPRESSED AIR ENERGY STORAGE SYSTEM Thesis by Ali Bahr Ennil – University of Birmingham. See also https://oatao.univ-toulouse.fr/17882/1/Binder_17882.pdf from Nicolas BINDER « Aéro-thermodynamique des Turbomachines en Fonctionnement Hors-Adaptation ». It has also to be mentioned the case of the large hydraulic turbines very slightly exceeding an efficiency of 95 % under certain conditions.

5.4_Incidence of air compressor pressure ratio on contactor pressure

As it can be seen on figure 2.2, the compressor exit temperature varies considerably with the compressor pressure ratio, reaching, for example 600 °C for a pressure ratio of 23 and considerably higher values for greater pressure ratio. This increase in temperature provides a larger energy exchange in the compression – expansion loop permitting the loop to operate with an increased pressure ratio, therefore, the contactor to operate at a higher pressure level.

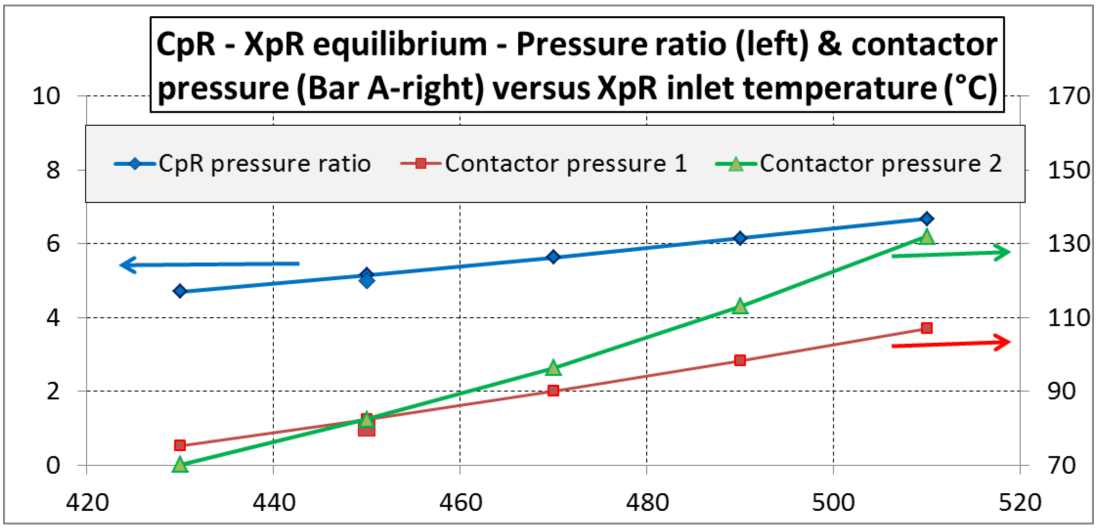

Figure 5.4 – Equilibrium pressure at contactor for several values of XpR inlet temperature – Temperature decreased in compression-expansion loop is assumed equal to 46°C. "Contactor pressure 1" is based on a main compressor exit pressure of 16 Bar abs.

Variation in the compression – expansion loop is presented on figure 5.4 with left and right ordinates representing, respectively, the loop pressure ratio and the pressure reached at the contactor versus the XpR inlet temperature. These calculations were performed assuming a temperature decrease in the compression – expansion loop of 46 °C (496 °C at main compressor exit, 450 °C at XpR inlet). See figure 5.2.

On figure 5.4, the "Contactor pressure 1" line is based on a main compressor pressure ratio of 16/1. On that basis, the contactor pressure varies of 0.40 Bar per °C at the inlet of the XpR expansion. An increase in temperature in the compression – expansion loop is the consequence of an elevation in pressure ratio in the main air compressor. This is represented on “Contactor pressure 2” where a temperature of 510 °C at XpR inlet (corresponding to a temperature of 555°C at main compressor exit) is obtained when the main air compression reaches a pressure ratio of 19.75 /1. On that basis, the contactor pressure varies of 0.77 Bar per °C at the inlet of the XpR expansion.

5.5_Gas turbine overall performance

This section 5.5 has been modified in April 2021.

5.5.1- Gas turbine without CO2 removal – Conventional design

The overall performance of a gas turbine operating with a 16/1 main air compressor pressure ratio is presented on figure 5.5.

Gas turbine features:	**No CO2 removal**	**CO2 removal – Pressure loss compensation**	**CO2 removal – Heat loss compensation**
Main GT compressor exit pressure & Equivalent CpR inlet pressure	16 Bar abs 16 Bar abs	19 Bar abs 16 Bar abs (2)	19 Bar abs 16 Bar abs (2)
Main GT compressor exit temperature & Comb.ion chamber inlet temperature	496 °C 496 °C	544°C 295 + 48 °C (3)	544°C 496 °C
Main GT compressor absorbed power (1)	202 KW	222 KW	222kW
Overall GT turbine supplied power (1)	337 kW	337 kW	337 kW
Fuel supply (1)	380 KW	380 KW	454 kW
GT turbine delivered power to ext. load (1)	135 kW	115 kW	115 kW
GT thermal efficiency	35.6 %	30.2 %	25.3 %

Figure 5.5 – Gas turbine performance without and with intermediate CO2 removal. Note (1): Data based on one mole of di-oxygen; Note (2): Considering pressure losses in CpR – XpR loop; Note (3): temperature reduction due to inefficiencies

Power data are presented for one mole of di-oxygen and water vapour therefore half - mole of methane and carbon dioxide. Based on stoichiometric condition (burnt air recycling) and an inlet dilution of 3, the temperature at the combustion chamber exit is 1428 °C. Based on a GT expansion efficiency of 80 %, the expansion power is 337

KW, the compressor absorbed power is 202 kW and the fuel heating is 380 kW. For a conventional turbine (no CO_2 removal), **the thermal efficiency is 35.6 %**.

5.5.2- Gas turbine with CO2 removal – Pressure and heat losses

To compensate for pressure losses in the compression – expansion loop, the pressure ratio of the main air compressor is increased from 16/1 to 19/1. This pressure increase represents either a 3 Bar pressure compensation at 16 Bar abs (18.75%) or a 15 Bar compensation at 80 Bar abs (also 18.75%). The increase in pressure ratio provides an increase in temperature of 48 °C at the compressor exit. This pressure rise is obtained by an increase in the GT compressor absorbed power (222 kW) **reducing the thermal efficiency to 30.2 %.**

To compensate for heat losses and temperature reduction in the compression – expansion loop (temperature at the combustion chamber inlet reduced from 496 to 295 °C) an additional 74 kW has to be supplied in the combustion chamber **reducing the thermal efficiency to 25.3%** (this takes into account the additional heat provided by the increase in pressure rise from 16/1 to 19/1).

5.5.3- Gas turbine with CO2 removal – Heat and energy recovery

Three types of heat and energy may be recovered. They include:

a) ***Residual heat from heat exchanger HC1***. Heat at a flow rate of 0.411 kg/s and 216 °C (82 kW) could be transferred, if required, either to a motor cycle (Carnot efficiency of 40 %) or to a solvent boiler. The first option has not been considered. With the second, it is assumed that the boiler requirement is approximately 1% of the fuel heat rate (454 kW) that is **4.5 kW** (figure 5.5).
b) ***Carbon dioxide pressurisation***, from 1 Bar a (following recycling) to 80 Bar a where the gas contacts the liquid solvent pumped to the same pressure level. This represents a potential energy of 9 kW. Depressurising together the liquid solvent - gas mixture with a let-down valve does not permit to recover energy. However, energy could be **recovered** by a two phase flow turbine: **up to 4.5 kW** assuming 50 % two phase efficacy (section 5.6). It should be noted that this power is significantly greater than the power required to pump the liquid solvent.
c) ***CpR – Xpr energy excess.*** In the present case, **1 kW may be recovered** (figure 5.2). This figure increases significantly with the combustion chamber pressure.

Recovering the above 5.5 kW mechanical energy and adding them to the driving load (115 kW) **the thermal efficiency is raised to 26.5 %.**

Recovering the 4.5 kW heat energy (item a) raises **the thermal efficiency to 27.5** % (8 points below the thermal efficiency of a conventional gas turbine). It will be seen in the annexe A1 that when the air compressor pressure ratio increases, the difference between the two thermal efficiency values reduces. Annexe A1 is based on a constant contactor pressure.

5.6_Physical solvent regeneration

Following sour gas absorption by fine solvent droplets, these latest fall at the bottom of the contactor vessel where the physical solvent is extracted for depressurisation in a second vessel where the sour gas is evaporated and directed to a treatment unit or transport system. The purified liquid at the bottom of the depressurisation vessel is pressurised in view of its reinjection at the top of the contactor vessel for a new cycle of sour gas removal.

The physical solvent may be depressurised, at least, in three different manners:

- Through a **let-down valve** with the pressure reduction occurring at constant enthalpy. In that instance, there is no energy recovery.
- Through a **single phase flow turbine** providing some energy recovery. Unfortunately, this equipment, using radial hydraulic cells, is only efficient in the first stage where the gas volume fraction is relatively small. In the following stages, the two phase flow efficacy is considerably reduced, the turbine recovering very little energy. See: https://yvcharron.com/index.php/two-phase-flow-pumps/)
- Through a **two phase flow turbine** recovering most of the energy resulting from the gas expansion. This equipment, using helico axial hydraulic cells, is efficient for all gas volume fractions. The two phase efficacy is slightly dependant on the gas volume fraction and the relative density of the liquid and gas phases but with a limited effect. Concerning the operation of two phase flow turbines see: https://yvcharron.com/index.php/two-phase-flow-turbines/

The regeneration process of a physical solvent using two phase flow turbines is detailed in the present web site: https://yvcharron.com/index.php/gas-treatment/.

It should be recalled that the energy required to pressurise a gas is considerably greater than the pressurisation of a liquid (based on same mass flow rates and same initial and final pressures). The same comparison applies for a depressurisation process. As a consequence, **the energy provided by the depressurisation of the two-phase solvent liquid – sour gas mixture is greater than the energy required for the pressurisation of the purified liquid solvent.**

6_Carbon dioxide inventory

In 2019, the total world emission of CO2 per year was 34 169 MTons according to Wikipedia: https://fr.wikipedia.org/wiki/%C3%89mission_de_dioxyde_de_carbone

In 2017, the total world electricity production was 25 000 TWh according to CEA Energy Handbook https://www.cea.fr/multimedia/Pages/editions/ouvrages/memento-sur-energie.aspx. Among that production, energy distribution was the following:

SECTOR	Coal	Liquid fuel-Oil	Gas fuel	Nuclear	Hydro	Solar Wind Tide	Waste biofuel
Per cent	38.3	3.7	23.1	10.4	16.6	5.6	2.3

CO_2 emission produced by solid and liquid fuels per electrical production unit according to EIA is as follows: https://www.eia.gov/tools/faqs/faq.php?id=74&t=11

Carbonated fuels	Coal	Petroleum	Natural gas
CO2 emission Kg/kWh	1.003 also 8.79 kT/MW/year	0.966 also 8.46 kT/MW/year	0.417 also 3.65 kT/MW/year

In this study, it is assumed that CO_2 emission is 4.0 kT/MW/year which may be considered a minimum when compared to the above values.

Limited to carbonated fuels, world electrical production and CO_2 emission per year are:

SECTOR	Coal	Liquid fuel	Gas fuel	TOTAL
Energy TWh	9 575	925	5 775	16 275
CO2 GTons	4.43	0.43	2.67	7 53
CO2 emission Per cent	12.97	1.25	7.82	22.0

Figure 6 – World electrical production and CO2 emissions for three carbon fuel types.
Note: Above, k, M, G and T stand for 10 with exponent, respectively, 3, 6, 9 and 12.

In recent years, **7.53 GTons per year of CO2 were emitted by conventional thermal electrical equipment burning coal, liquid and gas fossil fuels** that is 22 per cent of the total world emission (figure 6). Coal represents the largest emission followed by natural gas then liquid fuels.

Most electrical generators are driven by steam turbines with steam supplied by boilers burning mostly coal and to a lower extend liquid and gas fuels. In that instance, CO_2 removal can only be undertaken by using a treatment process located downstream the boiler (External treatment).

Some electrical generators are driven by gas turbines burning liquid or gas fuels. As described in above sections, depending on several parameters (pressure and temperature at the GT compressor exit and also the gas temperature at the GT exhaust), CO_2 removal may be undertaken at turbine intermediate position or at the turbine exhaust.

Nevertheless, **whatever boilers or gas turbines are considered and whatever External or Intermediate treatment systems are used, they offer the possibility to absorb a large fraction of the CO2 produced**.

7_Comparison of external and intermediate systems

PROCESS SYSTEM	Intermediate removal	External removal
Cmp-Exp loop bottom pressure	Main air compressor exit	Atmospheric
Cmp-Exp loop top pressure	Depends on main air compressor pressure ratio	Depends on gas turbine thermal efficiency
Cmp-Exp loop complexity	Only one “CpR-XpR” stage – Little complexity	Three “CpR-XpR” stages – Large complexity
Comp-Exp loop common to several gas turbines	**Not possible** -Limited CpR-XpR polytropic efficiency (small flow)	**Possible** - Large CpR-XpR polytropic efficiency (large flow rate)
Impact on existing installations	GT revamping - Impact on GT control system to be evaluated	No impact on GT package - Little impact on GT control system
Liquid solvent entrainment	No consequences – Burnt into combustion chamber	Rejection into atmosphere of finest droplets
Impact on GT perf.	Small degradation	Small degradation
CO2 and NOx removal	Partial – Approx. 60 %	Total – No atm. rejection

Table 7.1 – Comparison between GT with intermediate and external carbon dioxide removal.

Above is a comparison between the two systems with their relative advantages (blue) and disadvantages (red).

The main advantage of CO2 removal at GT intermediate stage is the lesser complexity of the compression-expansion loop (high bottom pressure) requiring only one compression - expansion stage. It should be noted that for a given contactor pressure, the energy loss in the compression-expansion loop reduces when the main compressor pressure ratio increases (blue parameters on figure 7.2 – See also figure 2.3) resulting at the same time by an increase in the air temperature (more energy available).

The main disadvantage of this arrangement is the significant rejection of CO2 in the atmosphere. The amount of CO2 rejected into the atmosphere reduces when the GT inlet air dilution increases (Figure 2.1). This arrangement requires also a modification of the gas turbine internals (between air compressor and combustion chamber).

The main advantage of a CO2 removal system downstream a gas turbine is the total removal of CO2.

The main disadvantage of this arrangement is a larger complexity of the compression-expansion loop (bottom pressure is atmospheric) requiring a large number (usually three) of compression and expansion stages. The compression requirement reduces when the exhaust temperature increases (green parameters on figure 7.2).

In both cases, the compression – expansion loop operates better when the CpR and XpR efficiencies are high. This is obtained mostly when the flow rate is large (red parameters on figure 7.2).

	Supplied power MW	Thermal efficiency per cent	**Air comp. Pressure ratio**	Mass flow out kg/s	**Vol flow km3/hr**	**Exhaust temper. °C (1)**	Heat rate kJ/kWh
T21	10.4	34.8	16.0	33.8	94	508	10 342
T22	24.5	33.6	14.0	81.3	225	543	10 720
T23	39.8	40.3	**24.3**	115	318	468	8 922
T24	50.0	39.4	**19.8**	125	346	**560**	9 147
T25	187	36.5	12.8	558	**1 545**	536	9 863
T26	329	41.0	**20.1**	724	**2 004**	**599**	8 780
T27	593	42.8	**24.1**	1050	**2 907**	**670**	8 411

Figure 7.2 – Characteristics of GT with different frame size (same manufacturer). Blue and green colours highlight GT which may be elected, respectively, for intermediate and external CO2 removal. Red colour identifies GT with large air flow rate (High CpR & XpR efficiencies).

8_Conclusion

Intermediate and external CO2 removal systems based on a physical solvent require a compression – expansion loop to raise the pressure of burnt gases and to permit the solvent contactor to operate at a relatively high pressure.

Complexity of the Compression – Expansion Loop (CEL)

CO2 removal in a GT intermediate system requires a relatively simple arrangement of the CEL (high bottom pressure). This complexity reduces as the pressure at the main air compressor exit increases requiring less pressure rise in the CEL and causing less energy losses. To the contrary, CO2 removal at GT exhaust requires a relatively complex arrangement (atmospheric bottom pressure): three sections of compression and expansion. In both cases, a high expansion temperature (CEL) facilitates the pressure rise.

Fraction of CO2 removal

CO_2 removal in a GT intermediate system is limited to 60 % (complement rejected to atmosphere) for a GT air dilution factor of three. This percentage increases with the air dilution factor. To the contrary, CO_2 removal at GT exhaust is total.

Efficiency of the CEL

Polytropic efficiency of the CEL is dependent mostly on the volume flow of the gas to be treated. The highest this parameter, the highest the Reynolds number, the lowest the friction losses and the highest the compression and expansion efficiencies.

Thermal efficiency of the overall system

This efficiency is mostly dependant on the gap between the combustion chamber pressure and the contactor pressure. The smaller the gap, the lower is the energy degradation. The losses in the CEL loop are of three types: pressure and heat losses and residual heat in one heat exchanger. Energy may be recovered in three directions: residual heat transferred to a motor cycle or to a solvent boiler (if any), depressurisation of the liquid solvent – sour gas mixture and energy excess in the CEL loop. **As an overall, the GT thermal efficiency is relatively unchanged when the air compressor pressure ratio is high.**

Gas turbines elected to CO_2 removal in a CEL using a physical solvent

Two cases may be considered. Case a) Gas turbines presenting a relatively large air compressor pressure ratio (therefore exit temperature) facilitating CO_2 removal at intermediate position. The higher this parameter; the lower is the impact on the GT thermal efficiency. **It should be preferably greater than 16/1**. Case b) Gas turbines presenting a relatively large exhaust temperature (low GT thermal efficiency) facilitating CO_2 removal at GT exhaust. **It should be preferably above 550 °C.**

Combined CO_2 removal in a multi turbine power plant

Combined CO_2 removal based on an intermediate extraction system is not possible for several gas turbines due to the lack of individual control (combustion chamber temperature, running speed, flow ratio). To the contrary, combined CO_2 removal based on an exhaust system is possible for several gas turbines (and boilers) as the turbine control systems are not active at this location. It should be added that the larger is the flow rate, the larger is the efficiency of the CEL.

Physical solvent regeneration

Energy may be recovered during the two phase expansion of the solvent liquid – sour gas mixture by using a two-phase flow turbine (helico-axial design). It is greater than the energy required to pressurise the purified solvent.

Carbon dioxide emission

In 2019, total world emission of CO_2 per year was of the order of 34 GTons. Electrical generation using carbonated fuels (coal, liquid and gas fuels activating boilers and gas turbines) would contribute to 22% of that emission. Most of these electrical generation emissions could be capted in contactors using physical solvents.

Annexe 1 – Combustion chamber pressure parameter

This annexe has been added in April 2021.

In section 5.2, the system performance has been analysed in the worst case of air compressor pressure ratio (16/1) corresponding to a pressure of 16 Bar abs (a relatively low pressure) at the combustion chamber. In this annexe, an analysis is performed based on a wider range of combustion chamber pressure (16 to 32 Bar abs) keeping constant the contactor pressure. In section 5.4, the incidence of the pressure at the combustion chamber was analysed in a different situation, particularly, the capability of the CpR – XpR loop to reach a high pressure level (variable contactor pressure).

In this annexe, calculations have been performed on the following basis:

- The contactor pressure is constant and equal to 80 Bar abs.
- One mole of di-oxygen per second (32 g/s) corresponding to a GT inlet flow of 411 g/s.
- Pressure losses in the compression – expansion loop are assumed proportional to the pressure difference between the contactor pressure and the air compressor exit pressure that is 3 % pressure losses for a pressure difference of 80 minus 16 Bars that is 64 Bar.
- Temperature reduction (due to heat losses) in the compression – expansion loop is equal to 30 °C.
- The temperature at the expander (XpR) inlet is equal to 450 °C.

As shown in section 5.5.2, when the combustion chamber pressure is equal to 16 Bar abs the thermal efficiency is reduced from 35.6 % (No CO2 removal) to 25.3 % (10 points less). From figure A.1, when that pressure is equal to 32 Bar abs, the thermal efficiency is only reduced from 39.6 % to 32.7 % (7 points less). These calculations do not take into account energy recovery.

It should be noted that when the combustion chamber pressure increases, the thermal efficiency increases due to a greater expansion in the hot section of the gas turbine. Note that these calculations were performed at a constant fuel gas flow rate.

As shown in section 5.5.3, three types of energy may be recovered: a) Residual heat from heat exchanger HC1; b) Carbon dioxide pressurisation; c) Energy excess in CpR – XpR loop.

When the combustion chamber pressure is equal to 16 Bar abs, the thermal efficiency is increased from 25.3 % (figure A.2) to 26.5 % by considering two types of energy recovery (carbon dioxide pressurisation and CpR – XpR energy excess). It would be increased to 27.5 % by supplying a relatively small fraction of the HC1 residual to the solvent boiler. This last figure is approximately **8 points below the "No CO2 removal" thermal efficiency**.

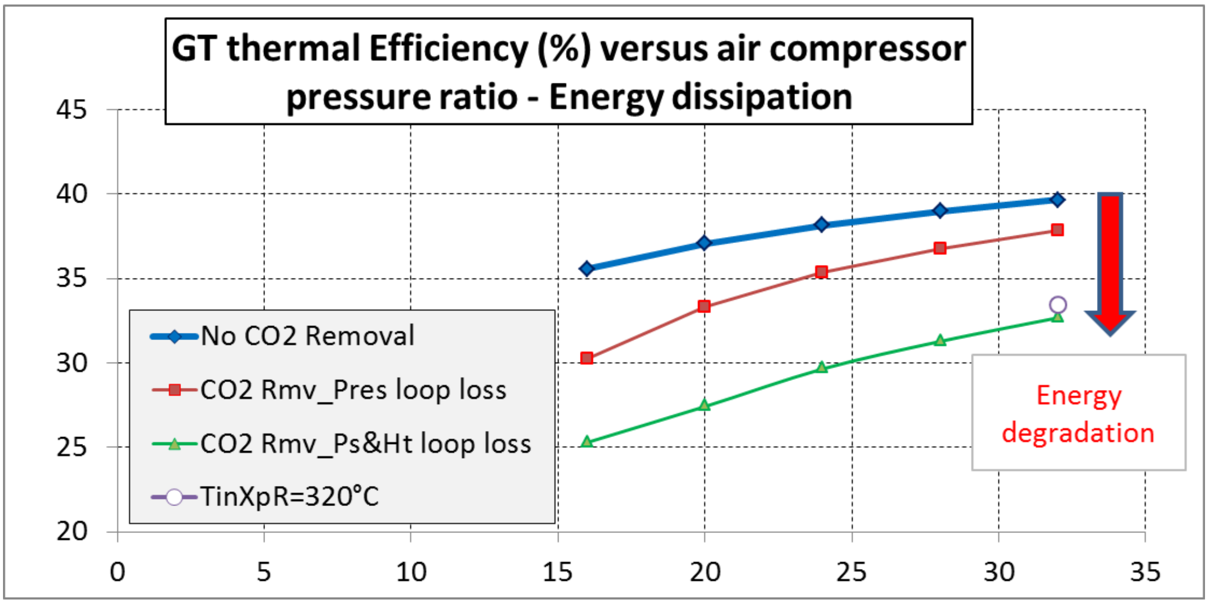

Figure A.1 – Gas turbine thermal efficiency versus air compressor pressure ratio for three cases: a) No CO2 removal; b) CO2 removal integrating pressure losses in CpR – Xpr loop; c) CO2 removal integrating pressure and heat losses in CpR – XpR loop. All calculations made for XpR inlet temperature at 450 °C. Only one calculation made at 320 °C.

When the combustion chamber pressure is equal to 32 Bar abs the thermal efficiency is raised from 32.7 % to 36.3 % by integrating the above two mechanical energy recoveries. It is then increased to 37.3 % by integrating heat recovery from heat exchanger HC1. This last figure is only **2 points below the "No CO2 removal" thermal efficiency.**

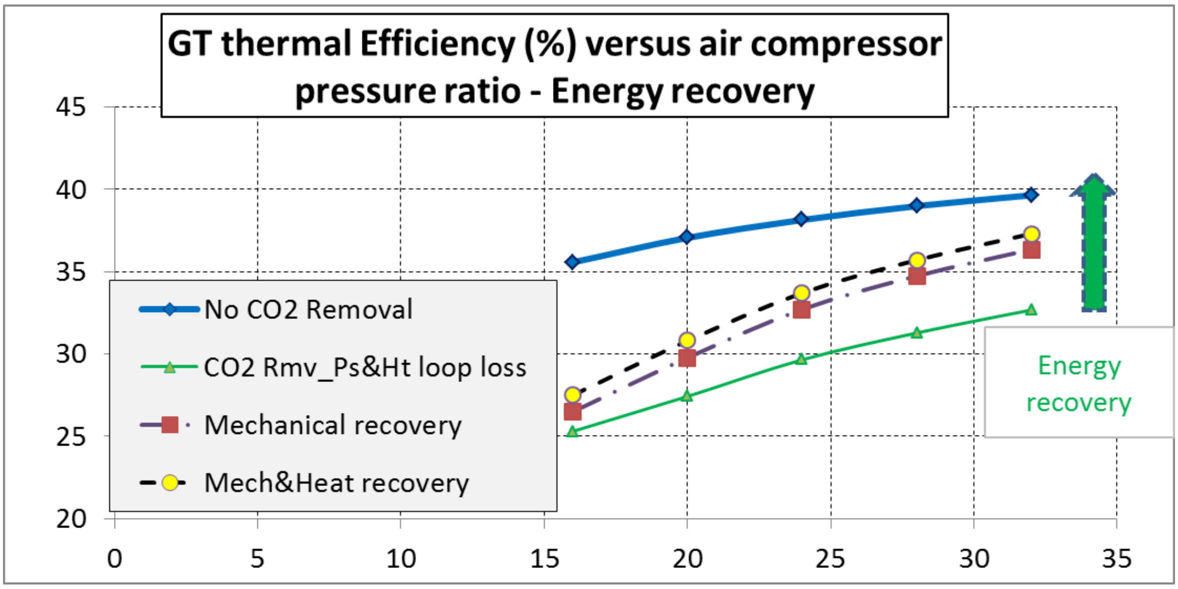

Figure A.2 – Gas turbine thermal efficiency versus air compressor pressure ratio for four cases: a) No CO2 removal; b) CO2 removal integrating pressure and heat losses in CpR – XpR loop; c) Mechanical energy recovery; d) Mechanical and heat energy recovery.

Annexe 2 – Artificial GT inlet air dilution increase

This annexe has been added in April 2021.

Adding a fraction of oxygen to the atmospheric air flow reduces the relative fraction of nitrogen at the gas turbine inlet. This reduction in nitrogen molecules is compensated by an increase in carbon dioxide molecules (increased flow recycling). This permits to increase the fraction of carbon dioxide flowing through the compression – expansion loop and, therefore, the degree of absorption of that gas.

In that situation, less carbon dioxide is emitted into the atmosphere.