OXY-COMBUSTION TURBINE POWER PLANTS

Report: 2015/05

August 2015

INTERNATIONAL ENERGY AGENCY

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an international energy programme. The IEA fosters co-operation amongst its 28 member countries and the European Commission, and with the other countries, in order to increase energy security by improved efficiency of energy use, development of alternative energy sources and research, development and demonstration on matters of energy supply and use. This is achieved through a series of collaborative activities, organised under more than 40 Implementing Agreements. These agreements cover more than 200 individual items of research, development and demonstration. IEAGHG is one of these Implementing Agreements.

DISCLAIMER

This report was prepared as an account of the work sponsored by IEAGHG. The views and opinions of the authors expressed herein do not necessarily reflect those of the IEAGHG, its members, the International Energy Agency, the organisations listed below, nor any employee or persons acting on behalf of any of them. In addition, none of these make any warranty, express or implied, assumes any liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product of process disclosed or represents that its use would not infringe privately owned rights, including any parties intellectual property rights. Reference herein to any commercial product, process, service or trade name, trade mark or manufacturer does not necessarily constitute or imply any endorsement, recommendation or any favouring of such products.

COPYRIGHT

ACKNOWLEDGEMENTS AND CITATIONS

This report describes research sponsored by IEAGHG. This report was prepared by:

Amec Foster Wheeler
Politecnico di Milano

The principal researchers were:

- Luca Mancuso (Amec Foster Wheeler)
- Noemi Ferrari (Amec Foster Wheeler)
- Paolo Chiesa (Politecnico di Milano)
- Emanuele Martelli (Politecnico di Milano)
- Matteo Romano (Politecnico di Milano)

Additional support provided by NET Power and CES is gratefully acknowledged.

To ensure the quality and technical integrity of the research undertaken by IEAGHG each study is managed by an appointed IEAGHG manager. The report is also reviewed by a panel of independent technical experts before its release.

The IEAGHG manager for this report was:

John Davison

The expert reviewers for this report were:

- George Booras – EPRI
- Scott Hume - EPRI
- Olav Bolland – NTNU
- Lars Olof Nord – NTNU
- Rohan Dutta – NTNU
- Wolfgang Sanz - Graz University of Technology

The report should be cited in literature as follows:

'IEAGHG, "Oxy-Combustion Turbine Power Plants, 2015/05, August, 2015.'

Further information or copies of the report can be obtained by contacting IEAGHG at:

IEAGHG, Pure Offices, Cheltenham Office Park,
Hatherley Lane, Cheltenham,
Glos., GL51 6SH, UK
Tel: +44(0) 1242 802911
E-mail: mail@ieaghg.org
Internet: www.ieaghg.org

OXY-COMBUSTION TURBINE POWER PLANTS

Key Messages

- The predicted thermal efficiencies of the oxy-combustion turbine power cycles assessed in this study range from 55% (LHV basis) for the NET Power cycle to around 49% for the other base case cycles. For comparison, a recent IEAGHG study predicted an efficiency of 52% for a natural gas combined cycle plant with post combustion capture using a proprietary solvent.

- There is scope for improving the thermal efficiencies in future, for example by making use of materials capable of withstanding higher temperatures. Proprietary improvements by process developers may also result in higher efficiencies.

- The levelised cost of electricity (LCOE) of base-load plants using natural gas at 8 €/GJ are estimated to be 84-95 €/MWh, including CO_2 transport and storage costs. The lowest cost oxy-combustion plant (NET Power) has a slightly lower LCOE than a conventional gas turbine combined cycle with post combustion capture using a proprietary solvent.

- The cost of CO_2 emission avoidance of the various cycles compared to a reference conventional natural gas combined cycle plant is 68-106 €/t CO_2 avoided.

- The base case percentage capture of CO_2 in this study was set at 90% but it was determined that it could be increased to 98% without increasing the cost per tonne of CO_2 avoided, or essentially 100% if lower purity CO_2 was acceptable.

- The water formed by combustion is condensed in oxy-combustion turbine cycles, which would mean that if air cooling was used the power plants could be net producers of water. This could be an advantage in places where water is scarce, although air cooling would reduce the thermal efficiency.

- Oxy-combustion cycles could have advantages at compact sites. The total area of an oxy-combustion combined cycle plant is estimated to be slightly less than that of a conventional combined cycle with post combustion capture. The ASU could be located off-site if required to further reduce the power plant area. In addition, regenerative oxy-combustion cycles are significantly more compact than combined cycles.

- Oxy-combustion turbines could be combined with coal gasification. The predicted thermal efficiency of a coal gasification plant with a semi-closed oxy-combustion combined cycle (SCOC-CC) is 34% (LHV basis). This is similar to that of more conventional CCS technologies (IGCC with pre-combustion capture and supercritical pulverised coal with post combustion amine scrubbing) but the estimated capital cost and cost of electricity of the oxy-combustion turbine plant are significantly higher.

Background to the Study

Post combustion capture is usually considered to be the leading option for capture of CO_2 at natural gas fired power plants but there is increasing interest in the alternative of oxy-combustion turbines which use recycled CO_2 and/or H_2O as the working fluid instead of air. Large component tests have taken place and a 50 MW_{th} demonstration plant is scheduled to be commissioned in 2017. Oxy-combustion turbines can also be combined with solid fuel gasification as an alternative to IGCC with pre-combustion capture.

This study provides an independent evaluation of the performance and costs of a range of oxy-combustion turbine cycles, mainly for utility scale power generation. The study was carried out by Amec Foster Wheeler in collaboration with Politecnico di Milano.

Scope of Work

The study includes the following:

- A literature review of the most relevant systems featuring oxy-combustion turbine cycles, discussing the state of development of each of the cycles and their components.
- Detailed modelling of the gas turbine for the most promising cycles, including efficiency, stage number and blade cooling requirements. This modelling was carried out using calculation codes developed by Politecnico di Milano for performance prediction of gas turbines.
- Technical and economic modelling of complete oxy-combustion turbine power plants, including sensitivity analyses for a range of technical design and financial parameters.
- Assessment of potential future improvement, including high temperature turbine materials.
- High level evaluation of the most promising niche market applications for oxy-combustion turbines, particularly in smaller power plants.
- An assessment of oxy-combustion turbines combined with coal gasification.

The study has been undertaken in consultation with technology developers but to avoid any possibility of restrictions on dissemination of the results no confidential information has been used.

Study Basis

Technical and economic basis

The technical and economic conditions for power plants will depend on many site specific factors. IEA GHG has used, as far as possible, a standard technical and economic basis in its studies of CCS plants to ensure comparability but it is recognised that the absolute values of efficiency, costs etc. will depend on local conditions and assumptions. The technical and economic basis for the study is described in detail in the main study report and the main base case assumptions are as follows:

- Greenfield site, North East coast of the Netherlands
- 9C ambient temperature
- Natural draught cooling towers
- European pipeline natural gas: 46.5 MJ/kg (LHV), 3% total inerts
- CO_2 to storage: 11MPa, 100ppm O_2, 50ppm H_2O
- 2Q 2014 costs
- Natural gas price: 8 €/GJ LHV basis (equivalent to 7.23 €/GJ HHV basis)
- Coal price: 2.5 €/GJ LHV basis (equivalent to 2.39 €/GJ HHV basis)
- Discount rate: 8% (constant money values)
- Operating life: 25 years
- Construction time – Natural gas fired plants: 3 years
  – Coal gasification plants: 4 years
- Capacity factor – Natural gas fired plants: 90% (65% and 85% in years 1 and 2)
  – Coal gasification plants: 85% (60% and 80% in year 1 and 2)
- CO_2 transport and storage cost: 10 €/t stored

The degree of CO_2 capture was set at approximately 90% but there were slight variations between cases. Sensitivities to higher percentage capture and CO_2 purity levels were also assessed.

The reference power plant without CO_2 capture in this study was based on two state-of-the-art 50Hz F class gas turbines, resulting in a net power output of 904 MW. The oxy-combustion turbine plants were designed to have the same fuel feed rates as the reference plant without capture, resulting in net power outputs in the range of 660-850 MW.

Cost definitions

Capital cost

The cost estimates were derived in general accordance with the White Paper "Toward a common method of cost estimation for CO_2 capture and storage at fossil fuel power plants", produced collaboratively by authors from IEAGHG, EPRI, USDOE/NETL, Carnegie Mellon University, IEA, the Global CCS Institute and Vattenfall[1].

The capital cost is presented as the Total Plant Cost (TPC) and the Total Capital Requirement (TCR). TPC is defined as the installed cost of the plant, including project contingency. In the report TPC is broken down into:

- Direct materials
- Construction
- EPC services
- Other costs
- Contingency

[1] Toward a common method of cost estimation for CO_2 capture and storage at fossil fuel power plants, IEAGHG Technical Review 2013/TR2, March 2013.

TCR is defined as the sum of:

- Total plant cost (TPC)
- Interest during construction
- Owner's costs
- Spare parts cost
- Working capital
- Start-up costs

For each of the cases the TPC has been determined through a combination of licensor/vendor quotes, the use of Foster Wheeler's in-house database and the development of conceptual estimating models, based on the specific characteristics, materials and design conditions of each item of equipment in the plant. The other components of the TCR have been estimated mainly as percentages of other cost estimates in the plant.

Estimation of costs of technologies that are at relatively low technology readiness levels (TRL) is inevitably subject to significant uncertainty and involves a balance of judgement which is provided in this study by an experienced process engineering contractor. The oxy-combustion turbine power plants include novel equipment that are either under development or at a conceptual stage only and overall integrated plants have not yet been operated. This study has investigated the potential of the oxy-combustion turbine plants with respect to benchmark technologies for capture of CO_2 that are generally assumed to be ready for commercial application. The study has therefore treated the oxy-combustion turbine plants as Nth-of-a-kind (NOAK) plants for estimating purposes. The cost of novel equipment has been evaluated as already developed and suitable for large-scale commercial application and no additional contingencies have been applied. Nevertheless, the sensitivity to the cost of the novel equipment has been assessed to take into account to a certain extent the intrinsic uncertainty of such estimates.

It is recognised that by the time the oxy-combustion turbine processes actually reach NOAK plant status there will have been further improvements in their technology but the technology of air blown gas turbines is also expected to continue to improve over this time scale. Prediction of future improvements in the technologies of air and oxy-combustion turbines would have involve excessive speculation and has therefore been avoided in this study.

Levelised cost of electricity

Levelised Cost of Electricity (LCOE) is defined as the price of electricity which enables the present value from all sales of electricity over the economic lifetime of the plant to equal the present value of all costs of building, maintaining and operating the plant over its lifetime. LCOE in this study was calculated assuming constant (in real terms) prices for fuel and other costs and constant operating capacity factors throughout the plant lifetime, apart from lower capacity factors in the first two years of operation. It should be noted that base-load LCOE is not a complete measure of the relative economic merits of different types of power plant. The ability to operate flexibly and to provide ancillary services to the electricity system can provide significant additional income but this is beyond the scope of this study.

Cost of CO_2 avoidance

Costs of CO_2 avoidance were calculated by comparing the CO_2 emissions per kWh and the levelised costs of electricity of plants with capture and a reference plant without capture.

$$CO_2 \text{ avoidance cost (CAC)} = \frac{LCOE_{ccs} - LCOE_{Reference}}{CO_2\ Emission_{Reference} - CO_2\ Emission_{ccs}}$$

Where:
CAC is expressed in Euro per tonne of CO_2
LCOE is expressed in Euro per MWh
CO_2 emission is expressed in tonnes of CO_2 per MWh

For calculation of the CO_2 avoidance cost, the reference plants in this study were assumed to be plants without CCS that use the same type of fuel as the oxy-combustion turbine plants and the type of technology that is most likely to be preferred for that type of fuel. Hence the reference plant for natural gas fired oxy-combustion turbine plants is a conventional natural gas combined cycle (NGCC) plant and the reference plant for the coal gasification oxy-combustion turbine plant is a supercritical pulverised coal (SC-PC) plant without CCS. It is assumed that for coal-fired plants without CO_2 capture a SC-PC plant would be preferred to an IGCC plant because of lower expected costs.

Findings of the Study

Literature review and cycle selection

A literature review was carried out to identify the leading natural gas fired oxy-combustion turbine cycles. The cycles were ranked on the basis of their expected efficiencies and the technological development still required for their key components. The following four cycles were selected for more detailed technical and economic assessment:

- Semi-closed oxy-combustion combined cycle (SCOC-CC)
- NET power cycle
- S-Graz cycle
- CES cycle

In the first two of these cycles the working fluid is mainly CO_2 and in the final two it is mainly H_2O.

Semi-closed oxy-combustion combined cycle (SCOC-CC)
The SCOC-CC resembles a conventional combined cycle, except that inlet gas to the turbine compressor is mainly recycled CO_2–rich gas rather than air from the atmosphere, and oxygen from an air separation unit (ASU) is fed to the combustor. The exhaust gas from the turbine passes through a heat recovery steam generator (HRSG) which generates steam for a steam cycle, as in a conventional combined cycle. Instead of being vented to the atmosphere the cooled gas from the HRSG is further cooled, which condenses most of the water produced by combustion. Most of the cooled CO_2-rich gas is then recycled to the gas turbine compressor and the rest is compressed and purified for CO_2 storage. The SCOC-CC has been proposed and studied by various organisations. Although it resembles a conventional combined cycle it would not be possible to retrofit a conventional gas turbine as an oxy-combustion turbine because of the substantially different physical properties of CO_2 compared to air.

NET power

The NET Power cycle utilises CO_2 as the working fluid in a high pressure, low pressure ratio recuperated Brayton cycle with an inlet pressure of around 300 bar and a pressure ratio of around 8-12. Oxygen and natural gas are fed to the high pressure combustor. A recuperative heat exchanger is used to transfer heat from the turbine exhaust gas to high pressure CO_2-rich recycle gas. Recompression of the recycle gas takes place firstly in a gas compressor and finally in a liquid pump. NET Power and 8 Rivers Capital, together with Toshiba, CB&I and Exelon are developing this cycle and a $50MW_{th}$ plant in Texas is scheduled to start commissioning in 2016.

Modified S-Graz cycle

Different variants of the S-Graz cycle have been proposed and two have been evaluated in this study. The working fluid in the Modified S-Graz cycle is mainly steam, along with some recycled CO_2. The cycle includes a high temperature gas turbine with an inlet pressure of about 45 bar followed by an HRSG. High pressure steam generated in the HRSG is fed to a steam turbine which exhausts into the gas turbine combustor and turbine. Some of the outlet gas from the HRSG is recompressed and sent to the turbine combustor and the remainder is compressed and cooled, thereby condensing the steam it contains, leaving a CO_2-rich gas to be fed to storage. Heat from cooling and condensation of the compressed gas is used to generate low pressure steam which is fed to a separate steam turbine. The S-Graz cycle is being researched primarily at the University of Graz, Austria.

CES cycle

The cycles being developed by Clean Energy Systems (CES) use water, both in vapour and liquid phases, as the combustor temperature moderator. Different versions of the CES cycle have been proposed and three versions representing near to longer term variants were evaluated in this study. The results presented in this overview are for a new supercritical pressure cycle as proposed by CES during this study. The cycle includes a high pressure oxy-fuel combustor where part of the fuel and oxygen are combusted using steam in supercritical conditions as the temperature moderator, while hot gas produced in the gas generator is expanded in a steam cooled HP turbine. The high pressure turbine exhaust gas is double reheated by supplementary oxy-fuel combustion and further expanded in medium and low pressure sections of the gas turbine, down to vacuum conditions. CES has been working with partners on development of pressurised oxy-combustion power systems for more than 15 years. The primary focus of this development work has been on the new or novel components of the cycle, including high pressure combustors, reheaters and turbines. Oxy-fuel gas generators of up to 170 MW_{th} and reheat combustors up to 28 MW_{th} have been operated at a test facility in California and hot gas has been fed to modified gas turbines.

Performance of natural gas-fired oxy-combustion power plants

A summary of the performance of the base case natural gas-fired oxy-combustion turbine power plants and the reference plant is given in Table 1. The plants all have the same natural gas feed rate of 1536 MW (LHV).

The highest efficiency of 55% is for the NET Power cycle, the other three oxy-combustion processes have lower efficiencies of around 49%. The developers of the NET Power cycle have estimated an efficiency of 59% for their cycle using proprietary improvements and CES has estimated an efficiency of 53% for its cycle. The supercritical version of the CES cycle is a relatively recent innovation. Adopting a lower coolant temperature would be likely more advantageous and is currently being pursued by CES as part of their on-going cycle optimization work.

The power consumption of the air separation unit (ASU) including oxygen compression is substantial in all of the oxy-turbine cycles, being equivalent to 10-11% points of overall thermal efficiency.

Table 1 Performance of natural gas-fired power plants

	Net power output	CO_2 captured	CO_2 emissions	Efficiency		Efficiency penalty for capture (LHV)
				LHV	HHV	
	MW	kg/MWh	kg/MWh	%	%	% points
Reference NGCC plant	904	-	348	58.8	53.2	
SCOC-CC	757	377	39	49.3	44.6	9.5
NET Power	846	336	37	55.1	49.9	3.7
Modified S-Graz	756	375	41	49.2	44.6	9.6
CES	751	379	41	48.9	44.3	9.9

Costs of natural gas fired oxy-combustion power plants

Capital costs, levelised costs of electricity and costs of CO_2 avoidance are summarised in Table 2. The NET Power process has the lowest costs while the costs of the three other processes are broadly similar. Breakdowns of the Total Plant Costs are shown in Figure 1.

Table 2 Costs of natural gas fired plants

	Total Plant Cost (TPC)		Total Capital Requirement (TCR)	Levelised Cost of Electricity (LCOE)		CO_2 Avoidance Cost
	€/kW	% increase for capture		€/MWh	% increase for capture	€/tonne
Reference NGCC	655	-	855	62.5	-	-
SCOC-CC	1470	124	1905	92.8	48	98
NET Power	1320	102	1715	83.6	34	68
Modified S-Graz	1500	129	1955	93.7	50	101
CES	1540	135	2000	95.1	52	106

It can be seen from Figure 1 that the main reason for the higher capital costs of the oxy-combustion turbine plants compared to the reference plant is the cost of the ASU (including oxygen compression), followed by the CO_2 compression and purification unit (CPU) and the balance of plant items.

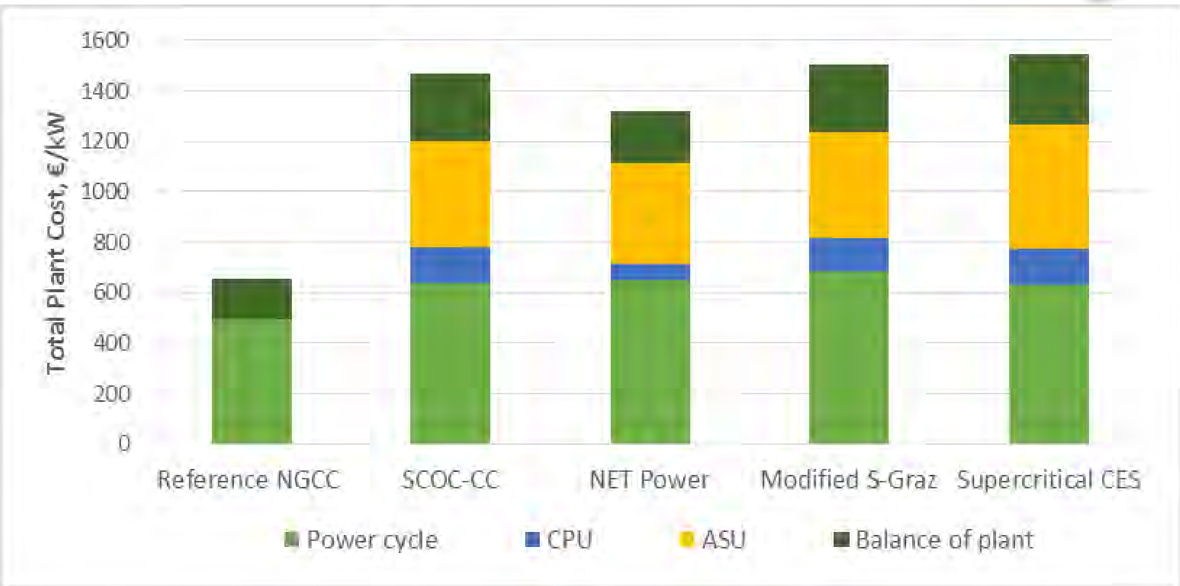


Figure 1 Specific Total Plant Costs – Natural Gas Fired Plants (2Q2014)

A breakdown of the levelised costs of electricity (LCOE) is shown in Figure 2. The main contribution to the LCOE in all cases is the fuel cost, which depends on the thermal efficiency, but the main contribution to the additional cost of capture is the additional capital cost.

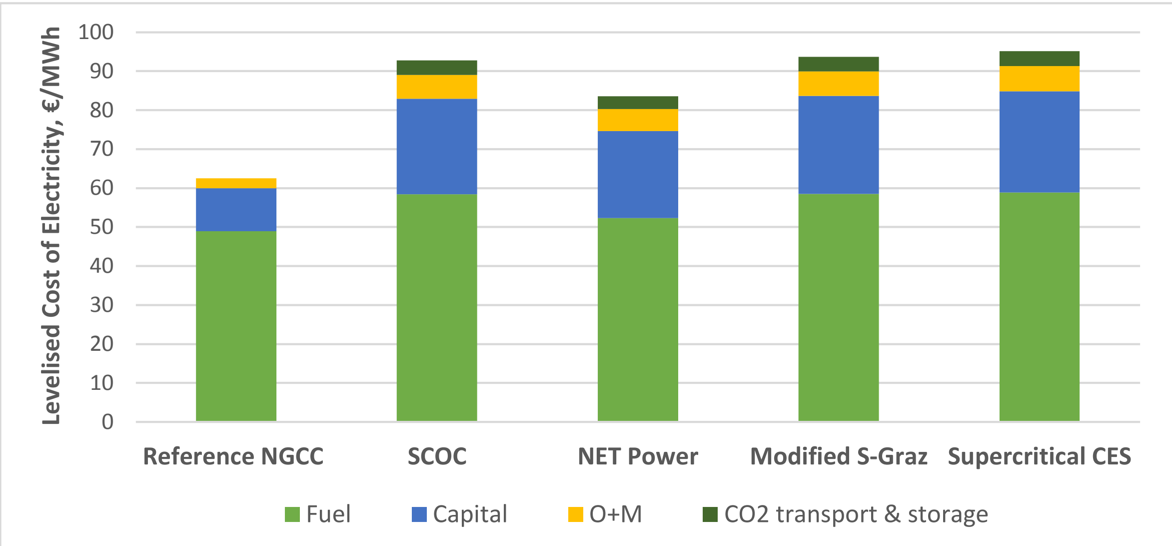


Figure 2 Levelised Costs of Electricity

Comparison of oxy-combustion and post combustion capture plants

In 2012 IEAGHG published a study on natural gas combined cycle plants with post combustion capture[2]. The study assessed plants using MEA scrubbing with and without gas turbine flue gas recycle and a plant using a proprietary capture solvent. A comparison of the efficiency and costs of the most favourable case evaluated in that study, i.e. the proprietary

[2] CO_2 capture at gas fired power plants, IEAGHG report 2012/8, July 2012

solvent case, and the two oxy-combustion cases from this study with the highest efficiencies and lowest costs (SCOC-CC and NET Power) are shown in Table 3.

Table 3 Comparison of oxy-combustion and NGCC without capture

	Efficiency (LHV)	Total Plant Cost	Levelised Cost of Electricity	CO_2 Avoidance Cost
	%	€/kW	€/MWh	€/tonne
Reference NGCC	58.8	655	62.5	-
SCOC-CC	49.3	1470	92.8	98
NET Power	55.1	1320	83.6	68
NGCC post combustion capture	52.0	1170	84.7	72

The SCOC-CC plant has a lower thermal efficiency than the plant with post combustion capture but the efficiency of the NET Power case is significantly higher. The Total Plant Costs of the oxy-combustion plants are higher than that of the post combustion capture plant.

The NET Power plant has a lower LCOE and cost of CO_2 avoidance than the post combustion capture plant because the higher efficiency and hence lower fuel cost more than offsets the higher capital cost. In contrast the LCOE and CAC of the SCOC-CC (and the other oxy-combustion plants evaluated in this study) are higher than those of the post combustion capture plant.

The capital costs in IEAGHG's post combustion capture study were on a 2011 basis but there is reported to have been no significant change in the costs of European power plants in general between 2011 and 2014 (i.e. the cost estimation date of this oxy-combustion study)[3]. This oxy-combustion study used some updated assumptions compared to the post combustion capture study, particularly regarding natural gas price and CO_2 transport and storage costs. The LCOE and CAC for post combustion capture shown in Table 3 were recalculated from the 2011 study using the updated assumptions.

The plants in this study were based on natural draught cooling towers, in common with IEAGHG's recent study on coal fired plants[4] but the earlier study on natural gas post combustion plants was based on mechanical draught cooling towers. As described later, a NET Power plant using mechanical draught cooling towers was assessed as a sensitivity case. The TPC of that plant was 1245 €//kW, i.e. closer to that of the post combustion capture plant and the LCOE was slightly lower at 82 €/MWh. The efficiency of the NET Power process estimated in this study was lower than NET Power's own estimate of 58.8%. Increasing the efficiency to 58.8% would reduce the LCOE by 3 €/MWh due to lower fuel costs, or 5 €/MWh if there was also a corresponding reduction in the capital cost per kW. NET Power provided their own commentary on the differences between their efficiency and cost estimates and those presented in this study and this is included in the detailed study report.

[3] IHS European Power Capital Cost Index (EPCCI), excluding nuclear, https://www.ihs.com/Info/cera/ihsindexes/index.html

[4] CO_2 capture at coal-based power and hydrogen plants, IEAGHG report 2014-3, May 2014.

Sensitivity to technical parameters

Sensitivities to the following technical parameters were assessed. The sensitivities were only evaluated for selected cycles to avoid an excessive number of cases.

- Turbine combustor outlet temperature
- Turbine maximum metal temperature
- CO_2 purity requirements
- Percentage capture of CO_2
- Oxygen purity in the range of 95-99.5%
- Natural gas with a high CO_2 concentration: 70% (vol.)
- Natural gas with a high N_2 concentration: 14% (vol.)
- Higher ambient temperature: 25C
- Alternative cooling system: mechanical draught cooling towers

Potential future improvements: turbine temperatures

The thermal efficiencies of conventional NGCCs are expected to improve in future due to various technological improvements, in particular higher firing temperatures which increase the thermodynamic cycle efficiency, and higher allowable material temperatures which reduce the turbine cooling gas requirement. It is important to assess whether the oxy-combustion turbine cycles would also be able to take advantage of such future technological advances to remain competitive. Increasing the combustor outlet temperature by 80C (SCOC-CC) and 50C (NET Power), and increasing the allowable turbine material temperature by 90C increased the efficiencies of the NET Power and SCOC-CC cycles by 1.6 and 0.5 percentage points respectively. Simply increasing the combustor outlet temperature without also increasing the metal temperature produced almost no improvement in the efficiency.

Percentage capture and CO_2 purity

CO_2 purity specifications for CCS are not yet clearly defined and they may vary between different applications, e.g. EOR and saline reservoir storage. In this study the CO_2 for storage is specified to have a conservative oxygen concentration of 100ppmv. This is achieved using a low temperature CO_2 purification unit which removes O_2 and also other impurities, mainly N_2 and Ar, resulting in an overall CO_2 purity of 99.6–99.8%. The vent gas stream of impurities also includes some CO_2, resulting in incomplete CO_2 capture. The base case plants in this study were designed to achieve 90% CO_2 capture, in common with IEAGHG's other techno-economic studies on CO_2 capture, but higher percentage capture could be achieved if required. If lower purity CO_2 were acceptable the CO_2 purification unit could be removed, in which case essentially 100% of the CO_2 would be captured. Alternatively if a high purity CO_2 product were required the vent gas from the low temperature purification unit could be processed, for example in a membrane unit. A substantial portion of the CO_2 would be recovered, resulting in around 98% overall CO_2 capture. These schemes for high percentage capture were assessed for the NET Power cycle and the results are summarised in Table 4.

Table 4 Sensitivity to percentage capture and CO_2 purity

CO_2 Capture	CO_2 Purity	Efficiency	TPC	LCOE	CAC
%	%	%	€/kW	€/MWh	€/t
90	99.8	55.1	1320	83.6	68
98	99.8	54.7	1340	84.8	65
100	97.9	55.3	1270	82.7	58

Oxygen purity

Nitrogen and argon enter the cycles in the natural gas and oxygen streams. These gases increase the duty of the CO_2 product compression and purification units and they can also increase the energy required for CO_2 re-pressurisation in very high pressure cycles such as the NET Power cycle in which the CO_2 forms a liquid during re-pressurisation. These disadvantages can be reduced by producing higher purity oxygen but this increases the power consumption and cost of the air separation unit (ASU). Sensitivity cases indicated that the oxygen purities selected for the study, i.e. 99.5% for the NET Power cycle and 97% for the other cycles are close to the optimum.

High-N_2 natural gas

Some sources of natural gas include significant concentrations of N_2. The effects on the cycles are similar to those of using lower purity oxygen. Using natural gas with 14% vol. N_2 instead of the base case of 0.9% reduced the efficiency of the S-Graz cycle by 0.2%.

High-CO_2 natural gas

Natural gas from some fields has a high CO_2 concentration. Oxy-combustion turbines may be an attractive option for use of such gas, when CO_2 abatement is required. If the quantity of CO_2 entering the cycle with the natural gas increases, the quantity of CO_2 or H_2O recycled to the turbine is correspondingly reduced, with little overall impact on the power generation cycle. The main impact is an increase in the throughput and power consumption of the CO_2 product compression and purification unit. Use of natural gas with 70% CO_2 in a Modified S-Graz cycle resulted in a 5.5% point reduction in efficiency and a 20% increase in the capital cost per kW of net output. In contrast, because the quantity of CO_2 captured is about 3 times higher than in the base case, the cost per tonne of CO_2 avoided is substantially lower.

Ambient conditions

The efficiencies of power cycles in general decrease when the ambient temperature increases. This is in line with the fundamental thermodynamic principle that the efficiency depends on the difference between the upper and lower absolute temperatures of the cycle. Increasing the ambient air temperature from 9 to 25C reduces the efficiencies of the Modified S-Graz and SCOC-CC cycles by 2.4-2.7 percentage points.

Alternative cooling water system

The base case cooling water system for this study was assumed to be natural draught cooling towers, in common with IEAGHG's recent study on coal fired plants with CCS[4]. The other common choice for natural gas power plants is mechanical draught cooling towers. A sensitivity case of the NET Power cycle was assessed in which natural draught towers with an approach of 7C were replaced by mechanical draught cooling towers with an aggressive approach of 4C. Using mechanical draught cooling towers increased the thermal efficiency from 55.1% to 55.4%, because the power requirement for cooling tower fans was more than offset by reductions in the compression power requirements, mainly the recycle gas compression, as well as small reductions in the ASU and the final CO_2 compression and purification unit. The Total Plant Cost reduced from 1320 to 1245 €/kW, the LCOE reduced from 83.6 to 81.7 €/MWh and the CO_2 avoidance cost reduced from 67.6 to 61.5 €/t CO_2.

The availability of water for cooling is an important constraint at some power plant locations. The water requirement for the cooling system could be avoided by using dry air coolers but this would reduce the overall plant efficiency. The use of air cooling was not assessed in this study but it could be assessed in future. The impacts of air cooling will depend on the ambient conditions, the type of power generation cycle and the cooling system design specifications, so it is recommended that several cases should be assessed as part of an overall study on the impacts of water availability on CCS plants. Because almost all of the water produced by combustion of natural gas is condensed in oxy-combustion turbine plants, use of air cooling would make the plants net producers of water.

Economic sensitivities

There is significant uncertainty in the estimated costs of innovative equipment used in the oxy-combustion cycles. The proportion of innovative equipment, mainly gas turbines and high temperature/high pressure heat exchangers, is different in the different cycles. The sensitivity of LCOE to variations in the costs of innovative equipment is shown in Figure 3.

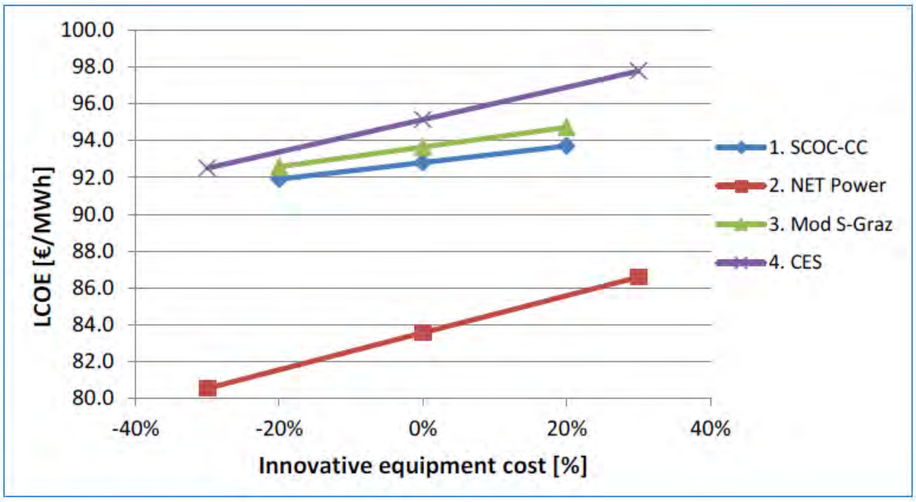


Figure 3 Sensitivity of LCOE to costs of novel equipment

The costs of CCS also depend on economic parameters that will vary over time and between different plant locations. The sensitivities of LCOE and CAC to the natural gas price, economic discount rate, plant life, cost of CO_2 transport and storage, operating capacity factor and the cost penalty for non-captured CO_2 emissions were evaluated for all of the cycles and the results are presented in the main report. As an example the results for the NET Power cycle are shown in Figures 4 and 5, in which the green bars represent increases from the base case and the red bars are reductions.

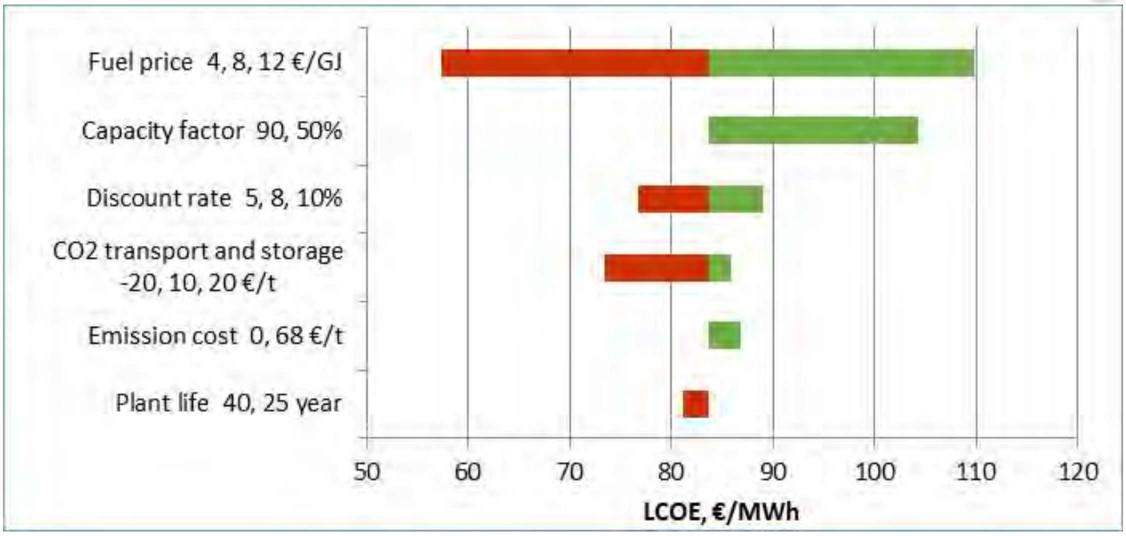


Figure 4 Sensitivity of Levelised Cost of Electricity

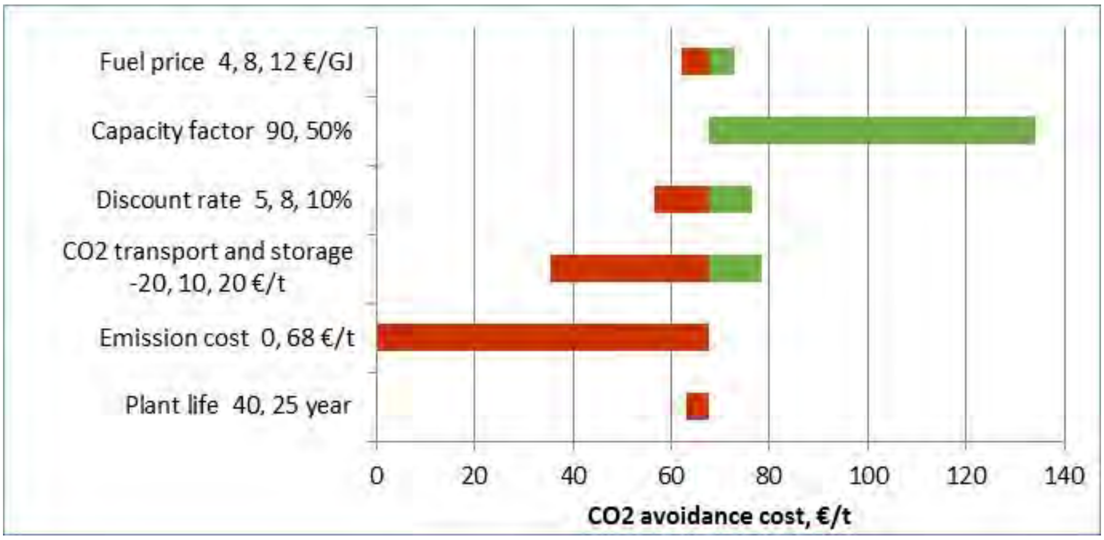


Figure 5 Sensitivity of CO_2 Avoidance Cost

The greatest sensitivity of LCOE is to the natural gas price. Gas prices vary substantially and in many parts of the world are now substantially lower than the €8/GJ base case price used in this study, for example due to the recent drop in global energy prices in general and high availability of shale gas. Reducing the annual capacity factor to 50% results in a substantial increase in the LCOE but if this is because the plant is only operated at times of relatively high power prices and is shut down when the power price is lower than the marginal operating cost, the overall economic viability of the plant may not necessarily be adversely affected. The next most significant sensitivity is to the economic discount rate. Doubling the CO_2 transport and storage cost to 20 €/t CO_2 stored has a relatively small impact on the LCOE but if CO_2 could be sold for EOR for 20 €/t for example, there would be a significant reduction in the LCOE. In this study there was assumed to be no cost associated with emissions of non-captured CO_2. A carbon tax of €68/t CO_2, equivalent to the base case cost of CO_2 avoidance of the NET Power plant, would result in only a small increase in the

LCOE. Increasing the operating lifetime of the plant from 25 to 40 years would have only a small impact on the LCOE, because of the effects of economic discounting.

The impacts of the economic parameters on CO_2 avoidance cost are substantially different to their impacts on LCOE. The avoidance cost is a function of the difference between the cost of the oxy-combustion turbine plant and the reference plant. Fuel price has only a small impact because it depends only on the relatively small difference between the efficiencies of the reference plant and the oxy-combustion turbine plant. Reducing the capacity factor has a much larger impact because the capital costs of oxy-combustion turbine plants are much higher than the cost of the reference plant, as shown in Table 2. CO_2 transport and storage cost has a much larger impact on the CO_2 avoidance cost than LCOE because it has no impact on the cost of the reference plant. In contrast, increasing the emission cost increases the cost of the reference plant but has only a small impact on the oxy-combustion plant.

Coal gasification plants

All of the oxy-combustion cycles assessed in this study could in principle be combined with coal gasification plants, as an alternative to IGCC with pre-combustion capture. In this study a plant involving coal gasification and a SCOC-CC was assessed. The plant uses the GE slurry feed, oxygen blown radiant/quench gasification process with fuel gas desulphurisation using the Selexol solvent scrubbing process. The inclusion of fuel gas desulphurisation is a conservative design assumption. It may be possible instead to remove sulphur compounds from the turbine exhaust gas, which may increase the thermal efficiency and reduce costs but it would also increase the risk of corrosion.

The performance and costs of a coal gasification SCOC-CC plant are summarised in Tables 5 and 6, Data for a reference Supercritical Pulverised Coal (SC-PC) plant without capture, a SCPC plant with post combustion capture using a proprietary solvent, a SC-PC oxy-combustion plant and an Integrated Gasification Combined Cycle (IGCC) plant with pre-combustion capture are also provided from a recent IEAGHG study carried out on the same basis by the same contractor[4]. The costs in that study are on a 2Q 2013 basis but there has been no significant change in costs of power plants in Europe between then and the costing date of this study (2Q 2014)[3]. The efficiency of the gasification SCOC-CC plant is broadly similar to the efficiencies of the other coal CCS technologies. However, the costs of the gasification SCOC-CC plant are higher than those of the other technologies, particularly the SC-PC plants.

Table 5 Performance of coal-fired plants

	Net power output	CO_2 captured	CO_2 emissions	Efficiency		Efficiency penalty for capture (LHV)
				HHV	LHV	
	MW	kg/MWh	kg/MWh	%	%	% points
Reference SC-PC	1030	-	746	42.2	44.1	-
SC-PC post combustion	822	840	93	33.6	35.2	8.9
SC-PC oxy-combustion	833	823	92	34.1	35.7	8.4
Conventional IGCC	874	844	94	33.3	34.9	9.2
Gasification SCOC-CC	740	876	94	32.5	34.0	10.1

Table 6 Costs of coal fired plants

	Total Plant Cost (TPC)	Total Capital Requirement (TCR)		Levelised Cost of Electricity (LCOE)		CO_2 avoidance cost (CAC)
	€/kW	€/kW	% increase	€/MWh	% increase	€/tonne
Reference SC-PC	1450	1890	-	52	-	-
SC-PC post combustion	2770	3600	91	95	82	65
SC-PC oxy-combustion	2760	3580	91	92	76	61
Conventional IGCC	3080	4240	124	114	120	96
Gasification SCOC-CC	3580	4920	160	128	146	116

Operating Flexibility

Power plants must face the challenges of liberalised electricity markets with variable electricity demands and high amounts of variable renewable electricity generation. Plants have to be able to operate flexibly and this is particularly so for gas fired power plants which have relatively high variable costs of operation (although relatively low fixed costs).

Due to the early status of technology development, specific information on the operating flexibility of oxy-combustion turbine plants is not yet in the public domain. The main limitation on the start-up time and ramp rate may be the ASU. Temporary storage of oxygen could overcome these constraints and also enable the throughput of the ASU to be reduced to provide increased net generation at times of peak electricity demand.

Plant area

Plant area is important in some cases, for example for retrofits to existing compact sites and application in industrial sites. For combined cycles such as the SCOC-CC and S-Graz cycles the only potential advantage in terms of space requirement is the lower plot area of the ASU and CPU compared to a post combustion capture unit. On the other hand, the potential for space saving is significant for a regenerative cycle such as NET Power. NET Power claims a footprint about a third that of a combined cycle with a similar output. The main limitation to the use of oxy-combustion turbines in compact plants is the space required for the ASU, which alone accounts for around 25% additional space with respect to a conventional combined cycle. A possible way to overcome this constraint would be to supply oxygen by pipeline from an off-site ASU.

Expert Review Comments

Comments on the draft report were received from reviewers at academic and industrial organisations involved in R&D on oxy-combustion turbine cycles and power generation and CCS in general. The contribution of the reviewers is gratefully acknowledged.

The reviewers' comments were mostly detailed and helpful suggestions to improve the clarity of the presentation and some questions regarding the design bases and differences between the cycle performance predictions in this study and other specific publications. The contractor provided IEAGHG with detailed responses which adequately addressed all of the comments and they made appropriate modifications to the report.

The two industrial cycle developers, namely CES and NET Power, continued to provide helpful information and comments after the draft report was issued. CES proposed an additional variant of their cycle based on a high pressure/high temperature supercritical turbine, to complement their base cases, which use a more conservative near-term design. IEAGHG decided that assessment of this cycle would be a worthwhile addition to the study. NET Power requested that IEAGHG include an additional case that uses mechanical draught cooling towers instead of natural draught cooling towers, which corresponds to the design and costing basis they have used internally. An extra sensitivity case was included in the study to address this suggestion. NET Power provided a helpful commentary on the differences between their own performance and cost assessments and those of Amec Foster Wheeler, and this was included in the study report.

Conclusions

- The predicted thermal efficiencies of the cycles assessed in this study range from 55% (LHV basis) for the NET Power cycle to around 49% for the other base case cycles. For comparison, a recent IEAGHG study predicted an efficiency of 52% for a natural gas combined cycle plant with post combustion capture using a proprietary solvent.

- There is scope for improving the thermal efficiencies in future for example by making use of materials capable of withstanding higher temperatures. Proprietary improvements by process developers may also result in higher efficiencies.

- The levelised cost of electricity (LCOE) of base-load plants using natural gas at 8 €/GJ are estimated to be 84-95 €/MWh, including CO_2 transport and storage costs. The lowest cost oxy-combustion plant (NET Power) has a slightly lower LCOE than a conventional gas turbine combined cycle with post combustion capture using a proprietary solvent.

- The cost of CO_2 emission avoidance of the various cycles compared to a reference conventional natural gas combined cycle plant is 68-106 €/t CO_2 avoided.

- The base case percentage capture of CO_2 in this study was set at 90% but it was determined that it could be increased to 98% without increasing the cost per tonne of CO_2 avoided, or essentially 100% if lower purity CO_2 was acceptable.

- The water formed by combustion is condensed in oxy-combustion turbine cycles which would mean that if air cooling was used, the power plants could be net producers of water, which could be an advantage in places where water is scarce, although air cooling would reduce the thermal efficiency.

- Oxy-combustion cycles could have advantages at compact sites. The total area of an oxy-combustion combined cycle plant is estimated to be slightly less than that of a conventional combined cycle with post combustion capture. The ASU could be located off-site if required to further reduce the power plant area. In addition, regenerative oxy-combustion cycles are significantly more compact than combined cycles.

- Oxy-combustion turbines could be combined with coal gasification. The predicted thermal efficiency of a coal gasification plant with a SCOC-CC is 34% (LHV basis). This is similar to that of more conventional CCS technologies (IGCC with pre-combustion capture and supercritical pulverised coal with post combustion amine scrubbing) but the estimated capital cost and cost of electricity of the oxy-combustion turbine plant are significantly higher.

Recommendations

- IEAGHG should continue to monitor the development of oxy-combustion turbines and report on significant developments.

- As oxy-combustion turbine cycles continue to evolve, a follow-on study could be carried out in future to assess sensitivities of performance and costs to further variations of cycle configuration, operating conditions, heat integration etc. Industrial applications could also be assessed in more detail if required. Further evaluation of operating flexibility should be carried out when sufficient data become available in the public domain.

- IEAGHG should carry out similar techno-economic studies on other emerging capture technologies when sufficient input data become available.

IEA

GREENHOUSE GAS R&D PROGRAMME

OXY-COMBUSTION TURBINE POWER PLANTS

FINAL REPORT

June 2015

FWI Project # 1-BD-0764A

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : GENERAL TABLE OF CONTENT AND ABBREVIATION
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

CHAPTER A – EXECUTIVE SUMMARY

CHAPTER B – GENERAL INFORMATION

CHAPTER C.1 – OXY-FUEL CYCLE REVIEW

CHAPTER C.2 – PERFORMANCE EVALUATION OF OXY-FUEL GAS TURBINE CYCLE REVIEW

CHAPTER D –BASIC INFORMATION ON SC PC PLANT ALTERNATIVES

CHAPTER D.1 – CASE 1: SCOC-CC

CHAPTER D.2 – CASE 2: NET POWER

CHAPTER D.3 – CASE 3a: S-GRAZ

CHAPTER D.4 – CASE 3b: MODIFIED S-GRAZ

CHAPTER D.5 – CASE 4a: CES CYCLE

CHAPTER D.6 – CASE 4b: REVISED CES CYCLE

CHAPTER D.6 – CASE 4c: SUPERCRITICAL CES CYCLE

CHAPTER E – ECONOMICS

CHAPTER F.1 – POTENTIAL FOR FUTURE IMPROVEMENTS

CHAPTER F.2 – OPERATING FLEXIBILITY

CHAPTER G – INDUSTRIAL AND NICHE APPLICATIONS OF OXY-TURBINE

CHAPTER H – GE BASED IGCC w SCOC-CC

ABBREVIATIONS

AGR	Acid Gas Removal
ASU	Air Separation Unit
BEDD	Basic Engineering Design Data
BFD	Block Flow Diagram
BFW	Boiler Feed Water
BL	Battery Limits
BOP	Balance Of Plant
CAC	CO_2 Avoidance Cost
CC	Combined Cycle
CCS	Carbon Capture and Storage
CES	Clean Energy Service
COT	Combustion Outlet Temperature
CPU	Cryogenic Purification Unit
CT	Cooling Tower
DC	Direct Current
DR	Discount Rate
FW	Foster Wheeler
GOX	Gaseous Oxygen
GT	Gas Turbine
H&M	Heat and Mass
HP	High Pressure
HRSG	Heat Recovery Steam Generator
IGCC	Integrated Gasification Combined Cycle
ITM	Ion Transport Membrane
LCOE	Levelized Cost Of Electricity
LHV	Low Heating Value
LIN	Liquid Nitrogen
LOX	Liquid Oxygen
LP	Low Pressure
MAC	Main Air Compressor
MP	Medium Pressure
MWe	Mega Watt electrical
MWth	Mega Watt thermal
NEE	Net Electrical Efficiency
NPO	Net Power Output
O&M	Operating and Maintenance
OTM	Oxygen Transport Membrane
PC	Pulverised Coal
PI	Power Island
PU	Process Unit
RH	Re-Heated
S/D	Shutdown

SCOC-CC	Semi-Closed Oxy-Combustion Combined Cycle
SC PC	Super Critical Pulverised Coal
SCR	Selective Catalytic Reduction
SH	Super Heater
SRU	Sulphur Recovery Unit
ST	Steam Turbine
TCR	Total Capital Requirement
TGT	Tail Gas Treatment
TIC	Total Investment Cost
TIT	Turbine Inlet Temperature
TOT	Turbine Outlet Temperature
TPC	Total Plant Cost
U&O	Utilities & Offsite
VLP	Very Low Pressure
WWT	Waste Water Treatment

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	EXECUTIVE SUMMARY
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	N. FERRARI
CHECKED BY	:	L. MANCUSO
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Background and objectives of the study

Post-combustion capture is commonly considered to be the leading option for capture of carbon dioxide (CO_2) at natural gas fired power plants, but today there is an increasing interest in the alternative of oxy-combustion turbines, which use recycled CO_2 and/or steam as the working fluid instead of air.

With this premise, the International Energy Agency Greenhouse Gas R&D (IEAGHG) programme has contracted Foster Wheeler (FW) to perform a study that provides an independent evaluation of the performance and costs of a number of oxy-turbine plants for utility scale power generation with capture of the carbon dioxide.

The study includes the following:

- Comprehensive literature review of the most relevant systems featuring oxy-combustion gas turbine cycles, discussing the state of development of each of the cycles and their main components.
- Detailed modelling of the gas turbine for the most promising and attractive cycles, including efficiency, stage number and blade cooling requirement of the specific component. The modelling is made using a calculation code developed by Politecnico di Milano (POLIMI) for the performance prediction of commercially available gas turbines.
- Complete technical and economic modelling of selected oxy-turbine power plants, including sensitivity analyses for a range of technical design and financial parameters.
- Assessment of potential for future improvements, including high temperature turbine materials and advanced oxygen membrane separation technologies.
- High-level evaluation of performance and costs of the most promising niche markets applications of oxy-turbines, in particular smaller power plants for EOR application and thermal integration with industrial power plants.
- Assessment of oxy-turbine technology applied to coal gasification.

FW likes to acknowledge the following company/university for their fruitful support to the preparation of the report:

- Clean Energy Systems, Inc;
- Graz University of Technology;
- NET Power, LLC
- SPIG S.p.A.

2. Literature review of oxy-turbine cycles

The identified leading natural gas fired oxy-combustion turbine cycles, including ones that are being developed commercially and ones that have been proposed by academics, are the following:

- Semi-closed oxy-combustion combined cycle (SCOC-CC)
- MATIANT cycles
- NET Power cycle
- Graz cycle
- CES cycle
- AZEP cycle
- ZEITMOP cycle.

The SCOC-CC, MATIANT and NET Power cycles use recycled CO_2 as moderator of the combustion temperature, while the Graz and CES cycles use water.

The AZEP and ZEITMOP cycles differentiate from the other ones because they integrate a high temperature membrane for oxygen production (OTM) in the power cycle. Since these membranes require a hot pressurized air stream from which oxygen is separated, an externally heated air cycle is also included as main power cycle (AZEP) or as side cycle of the main CO_2 loop (ZEITMOP).

These oxy-fuel cycles have been gauged on the basis of their expected efficiency and of the technological development still required for some of their key components (refer to Table 1). An index with value between 1 and 4, which has a qualitative nature and reflects the sensibility of the authors, is attributed to each novel plant component, according to the expected efforts still required for its commercial development.
This analysis has allowed to select the following four cycles for a more detailed technical and economic assessment:

- Case 1. SCOC-CC
- Case 2. NET Power cycle
- Case 3. Graz cycle
- Case 4. CES cycle.

Table 1. Development index and cycle efficiencies of the reviewed oxy-fuel gas cycles

Description	Novel equipment development index (1)	Cycle efficiency	Cycle efficiency score	Development index penalty	Total score
SCOC-CC	Compressor, combustor, HRSG, ASU: 1 Turbine: 2	45 – 48	7	1	**6**
MATIANT	Compressor, combustors, ASU: 1 Turbine, regenerator: 2	40 – 49	7	4	**3**
E-MATIANT	Compressor, combustor, ASU: 1 Turbine, regenerator: 2	40 – 49	7	2	**5**
NET-POWER	Compressor, ASU: 1 Turbine: 3 Combustor, Main heat exchanger: 2	55 – 59	10	4	**6**
Graz	Combustor, HRSG, ASU: 1 Turbine: 2 Compressor: 2	49 – 54	9	2	**7**
CES	HRSG, ASU: 1 Turbine, combustor: 2	45 – 50	8	2	**6**
AZEP	Compressors, turbine, combustor, HRSG: 1 OTM: 4	49 – 53	9	6	**3**
ZEITMOP	Compressors, N2 expander: 1 Turbine, combustor, heaters: 2 OTM: 4	46 – 51	8	9	**-1**

(1) Development index.
1: Already available on the market or easily adaptable from commercial components;
4: Highly immature components, still at a status of lab scale material testing.

3. Technical assessment

3.1. Main plant design bases

The main plant design bases, used as common bases for each study case, are listed in the following:

- The site is a greenfield location in the North East coast of The Netherlands, at sea level and with an average reference ambient temperature of 9 °C.
- Natural gas, available at 70 bar, has a heating value of 46.5 MJ/kg (LHV) and a total inert content (nitrogen, CO_2) lower than 3%.
- Plant capacity is selected in order to fully load two gas turbines, equivalent to the commercially available, air-fired, heavy duty F-class turbine, in terms of thermal duty, mass flowrates and temperature profile.
- Net power output of the reference combined cycle plant without CO_2 capture is around 900 MWe. The net power output of the oxy-fuel plants with CO_2 capture, based on the same gas turbine thermal capacity varies in the range of 660-850 MWe, depending on the cycle type.
- CO_2 is delivered from the plant site to the pipeline at the following main conditions:

- Pressure	11	MPa
- Temperature	30	°C
- Oxygen	100	ppm
- Water	50	ppm

- The plant has access to sweet water, mainly used as make-up water for the cooling water system, this latter based on natural draft cooling tower.

3.2. Key features of oxy-turbine cycles

3.2.1. *Semi-closed oxy-combustion combined cycle (SCOC-CC)*

The SCOC-CC resembles a conventional combined cycle, whilst the gas turbine compressor recycles part of the cooled CO_2 resulting from the fuel combustion. The hot combustion products, at a pressure of around 45 bar, are expanded in the turbine and then heat is recovered in a heat recovery steam generator, feeding the steam cycle. Flue gases from the HRSG are finally cooled to nearly ambient temperature in a flue gas cooler, where most of the water in the combustion products is condensed.

3.2.2. *NET Power*

The NET Power cycle (Figure 1) utilizes carbon dioxide as working fluid in the high-pressure cycle, operating with a single turbine that has an inlet pressure around 300 bar and pressure ratio around 9. The high pressure oxy-fuel combustor burns natural gas in an oxidant stream resulting from the mixture of high-purity oxygen stream with the recycle gas stream and provides the feed to a direct-fired sCO_2 turbine. A regenerative heat exchanger transfers heat from the high temperature turbine exhaust to the high pressure recycle required to control the combustion temperature and cool the turbine blades.

3.2.3. *Modified S-GRAZ*

The Modified S-Graz Cycle (Figure 2) consists of a high-temperature cycle, including the gas turbine, the HRSG, the compressor and a high pressure steam turbine (back-pressure type) and a low temperature steam cycle. The fuel along with the nearly stoichiometric mass flow of oxygen is fed to the combustion chamber, which is operated at a pressure around 45 bar. The steam generated in the HRSG and the recycled gas mixture of water and carbon dioxide are used to cool the burners and the turbine blades.

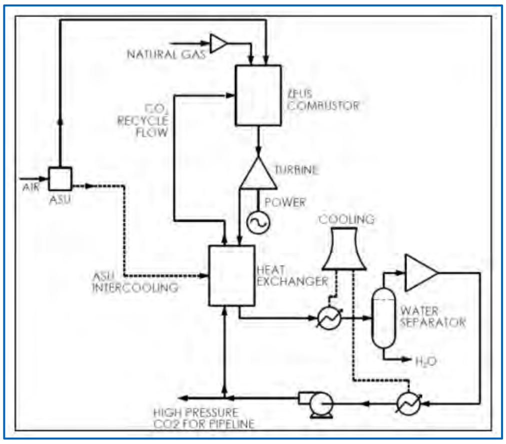


Figure 1. Configuration of the NET Power cycle

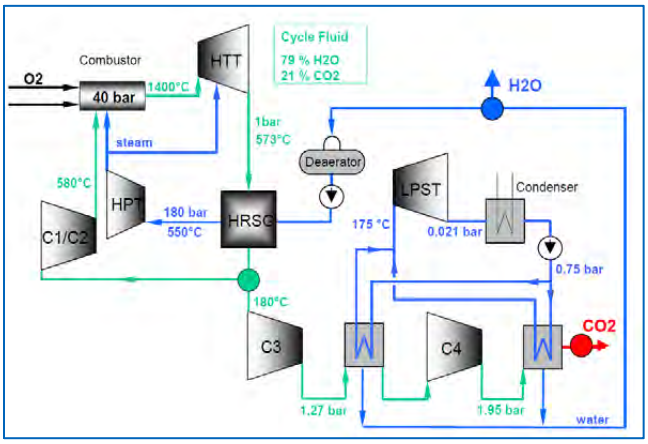


Figure 2. Configuration of the modified S-Graz cycle

3.2.4. *CES*

This cycle, proposed by Clean Energy Systems (CES) uses water, both in vapour and liquid phases, as combustion temperature moderator (Figure 3). Though different versions have been proposed, the most promising cycle consists of a high pressure oxy-fuel combustor where part of the fuel and oxidant are combusted utilizing steam in supercritical conditions as temperature moderator, while hot gas produced in the gas generator is expanded in a steam cooled HP turbine. The HP turbine exhaust gas is double-reheated by supplementary oxy-fuel combustion and further expanded in a MP and a LP section of the gas turbine, down to vacuum conditions.

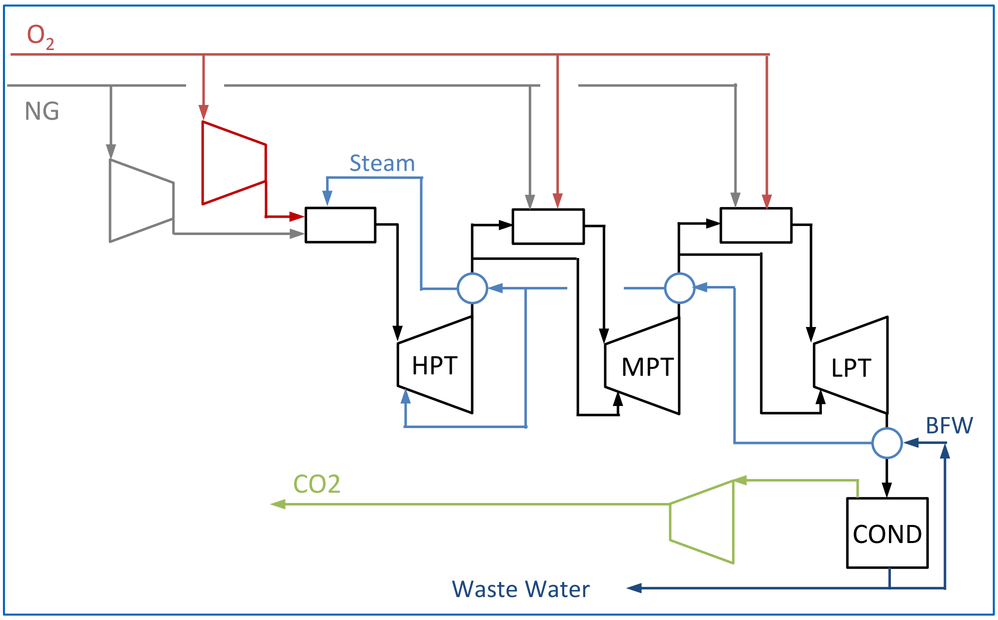


Figure 3. Configuration of the supercritical CES cycle

3.3. Performance summary

The performance data of the oxy-turbine cycles assessed in the study are summarised in the following Table 2.

Table 2. Performance summary of oxy-turbine power plants

		Case 1 SCOC-CC	Case 2 NET Power cycle	Case 3b Modified S-Graz cycle	Case 4c Supercrit CES cycle
OVERALL PERFORMANCE					
Natural gas flowrate	**t/h**	118.9	118.9	118.9	118.9
Thermal input (LHV basis)	**MWth**	1536	1536	1536	1536
Gross Electric Power Output	**MWe**	968	1056	995	992
Auxiliary power demand (1)	**MWe**	211	210	239	241
Net Electric Power Output	**MWe**	757	846	756	751
Net electrical Efficiency (LHV basis)	**%**	**49.3**	**55.1**	**49.2**	**48.9**
CO_2 REMOVAL EFFICIENCY					
CO_2 capture rate	**%**	**90.6**	**90.1**	**90.1**	**90.3**
CO_2 to storage	**t/h**	285.4	283.8	283.8	284.5
CO_2 to atmosphere	**t/h**	29.6	31.2	31.2	30.5

Notes: (1) Including step-up transformer losses.

3.4. Sensitivity to key design parameters

Plant performances have been evaluated modifying some key design parameters, such as ambient temperature, gas combustor outlet temperature, gas turbine inlet and outlet pressure, heat exchanger approaches. The results, in terms of delta efficiency with respect to the base case, are shown in the following figures for each oxy-turbine power plant considered in the analysis.

Figure 4. SCOC-CC sensitivity

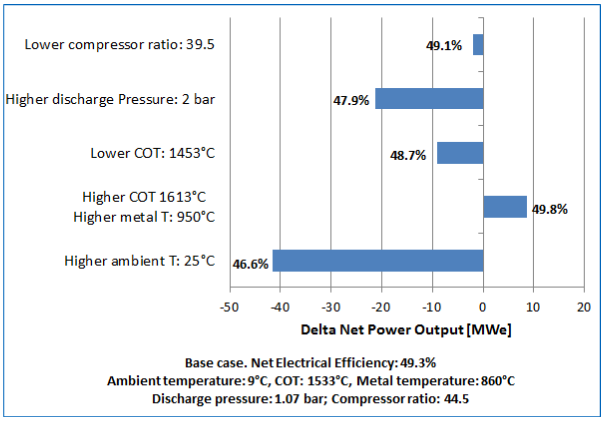


Figure 5. Modified S-Graz cycle sensitivity

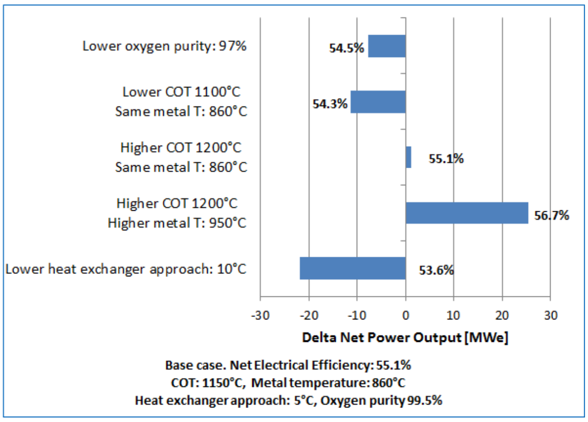
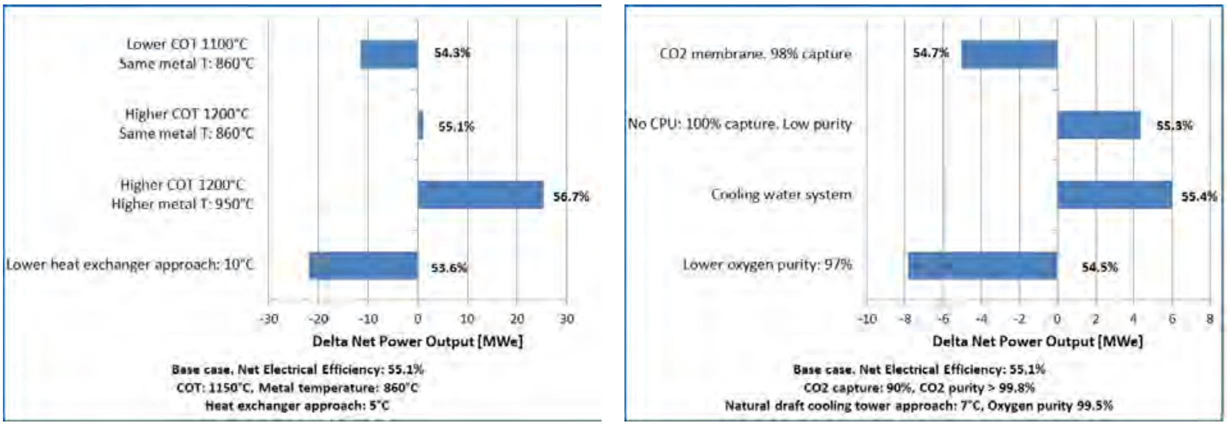


Figure 6. NET Power cycle sensitivity



Among the sensitivity analyses the following outcomes are highlighted:

- High purity oxygen (99.5%) maximizes the efficiency of the NET Power cycle because the lower power demand of the recycle compressors more than offsets the additional power demand of the ASU. For the other cycles, the optimum oxygen concentration lies in the range 95-97%.
- Future improvements on high temperature turbine materials have the potential to increase the net electrical efficiency of the oxy-turbine power plants, particularly for the NET Power cycle (~ 2 percentage points). On the other hand, the use of oxygen transport membranes in ASU may also improve significantly the project economics due to the lower power demand and related cost of the unit.

4. Economic assessment

4.1. Total investment costs

The Total Plant Cost (TPC) and the Total Capital Requirement (TCR) are defined in general accordance with the White Paper "*Toward a common method of cost estimation for CO_2 capture and storage at fossil fuel power plants*" (March 2013), produced collaboratively by authors from EPRI, IEAGHG, Carnegie Mellon University, MIT, IEA, GCCSI and Vattenfall [1].

The **Total Capital Requirement** (**TCR**) is defined as the sum of:

- Total Plant Cost (**TPC**): installed cost of the plant, including project contingencies;
- Interest during construction;
- Spare parts cost;
- Working capital;
- Start-up costs;
- Owner's costs.

The oxy-turbine power plants include novel equipment that are either under development or at conceptual stage only, so for estimating purpose they should be considered as first of a kind (FOAK) plants. The study, however, has investigated the potential of the oxy-turbine power plants with respect to benchmark technologies for capture of the CO_2, these latter generally assumed as ready for commercial application. Therefore, the study has treated the oxy-turbine cycles at Nth-of-a-kind (NOAK) plants for estimating purposes and has evaluated the cost of novel equipment as already developed and suitable for large-scale commercial application. Nevertheless, sensitivity to the cost of this equipment has been performed to take into account, to a certain extent, the intrinsic uncertainty of such estimates.

The estimate is in Euro (€), based on 2Q2014 price level. Overall expected estimate accuracy is in the range of ±35% (AACE Class 4).

The overall TPC of the different study cases and the relative weight of each unit are shown in the overleaf graphs. Total Plant Cost and Total Capital Requirement figures for the different cases, as well as the specific figures, defined as the ratio between either the TPC or the TCR and the net power output, are also reported in the following table for summary purpose.

[1] IEAGHG report 2013/TR2, http://www.ieaghg.org/publications/technical-reports

Table 3. TPC and TCR of study cases (2Q2014)

Case	**Plant type**	**TPC** (M€)	**TCR** (M€)	**Specific cost** [TPC/PO] (€/kW)	**Specific cost** [TCR/PO] (€/kW)
Case 0 (reference)	CCGT	592	773	655	855
Case 1	SCOC-CC	1,111	1,441	1,470	1,905
Case 2	Net Power	1,118	1,475	1,320	1,715
Case 3	Mod. S-Graz	1,136	1,474	1,500	1,955
Case 4	Supercrit CES	1,160	1,505	1,540	2,000

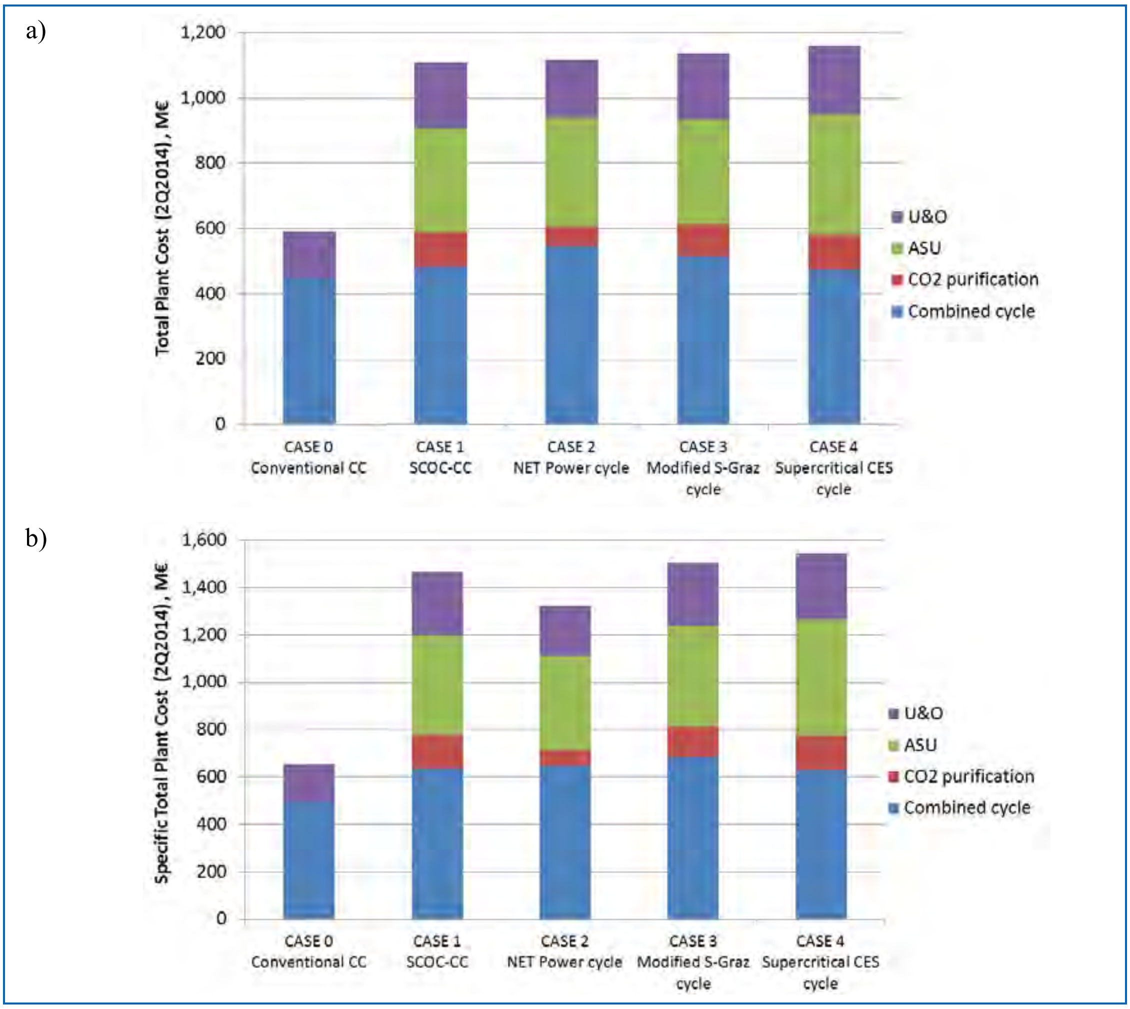


Figure 7. a) Total Plant Cost (2Q2014); b) Specific Total Plant Cost (2Q2014)

4.2. Financial analysis

4.2.1. Bases

A simplified financial analysis has been performed to estimate, for each case, the Levelized Cost of Electricity (LCOE) and the CO_2 Avoidance Cost (CAC), based on a specific set of macroeconomic assumptions.

The Cost of Electricity (COE) in power production plants is defined as the selling price at which electricity must be generated to reach the break even at the end of the plant lifetime for a targeted rate of return. However, with the purpose of screening different technology alternatives, the levelized value of the cost of electricity (LCOE), defined as the uniform annual amount that returns the same net present value as the year-by-year amounts, is commonly preferred to the COE.

The LCOE predictions are calculated under the assumption of obtaining a zero Net Present Value (NPV) for the project, corresponding to an Internal Rate of Return (IRR) equal to the Discount Rate (DR).

The CO_2 Avoidance Cost (CAC) is calculated by comparing the costs and specific emissions of a plant with CCS with those of the reference case without CCS. For a power generation plant, it is defined as follows:

$$CO_2 \text{ Avoidance Cost (CAC)} = \frac{LCOE_{CCS} - LCOE_{Reference}}{CO_2Emissions_{Reference} - CO_2Emissions_{CCS}}$$

where:

Cost of CO_2 avoidance is expressed in € per ton of CO_2
LCOE is expressed in € per MWh
CO_2 emissions is expressed in tons of CO_2 per MWh.

The selected reference case for the evaluation of the CAC is Case 0, i.e. the conventional air-fired combined cycle power plant without CO_2 capture.

The main financial bases assumed for the economic model are reported in the below table.

Table 4. Main financial bases

ITEM	DATA
Natural gas cost	8 €/GJ (LHV basis)
Discount Rate	8%
Financial leverage	100% debt
Maintenance cost (% of TPC)	2.5% (novel components) 1.5% (other units)
Capacity factor	90%

Plant life	25 years
CO_2 transport & storage cost	10 €/t $_{STORED}$
CO_2 emission cost	0 €/t $_{EMITTED}$
Inflation Rate	Constant Euro
Currency	Euro reported in 2Q2014

4.2.2. Results

A summary of the economical modeling results (LCOE and CAC) for the main cases is reported in the following Table 5 and Figure 8.

LCOE figures also show the relative weight of:

- Capital investment,
- Fixed O&M (Operating Labor costs, Overhead Charges, Maintenance costs),
- Variable O&M (Raw water make-up and Chemicals),
- Fuel,
- CO_2 transportation & storage.

Table 5. Financial results summary: LCOE and CO_2 avoidance cost

Case	Description	LCOE (€/MWh)	CAC (€/t)
Case 0	Conventional CC	62.5	-
Case 1	SCOC-CC	92.8	97.9
Case 2	NET Power cycle	83.6	67.6
Case 3	S-GRAZ cycle	93.7	101.4
Case 4	CES cycle	95.1	106.0

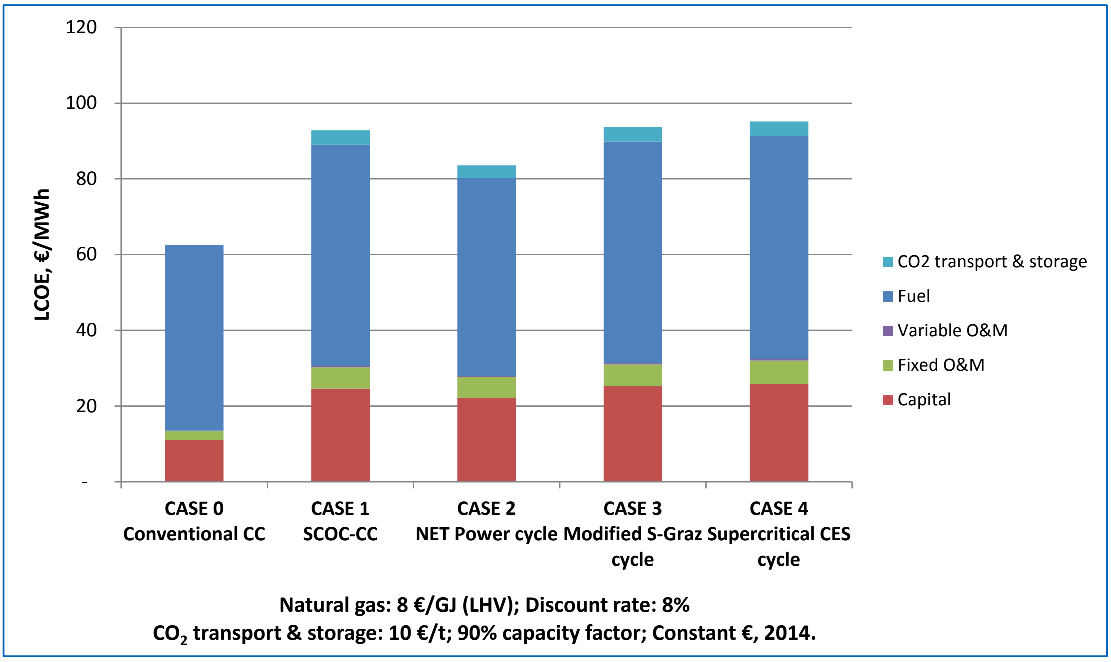


Figure 8. Levelised cost of electricity (2Q2014)

Figure 9 shows the results of the sensitivity financial analyses performed to estimate the LCOE when varying the capital cost of novel equipment, mainly the gas turbine and key heat exchangers. A wider sensitivity range is considered for the cycles with components that require more development, as defined in accordance with the development indexes of Table 1.

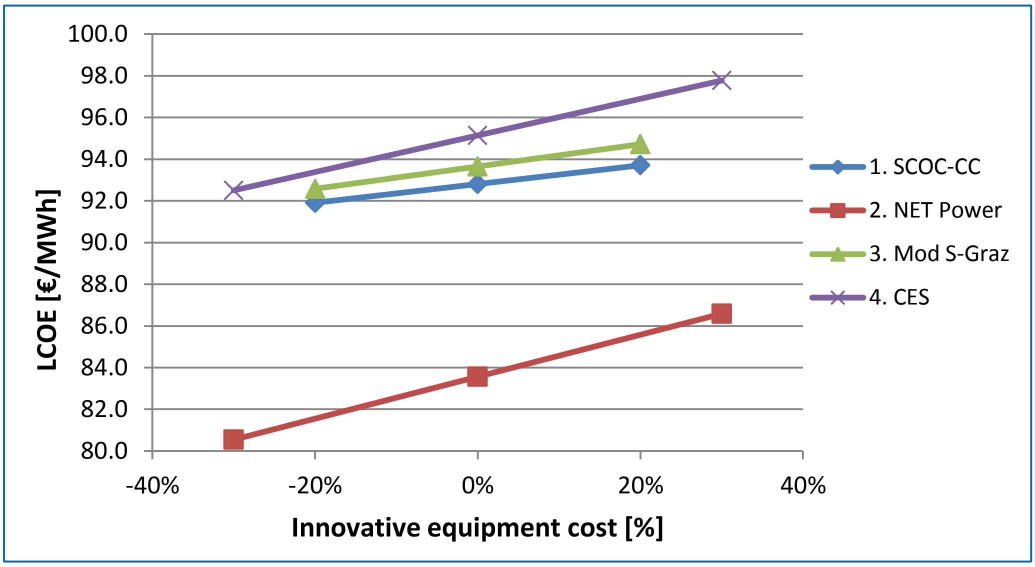


Figure 9. LCOE sensitivity to novel equipment cost

The following figures show the results of the sensitivity financial analyses performed to estimate the LCOE when varying key design parameters that affect both the net electrical efficiency (NEE) and the capital cost, such as the combustion temperature (Figure 10 and Figure 11 respectively for the SCOC-CC and the NET Power cycle), the cooling water system (Figure 12 for the NET Power cycle), the CO_2 capture rate and purity (Figure 13 for the NET Power cycle) and the oxygen purity (Figure 14 for the modified S-Graz cycle).

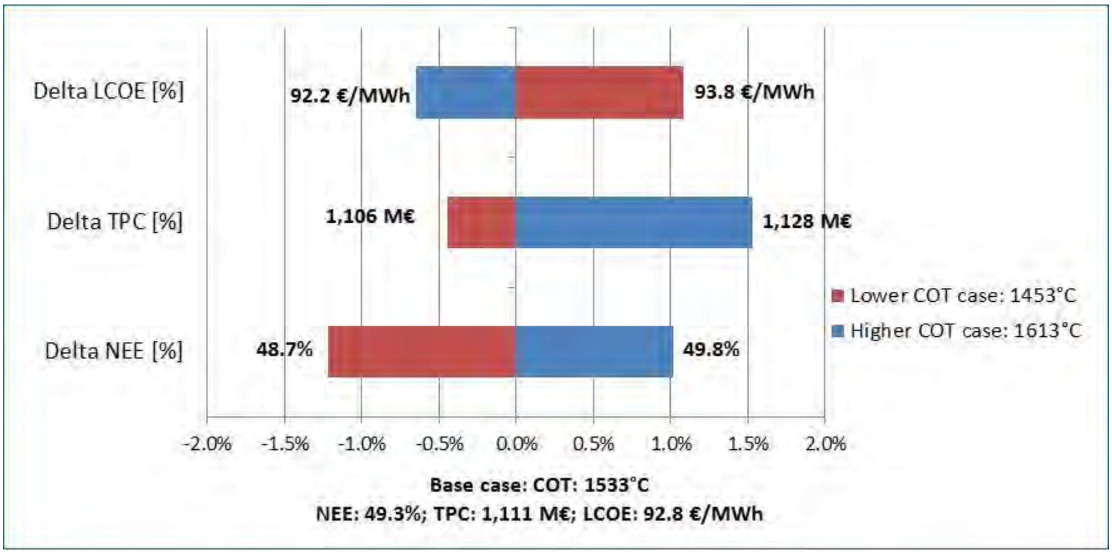


Figure 10. SCOC-CC: LCOE sensitivity COT

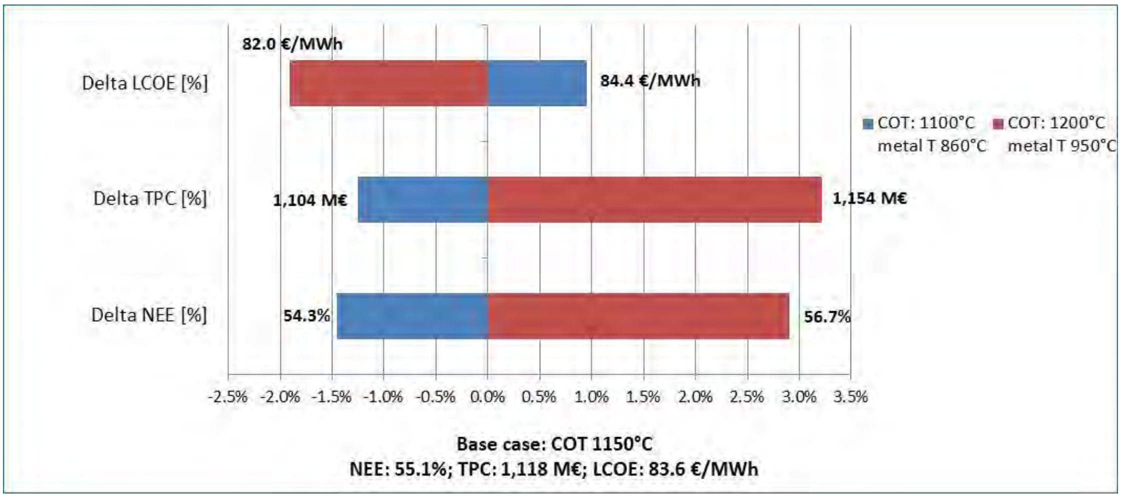


Figure 11. NET Power: LCOE sensitivity COT

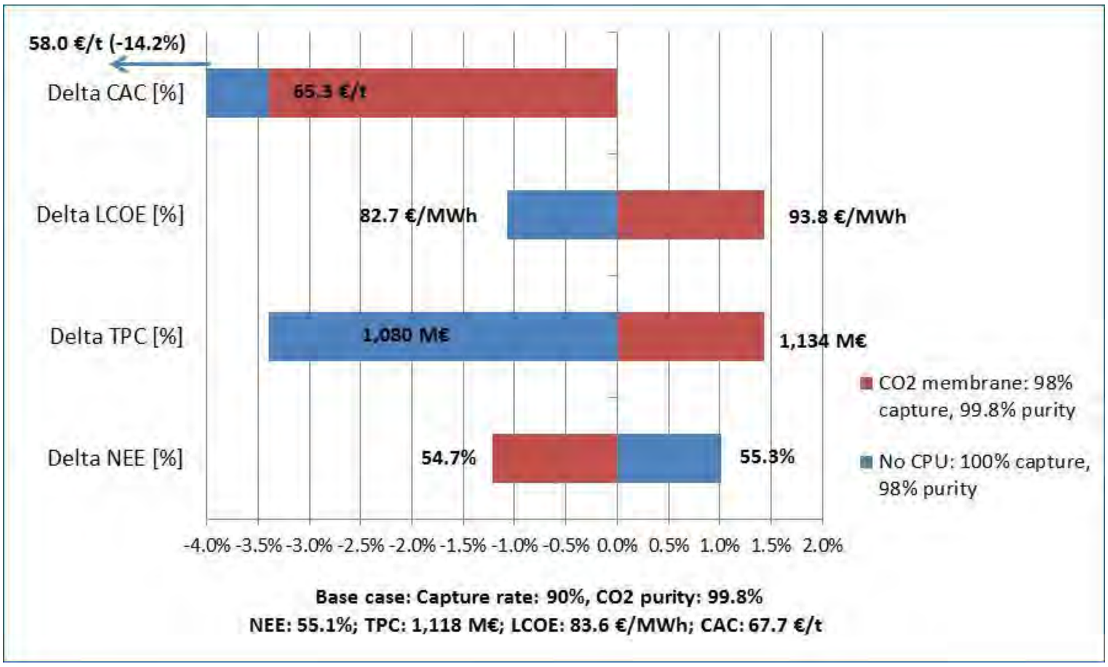


Figure 12. SCOC-CC: LCOE and CAC sensitivity CO_2 capture rate and purity

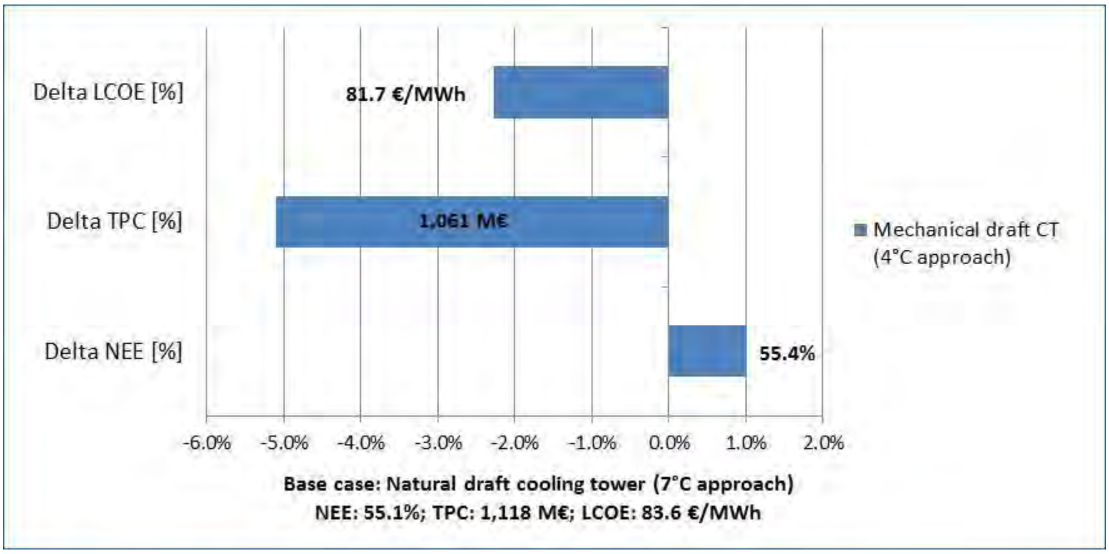


Figure 13. NET Power: LCOE sensitivity cooling water system design

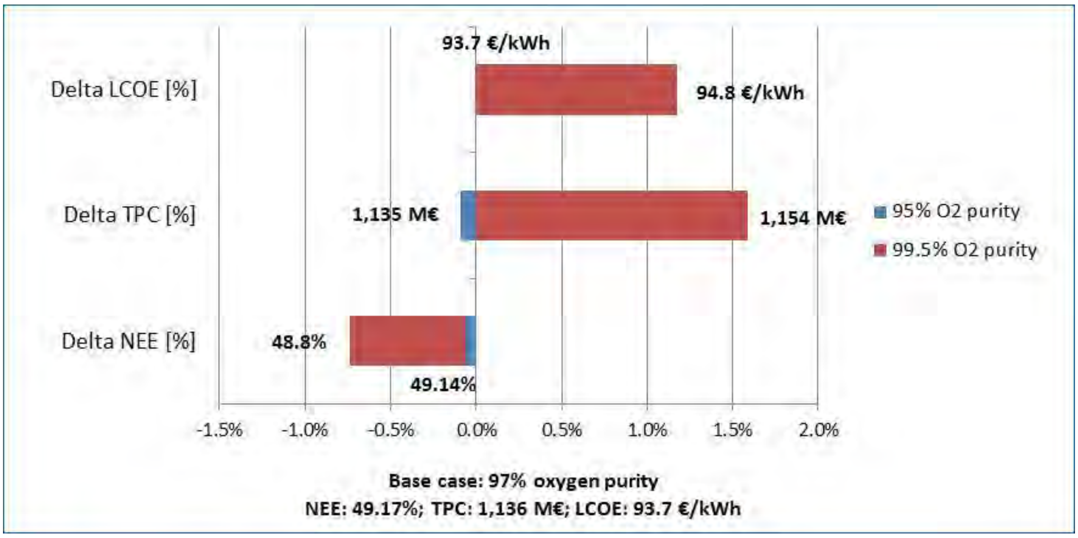


Figure 14. S-Graz cycle: LCOE sensitivity oxygen purity

Figure 15 shows the results of the sensitivity financial analyses performed to estimate the LCOE and the CAC of the main study cases versus the variation of the natural gas cost and the plant load factor.

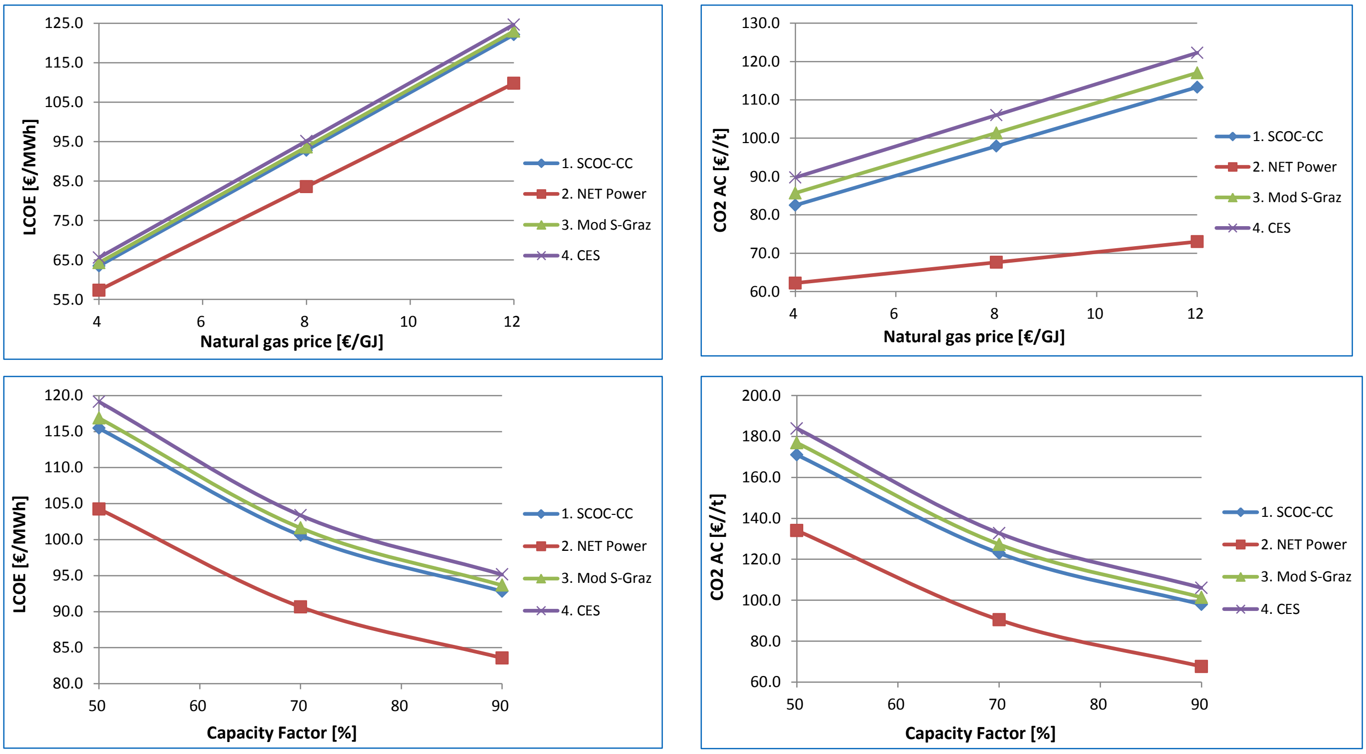


Figure 15. Sensitivity to natural gas cost and capacity factor

5. Application of oxy-turbine cycles

5.1. Industrial applications and niche markets

Use of high-CO_2 natural gas

Oxy-turbine cycles are attractive for applications based on natural gas with high CO_2 concentration because they use recycled CO_2 as working fluid instead of air, so the carbon dioxide in the natural gas leads to a lower recycle flowrate and its associated energy penalty, with a limited impact on plant performances for the captured CO_2.

For a natural gas with 70% CO_2 content, which determines a CO_2 capture quantity three times the one of the base case, the energy penalty is around 5.5 percentage points, mainly due to the larger CO_2 compressor. The associated TPC increase, related to the larger size of the CO_2 purification unit, is around 6.5% that of the base study case.

Provision of CO_2 for EOR

Oxy-turbine power plants of smaller size (e.g. 200 MWe) could be an attractive source of power and CO_2, if built near the EOR fields. Performance and cost data for SCOC-CC and Modified S-Graz cycles, each composed by one gas turbine and one downstream steam cycle, have been estimated: the efficiency reduction is 1.3 percentage points with respect to the 800 MWe utility plants assessed in the study case, mainly related to the lower efficiency of the smaller size machines, which also leads to the higher specific costs (+40%).

Energy integration with industrial sites

Some oxy-turbine cycles have a requirement for an external source of low to moderate temperature heat and others are a net producer of such heat. If a steam cycle is included, as in the SCOC-CC and the S-Graz cycles, this can be easily used as a source for district heating. In fact for the modified S-Graz power plant, an electrical efficiency of around 42% (LHV basis) and combined heat and power efficiency higher than 90% can be achieved. Vice versa additional heat available from outside plant B.L. can be used to produce additional power. In both cases, the specific total plant cost remains unchanged with respect to the reference case without heat integration (~ 1,540 €/kWh).

For a regenerative oxy-turbine cycle as the NET Power one, heat from OSBL could be used for the pre-heating of recycle streams, maximising the recycle stream temperature and consequently increasing the recycle flowrate and the power production from the gas turbine.

Applications requiring compact plants

Some oxy-turbine plants are expected to be more compact than conventional power plants with post combustion capture of CO_2, resulting in a significant advantage for energy users with limited space. As there are several differences among the oxy-turbine cycles that have an impact on the lay-out, two different categories have been identified: oxy-turbine cycles based on the 'combined cycle' concept (SCOC-CC and S-Graz) and regenerative cycles (NET Power).

The only potential advantage of the oxy-turbine 'combined cycle' concept in terms of space requirements is the lower plot area of the ASU and CPU compared to a post-combustion capture unit.

On the other hand, the potential for space saving considering a regenerative cycle is significant, as the steam cycle is not used and the key components are more compact due to the higher operating pressure. NET Power claims for their cycle application a footprint of about 1/3 the size of a combined cycle with a similar power output.

Main limitation to the use of the oxy-turbine technology to compact plants is the space requirement for the ASU, which alone accounts for around 25-30% additional space with respect to the conventional combined cycle. A possible way to overcome this constraint is the construction of the oxy-fuel power plants in a location having a nearby oxygen pipeline.

5.2. Oxy-turbine combined with coal gasification

The main performance data of the IGCC, based on GE gasification technology, with oxy-fuel combined cycles analysed in the study are summarised in the following Table 6, showing the comparison with the conventional IGCC plant based on the same gasification technology. The table also shows the performance data of the reference plant considered for the CAC evaluation, which is the SC PC plant w/o CCS, as assessed in the IEAGHG report 2014/3, '*CO_2 capture at coal based power and hydrogen plant*'.

Table 6. Coal based power plant performance summary

		SC-PC w/o CCS	GE based IGCC w SCOC-CC	GE based IGCC w conventional CC
OVERALL PERFORMANCE				
Coal flowrate (A.R.)	**t/h**	325.0	302.9	349.1
Thermal input (1)	**MWth**	2335	2177	2509
Thermal input to GT (1)	**MWth**	n/a	1536 (3)	1600 (3)
Auxiliary power demand (2)	**MWe**	47.1	339.0	266.4
Net Electric Power Output	**MWe**	1029.6	739.7	874.3
Net Electrical Efficiency (1)	**%**	**44.1**	**34.0**	**34.9**
CO_2 REMOVAL EFFICIENCY				
CO_2 capture rate	**%**	-	90.3	90.1
CO_2 to atmosphere	**t/h**	767.4	69.3	81.9
CO_2 to storage	**t/h**	-	647.7	737.9

Notes: (1) LHV basis; (2) Including step-up transformer losses; (3) 2 x equivalent F-class GT.

Total Plant Cost and Total Capital Requirement figures for the different cases are summarized in the following Table 7. Main economic modeling results (LCOE and CAC) are also reported, while LCOE breakdown into main capital and operating costs is shown in Figure 16.

The main advantage of oxy-fuel cycle applied to gasification plant is that some units and equipment for the pre-combustion capture (mainly CO shift reactors and AGR for CO_2 removal) are no longer required, reducing the overall process unit size and their investment cost.
On the other hand, larger sized ASU and cryogenic CO_2 purification units are required, whose additional consumptions and investment costs more than offset the previous advantages.

Table 7. Coal based power plant investment cost figures (2Q2014)

Case	**TPC** (M€)	**TCR** (M€)	**Specific cost** [TPC/PO] (€/kW)	**Specific cost** [TCR/NPO] (€/kW)	**LCOE** (€/MWh)	**CAC** (€/t)
SC-PC w/o CCS	1,490	1,943	1,450	1,890	52	-
GE based IGCC w SCOC-CC	2,612	3,597	3,540	4,880	126.8	114.8
GE based IGCC w conventional CC	2,668	3,705	3,080	4,240	114.4	95.8

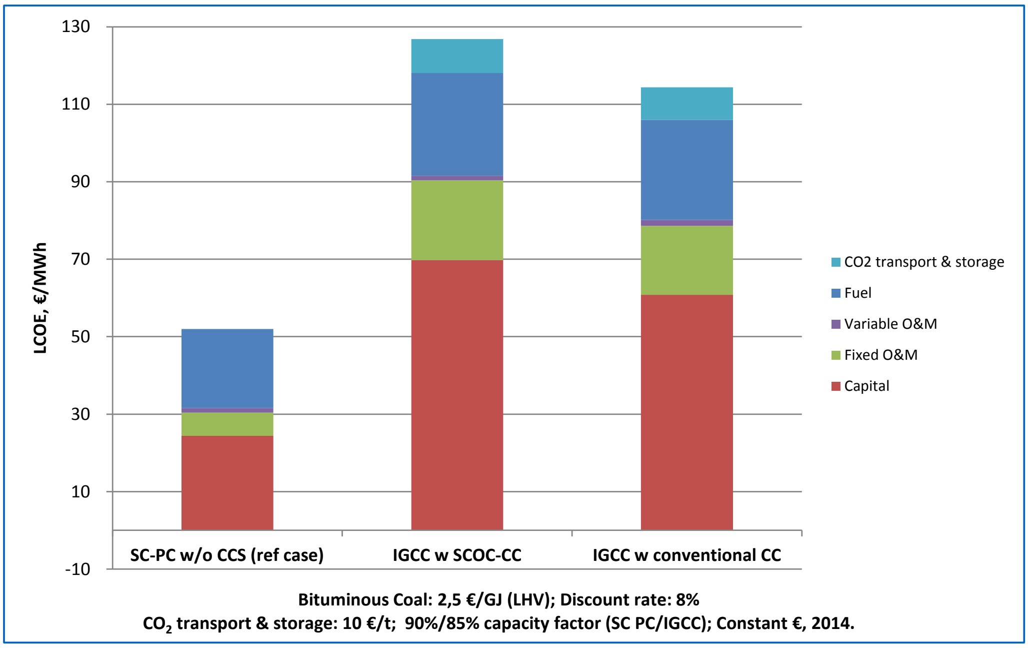


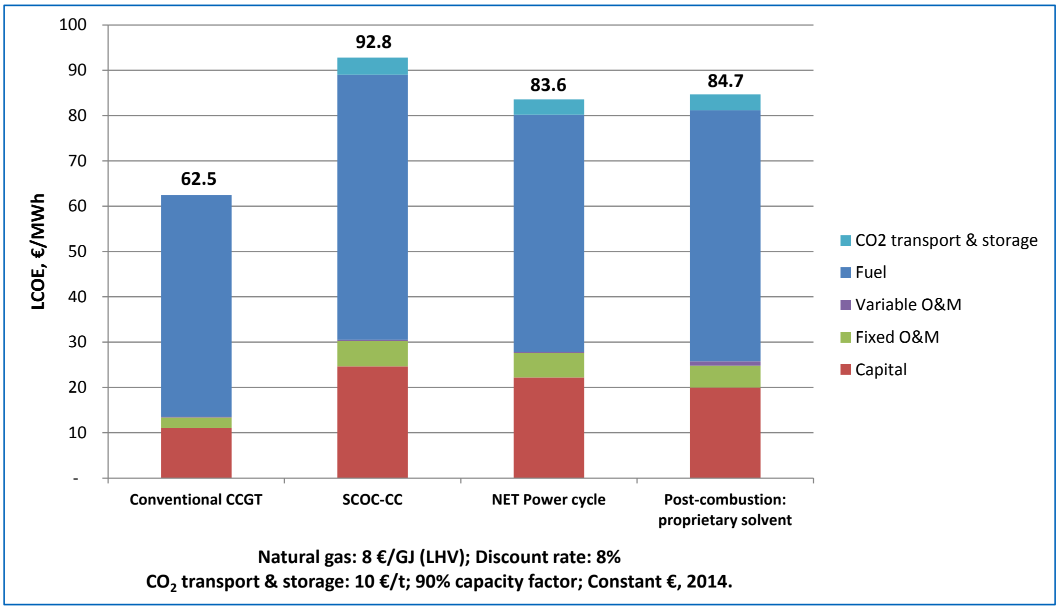
Figure 16. Levelised cost of electricity (2Q2014)

6. Summary findings

The primary conclusions of the technical and economic assessments made in this study are given below:

- The regenerative NET Power cycle is the most efficient (~55%) of the oxy turbine power plants, while the other cycles show a net electrical efficiency around 49%.
- Depending on the cycle type, the specific total plant cost of oxy-turbine power plants varies from 1,300 to 1,550 €/kWe, approximately 2-2.4 times greater than the specific cost of a standard combined cycle without CO_2 capture (655 €/kWe).
- NET Power cycle shows the best economics, mainly due to its outstanding efficiency, which leads to a LCOE of around 84 €/MWh. Mainly due to the similar net electrical efficiency, the other cycles show also a similar LCOE, in the range of 93-95 €/MWh.
- The CO_2 avoidance cost (CAC) reflects the trend of the LCOE: the lowest value is for the NET Power cycle (67 €/t), followed by the other cycles: SCOC-CC and Graz (~100 €/t) and the CES (106 €/t).

As noted already in the report, the post-combustion capture technology is commonly considered as the leading option for capture of carbon dioxide at natural gas fired power plants. Taking reference from the IEAGHG report 2012/08 '*CO_2 capture at gas fired power plants*', latest developments in the post combustion capture technology, mainly related to the new generation proprietary amine-based solvent, lowers the specific cost of the plant to around 1,200 €/kWe, i.e. approximately 10% less than the NET Power plant and 25% less than the cost of the other oxy-turbine cycles. However, the higher efficiency of the NET Power cycle (~55% vs. 52% of the post-combustion) allows decreasing the cost component of the natural gas and its related LCOE. Figure 17 displays the LCOE of two representative oxy-turbine cycles (SCOC-CC and NET Power) and the costs of conventional combined cycles with post-combustion capture, showing that the NET Power's LCOE is expected to be slightly lower (84 €/MWh vs. 85 €/MWh). Also the CO_2 avoidance cost of the NET Power cycle is now competitive with the post-combustion capture plant cost (68 €/t vs. 72 €/t).



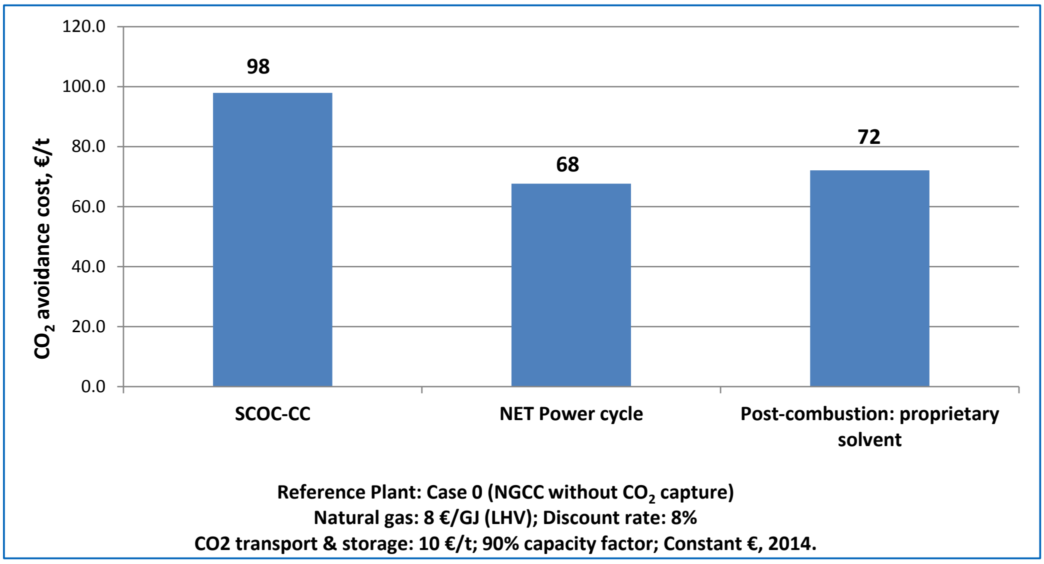


Figure 17. LCOE and CAC: Post-combustion capture vs. oxy-combustion

Likewise, it should be noted that NET Power and CES are commercially deploying their technologies in partnership with commercial gas turbine suppliers, like Toshiba, General Electric or Siemens, so further improvements are expected in the next years. As a matter of fact, NET Power is confident that by fully implementing their proprietary and confidential information and trade secret learnings, the efficiency of their cycle can be as high as 59%, leading to an evident economic benefit compared to the post-combustion capture technology that, as of today, has almost reached a complete level of commercial deployment, while CES claims for its supercritical cycle an efficiency target of 53%.

More in general, the study has demonstrated that the oxy-turbine power plants have the technical and economic potential in the coming years to be a valid alternative to the post-combustion capture technology, commonly considered as the leading option for capture of carbon dioxide at natural gas fired power plants.

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : GENERAL INFORMATION
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI & (*)
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

(*) E. Martelli, P. Chiesa, M. Romano, Politecnico di Milano

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Background and objectives of the study

Post-combustion capture is usually considered to be the leading option for capture of carbon dioxide (CO_2) at natural gas fired power plants, but today there is an increasing interest in the alternative of oxy-combustion turbines. Such turbines use recycled CO_2 and/or steam as the working fluid instead of air. Oxy-turbines can also be combined with solid fuel gasification processes as an alternative to IGCC with pre-combustion capture.

The International Energy Agency Greenhouse Gas R&D programme (IEAGHG) undertook a preliminary techno-economic evaluation of an oxy-turbine power plant as part of the report 2005/9, *Oxy combustion processes for CO_2 capture from power plant*. A study was published also by NETL evaluating two alternate oxy- turbine cycles.

In all of these cases the costs of oxy-turbines were higher than those of gas turbine combined cycles with post combustion capture. However, there are a large variety of oxy-turbine cycles and recently available information indicates some oxy-turbine plants could be competitive with post combustion capture.

With this premise, IEAGHG has contracted Foster Wheeler (FW) to perform a study with the main purpose of providing an independent evaluation of the performance and costs of a range of oxy-turbine cycles for utility scale power generation with capture of carbon dioxide. The study assesses also smaller scale plants, for example for use in combination with enhanced oil recovery (EOR) and other industrial applications.

The study features the following main aspects:

- Complete review of the most relevant systems featuring oxy-combustion gas turbine cycles, as available in the public domain. Conceptual plant arrangements are discussed in terms of achievable performance, technological feasibility, as well as uncertainties related to the development of non-conventional components. In addition, the main projects aiming at the development and manufacturing of critical components (in particular combustors, turbines and heat exchangers) are examined.
- A comparative thermodynamic analysis with conventional air-blown combined cycles based on the same technology level, carried out in order to assess achievable performance and identify critical aspects inherent to their technical development. Oxy-fuel combined cycles are assessed as reference benchmark plants of the oxy-combustion technology for this comparison.
- Based on the oxy-fuel cycles review, the most promising and attractive cycles are selected for a detailed technical and economic assessment. Detailed modelling of the gas turbine, including efficiency, stage number and blade

cooling requirement is also developed. Sensitivity analyses are carried out varying both technical design features and financial parameters.

- The impact on oxy-turbine cycles performance related to the application of improved high temperature turbine materials is discussed. The potential advantages related to the introduction of advanced oxygen separation technologies (e.g. membranes) in oxy-turbine based plants are also assessed. In addition to the above, any possible constraints on flexible operation, including plant start-up and shut-down, low load operation and ramp rates, and possible way to overcome them are identified.
- The performance and costs of the most promising niche markets applications of oxy-turbines are assessed at high-level, in particular EOR application, impacts of firing low-quality natural gas and thermal integration with industrial power plants. Applications requiring compact plants are also briefly discussed.
- The performance and costs of an oxy-turbine plant combined with coal gasification is assessed.

It has to be highlighted that in the technical assessment of the oxy-turbine power plants, a crucial point is the evaluation of the performance of turbomachines, particularly the high temperature expander.

The evaluation of the expander performance, mainly efficiency, cooling requirements and exhaust gas temperature for all the cases assessed in the present study are based on a calculation code and model, internally developed by Politecnico di Milano (POLIMI) for the performance prediction at design point of commercially available gas turbines.

FW likes to acknowledge the following companies/university, listed in alphabetical order, for their fruitful support to the preparation of the report:

- Clean Energy Systems, Inc
- Graz university of technology;
- NET Power, LLC
- SPIG S.p.A.

2. Project Design Bases (PDB)

This section describes the general plant design and cost estimating criteria, used as common basis for the design of the plant for the different study cases.

2.1. Location

The site is a Greenfield location on the North East coast of The Netherlands, with no major site preparation required. No restrictions on plant area and no special civil works or constraints on delivery of equipment are assumed. Rail lines, roads, fresh water supply and high voltage electricity transmission lines, high pressure CO_2 and natural gas pipeline are considered available at plant battery limits.

2.2. Climatic and meteorological data

Main climatic and meteorological data are listed in the following. Conditions marked (*) are considered reference conditions for plant performance evaluation.

• Atmospheric pressure	101.3	kPa	(*)
• Relative humidity			
average	80	%	(*)
maximum	95	%	
minimum	40	%	
• Ambient temperatures			
minimum air temperature	-10	°C	
maximum air temperature	30	°C	
average air temperature	9	°C	(*)

2.3. Plant capacity

The plant capacity of both the reference case without CCS and the oxy-fuel base cases will be selected in order to fully load two (2) gas turbines, equivalent to the commercially available conventional air fired gas heavy duty F-class gas turbine.

A 200 MWe power plant will be considered for the EOR application (refer to chapter G of the present report – Industrial and niche application of the oxy-turbine), corresponding to a cycle based on one (1) E-class equivalent gas turbine.

2.4. Feedstock specification

2.4.1. *Natural Gas*

Natural gas is delivered to the plant battery limits from a high pressure pipeline.

The main characteristics of the natural gas are shown in the following Table 1.

Table 1. Natural Gas characteristics

Natural Gas analysis, vol%	
Methane	89.0
Ethane	7.0
Propane	1.0
Butane	0.1
Pentane	0.01
CO_2	2.0
Nitrogen	0.89
Total	**100.00**

HHV, MJ/kg	51.473
LHV, MJ/kg	46.502

Conditions at plant B.L.	
Pressure, MPa	7.0

2.4.2. *Coal (IGCC case only)*

The main fuel of the IGCC based power plant is bituminous coal type, with the characteristics and properties as shown in the following Table 2.

The reference coal is an Eastern Australian internationally traded open-cast coal, assumed delivered from a port to the plant site by unit trains.

Table 2. Bituminous Eastern Australian Coal characteristics

Proximate Analysis, wt% - As Received	
Inherent moisture	9.50
Ash	12.20
Coal (dry, ash free)	78.30
Total	**100.00**

Ultimate Analysis, wt% - Dry, ash free	
Carbon	82.50
Hydrogen	5.60
Oxygen	8.97
Nitrogen	1.80
Sulphur	1.10
Chlorine	0.03
Total	**100.00**

Ash analysis, wt%	
SiO_2	50.0
Al_2O_3	30.0
Fe_2O_3	9.7
CaO	3.9
TiO_2	2.0
MgO	0.4
Na_2O	0.1
K_2O	0.1
P_2O_5	1.7
SO_3	1.7

HHV (As Received), MJ/kg (*)	27.06
LHV (As Received), MJ/kg (*)	25.87
Grindability, Hardgrove Index	45
Ash Fusion Temperature at reduced atm., °C	1350

(*) based on Ultimate Analysis, but including inherent moisture and ash.

2.5. Products and by-products

The main products and by-products of the study cases are listed here below, together with their main characteristics.

2.5.1. Electric Power

Grid Connection Voltage:	380	kV
Electricity Frequency:	50	Hz
Fault duty:	50	kA

2.5.2. *Carbon Dioxide*

Plants are generally designed for a capture rate not less than 90%.

CO_2 is delivered from the plant site to the pipeline at the following conditions and characteristics.

Table 3. CO_2 characteristics

CO_2 conditions at plant B.L.	
Pressure, MPa	11
Maximum Temperature, °C	30

CO_2 maximum impurities, vol. Basis (0)	
H_2	4% (1,3)
N_2 / Ar	4% (2,3)
CO	0.2% (5)
H_2O	50 ppm (4)
O_2	100 ppm (6)
H_2S	20 ppm (7)
SO_X	100 ppm (5)
NO_X	100 ppm (5)

(0) Based on information available in 2012 on the requirements for CO_2 transportation and storage in saline aquifers
(1) Hydrogen concentration to be normally lower to limit loss of energy and economic value. Further investigation is required to understand hydrogen impact on supercritical CO_2 behaviour.
(2) The limits on concentrations of inerts are to reduce the volume for compression, transport and storage and limit the increase in Minimum Miscibility Pressure (MMP) in Enhanced Oil Recovery (EOR).
(3) Total non-condensable content (N_2 + O_2 + H_2 + CH_4 + Ar): maximum 4% vol. basis.
(4) Water specification is to ensure there is no free water and hydrate formation.
(5) H_2S, SO_2, NO_2 and CO limits are set from a health and safety perspective.
(6) O_2 limit is tentative in view of the lack of practical experience on effects of O_2 in underground reservoirs. EOR may require tighter specification.
(7) H_2S specification is for a corrosion and pipeline integrity perspective.

2.5.3. *Sulphur (IGCC case only)*

Sulphur characteristics at IGCC plant B.L. are the following:

Status:	solid/liquid
Colour:	bright yellow
Purity:	99.9%wt. S (min)

H_2S content:	10 ppm (max)
Ash content:	0.05%wt (max)
Carbonaceous material:	0.05%wt (max)

2.6. Environmental limits

The environmental limits set up for each case are outlined hereinafter.

2.6.1. Gaseous emissions

The overall gaseous emissions from the plant to air do not exceed limits, as specified in the EU directives 2010/75/EU (Part 2 of Annex V).

These emission figures, expressed in mg/Nm^3 @15% O_2, dry basis, are applicable to the air fired combined cycle plants only. For the oxy-combustion fired power plant this is not relevant, due to the very low flowrate of the inerts gas stream discharged to atmosphere from the CO_2 purification process. Regulatory approaches for this plant type are not yet defined.

2.6.2. Liquid effluent

Characteristics of waste water discharged from the plant comply with the standard limits included in the EU directives currently in force.

The main continuous liquid effluent is the blow-down from the cooling towers. Effluent from the Waste Water Treatment is generally recovered and recycled back to the plant as process water, where possible, or discharged to the final receiver.

2.6.3. Solid wastes

No significant solid waste is foreseen from the oxy-turbine power plant.

The solid wastes of the IGCC based case are:

- Slag, which is potentially saleable to the building industry
- Filter cake, which contains some toxic compounds.

Other potential solid wastes are typical industrial plant waste (e.g. sludge from Waste Water Treatment etc.).

2.6.4. Noise

All the equipment of the plant are designed to obtain a sound pressure level of 85 dB(A) at 1 meter from the equipment.

2.7. Availability

The table hereafter reports the expected maximum availability (average yearly load factor) assumed for each study case, along with the availability curve for the first years of operation.

Plant type	Year	Average Load factor
Oxy-turbine power plant (NGCC ref case)	1st year of operation	65%
	2nd year of operation	85%
	3rd – 25th year of operation	90%
IGCC	1st year of operation	60%
	2nd year of operation	80%
	3rd – 25th year of operation	85%

2.8. Cost estimating bases

The following sections describe the main cost estimating bases used to make the economic assessment of the various cases.

2.8.1. *Total Capital Requirement*

The Total Capital Requirement (TCR) includes:

- Total Plant Cost (TPC)
- Interest during construction
- Spare parts cost
- Working capital
- Start-up costs
- Owner's costs.

The estimate is in euro (€), based on 2Q2014 price level.

2.8.2. *Total Plant Cost*

The Total Plant Cost (TPC) is the installed cost of the plant including contingencies.

The TPC is broken down into the main process units and, for each unit, split into the following items:

- Direct materials
- Construction
- Other costs
- EPC services
- Contingency.

2.8.3. *Estimate accuracy*

Estimate accuracy is in the range of ±35% (AACE Class 4).

2.8.4. *Contingency*

A project contingency is added to the capital cost to give a 50% probability of a cost over-run or under-run.

For the accuracy considered in this study, FW's view is that contingency should be in the range of 10-15% of the total installed cost. 10% is assumed for this study for the conventional units of the plant, for consistency with the other IEAGHG studies.

A process contingency is not added to the plant cost, because novel technology in the power island are considered as state-of-art gas equipment and the process units as CPU and ASU are not considered to be at very early stage of development and their design, performance, and costs are not highly uncertain.

2.8.5. *Design and construction period*

Plant design and construction period and curve of capital expenditure during construction depend on the plant type, as detailed in the following table.

	NGCC ref. case Oxy-turbine cases	IGCC cases
Construction period [1]	3 years	4 years
Curve of capital expenditure		
Year	Investment cost %	
1	20	15
2	45	40
3	35	30
4	-	15

Note: (1) Starting from issue of Notice to Proceed to the EPC contractor

2.8.6. *Financial leverage (debt / invested capital)*

All capital requirements are treated as debt, i.e. financial leverage equal to 100%.

2.8.7. *Discount rate*

Discount cash flow calculations are expressed at a discount rate of 8%.

2.8.8. *Interest during construction*

Interest during construction is calculated from the plant construction schedule and interest rate is assumed same as the discount rate. Expenditure is assumed to take place at the end of each year and interest during construction payable in a year is calculated based on money owed at the end of the previous year.

2.8.9. *Spare parts cost*

0.5% of the TPC is assumed to cover spare part costs. It is assumed that spare parts have no value at the end of the plant life due to obsolescence.

2.8.10. *Working capital*

Working capital includes inventories of fuel and chemicals (materials held in storage outside of the process plants). Storage for 30 days at full load is considered for coal, chemicals and consumables.

It is assumed that cost of these materials is recovered at the end of the plant life.

2.8.11. *Start-up cost*

Start-up costs consist of:

- 2 percent of TPC, to cover modifications to equipment that needed to bring the unit up to full capacity.
- 25% of the full capacity fuel cost for one month, to cover inefficient operation that occurs during the start-up period.
- Three months of operating and maintenance labour costs, to include training.
- One month of catalysts, chemicals and waste disposal and maintenance materials costs.

2.8.12. *Owner's cost*

7% of the TPC is assumed to cover the Owner's cost and fees.

Owner's costs cover the costs of feasibility studies, surveys, land purchase, construction or improvement to roads and railways, water supply etc. beyond the site boundary, owner's engineering staff costs, permitting and legal fees, arranging financing and other miscellaneous costs. Owner's costs are assumed to be all incurred in the first year of construction, allowing for the fact that some of the costs would be incurred before the start of construction.

2.8.13. *Insurance cost*

0.5% of the TPC is assumed to cover the insurance cost.

2.8.14. *Local taxes and fees*

0.5% of the TPC is assumed to cover the Local taxes and fees.

2.8.15. *Decommissioning cost*

For fossil fuel and CCS plants the salvage value of equipment and materials is normally assumed to be equal to the costs of dismantling and site restoration, resulting in a zero net cost of decommissioning.

2.9. Operating and Maintenance costs

Operating and Maintenance (O&M) costs include:

- Chemicals
- Catalysts
- Solvents
- Raw Water make-up
- Direct Operating labour
- Maintenance
- Overhead Charges.

O&M costs are generally allocated as variable and fixed costs.

Variable costs depend on the plant operating load. They can be expressed as €/kWh or €/h.

Fixed operating costs are essentially independent from the plant operating load. They can be expressed as €/y.

2.9.1. *Variable costs*

Consumables are the principal components of variable O&M costs. These include feedstock, water, catalysts, chemicals, solid waste disposal and other.

Reference values for fuels and main consumables prices are summarised in the table below.

Item	Cost
Natural gas, €/GJ (LHV)	8
Coal, €/GJ (LHV)	2.5
Raw process water, €/m^3	0.2
CO_2 transport and storage, €/t CO_2 stored (1)	10
CO_2 emission cost, €/t CO_2 emitted	0

(1) Transport and storage cost as specified by IEAGHG, in accordance with the range of costs information in the European Zero Emissions platform's report "The costs of CO_2 capture, transport and storage", published in 2009. Sensitivity to transport and storage costs are assessed to cover lower or

negative cost for EOR, due to the revenue for sale of CO_2, or higher cost, in case of off shore storage with long transport distances.

2.9.2. *Fixed costs*

The fixed costs of the different plants include the following items:

Direct labour
The yearly cost of the direct labour is calculated assuming, for each individual, an average cost equal to 60,000 €/y. The number of personnel engaged is estimated for each plant type, considering a 5 shift working pattern.

Administrative and support labour
All other company services not directly involved in the operation of the plant fall in this category, such as:

- Management
- Administration
- Personnel services
- Technical services
- Clerical staff.

These services vary widely from company to company and are also dependent on the type and complexity of the operation.

Administrative and support labour is assumed to be 30% of the direct labour and maintenance labour cost (see below).

Maintenance
A precise evaluation of the cost of maintenance would require a breakdown of the costs amongst the numerous components and packages of the plant. Since these costs are all strongly dependent on the type of equipment selected and statistical maintenance data provided by the selected supplier, this type of evaluation of the maintenance cost is premature at study level.

For this reason the annual maintenance cost of the plant is estimated as a percentage of the Total Plant Cost of each case, as shown in the following:

- Novel technologies 2.5%
- Other units (ASU, CPU, steam turbines, utilities…) 1.5%

Maintenance labour is assumed to be 40% of the overall maintenance cost.

3. Basic Engineering Design Data (BEDD)

Scope of the Basic Engineering Design Data is the definition of the common bases used for the design of the process and utility units of the different study cases.

3.1. Units of measurement

The units of measurement are in SI units.

3.2. Plant Battery Limits (main)

3.2.1. Electric Power

High voltage grid connection:	380 kV
Frequency:	50 Hz
Fault duty:	50 kA

3.2.2. Process and utility streams

Oxy-turbine power plants

- Natural gas
- Cooling tower make-up water
- Waste water streams, including cooling tower blow-down
- Plant/Raw/Potable water
- CO_2 rich stream.

Gasification plants with SCOC combined cycle

- Coal
- Natural gas (start-up, emergency, flare)
- Cooling tower make-up water
- Waste Water streams, including cooling tower blow-down
- Gasification solid wastes
- Plant/Raw/Potable water
- Sulphur product
- CO_2 rich stream

3.3. Oxy-turbine design

The following sections summarise the main operating and design parameters of the gas turbines, considered for the assessments of the different study cases.

3.3.1. *Definitions*

The below definitions are shown also in the following Figure 1.

Combustor outlet temperature (COT): Flue gas temperature at combustor outlet

Turbine inlet temperature (TIT): Flue gas total temperature at first gas turbine stator outlet, after mixing with the first cooling flow.

Turbine outlet temperature (TOT): Flue gas temperature at gas turbine expander outlet

Pressure ratio: ratio between that gas turbine compressor outlet pressure (P_1) and the compressor inlet pressure (P_0)

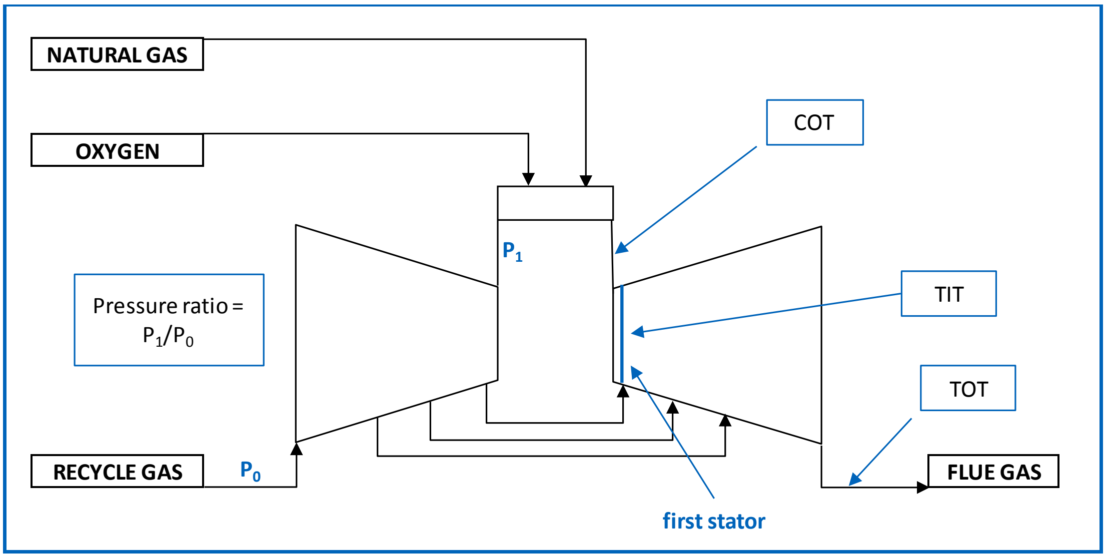


Figure 1. Main gas turbine temperature parameters

3.3.2. *Design criteria*

The following design parameters have been considered for the design of the gas turbine. These figures have been kept constant throughout the study base cases, when not set by constraints or specific design requirements of the oxy-cycle.

Oxygen excess

3% oxygen excess in the combustion chamber, accounting also the oxygen in the recycle gas, is considered in all study cases.

COT

Recycle flowrate is set in order to have a COT of 1533°C for the gas turbine in the SCOC-CC, the S-GRAZ cycle and the medium pressure gas turbine of the CES cycle.

The following exceptions are considered:

- NET Power cycle: the gas turbine COT is set to 1150°C as it corresponds to the optimum condition for the regenerative section downstream the gas turbine. In addition, higher temperature would lead to prohibitive conditions in the combustor and in the gas turbine due to its high operating pressure (i.e. 300 bar).
- CES cycle: the high pressure gas turbine is a not-cooled machine. Therefore, the COT is set to the highest value acceptable by the blades materials without any cooling, i.e. 900°C.
- Supercritical CES cycle: the high pressure gas turbine COT is set to 1150°C as higher temperature would lead to prohibitive conditions in the combustor and in the gas turbine due to its high operating pressure (i.e. 300 bar).

Pressure ratio

For the cases based on a gas turbine configuration similar to a conventional combined cycle (i.e. the SCOC-CC and the S-Graz cycle characterised by a gas turbine expanding flue gas from 40-50 bar to atmospheric pressure), the pressure ratio is fixed at 44.5, in order to keep the same temperature increase through the compressor of the reference air blown plant.

For the cycles based on high pressure expansion stages, i.e. the NET Power cycle and the CES cycle the exhaust gas pressure is set at 34 bar in order to deliver the net CO_2 product at the pressure required by the inert removal section in the CPU.

Maximum allowed blades metal temperature

The gas turbine cooling flowrate is set in order to control the blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st stator	860°C
1st rotor	830°C
from 2nd stator	830°C
from 2nd rotor	800°C

For the turbine modelled with Aspen code (i.e. NET Power cycle and supercritical CES cycle), requiring a single input for all the turbine stages, a maximum blade metal temperature of 860°C is considered.

3.4. Utility and service fluids characteristics/conditions

Following sections list main utilities and service fluids generated and distributed inside the plant.

3.4.1. *Cooling Water*

The cooling water system is based on natural draft cooling tower.

Cooling water approach to wet bulb temperature:	7 °C
Supply temperature	
- normal:	15 °C
- maximum:	36 °C
Operating pressure at user:	3.5 bar
Mechanical design pressure:	6.0 bar
Maximum allowable ΔP for users:	1.0 bar
Mechanical design temperature:	50°C
Maximum temperature difference at users:	11°C
Turbine condenser minimum ΔT:	3°C
Turbine condenser conditions	
Temperature	29°C
Pressure	4 kPa

3.4.2. *Waters*

Potable water

Source : from grid
Type : potable water

Operating pressure at grade (min):	0.8 barg
Design pressure:	5.0 barg
Operating temperature:	Ambient
Design temperature:	38°C

Raw water

Source : from grid
Type : raw water

Operating pressure at grade (min):	0.8 barg
Design pressure:	5.0 barg
Operating temperature:	Ambient
Design temperature:	38°C

Plant water

Source : from storage tank of raw water
Type : raw water

Operating pressure at grade:	3.5 barg
Design pressure:	9.0 barg
Operating temperature:	Ambient
Design temperature:	38°C

Demineralised water

Type : treated raw water

Operating pressure at grade (min):	5.0 barg
Design pressure:	9.5 barg
Operating temperature:	Ambient
Design temperature:	38°C

Characteristics:

- pH		6.5÷7.0	
- Total dissolved solids	mg/kg	0.1	max
- Conductance at 25°C	µS	0.15	max
- Iron	mg/kg as Fe	0.01	max
- Free CO_2	mg/kg as CO_2	0.01	max
- Silica	mg/kg as SiO_2	0.015	max

3.4.3. *Steam, Steam Condensate and BFW*

Steam, BFW and condensate conditions are highly dependent on the selection of the oxy-fuel cycle, and will be specified case by case in the dedicated section of the report.

3.4.4. *Instrument and Plant Air*

Instrument air

Operating pressure	
- normal:	7.0 barg
- minimum:	5.0 barg
Design pressure:	10.0 barg
Operating temperature (max):	40°C
Design temperature:	60°C
Dew point @ 7 barg:	-30°C

Plant air

Operating pressure:	7.0 barg
Design pressure:	10.0 barg

Operating temperature (max):	40°C
Design temperature:	60°C

3.4.5. *Oxygen*

Oxygen for the oxy-turbine

Oxygen pressure and temperature conditions are highly dependent on the selection of the oxy-fuel cycle, and will be specified case by case in the dedicated section of the report.

Oxygen purity considered in the study is detailed below:

SCOC-CC, S-GRAZ, CES cycle

Purity:	97% mol. O_2 min
	2.0% mol Ar
	1.0% mol N_2
H_2O content:	1.0 ppm max
CO_2 content:	1.0 ppm max
HC as CH4 (number of times the content in ambient air):	5 max

NET Power

Purity:	99.5% mol. O_2 min
	0.3% mol Ar
	0.2% mol N_2
H_2O content:	1.0 ppm max
CO_2 content:	1.0 ppm max
HC as CH4 (number of times the content in ambient air):	5 max

Oxygen for the gasifier (IGCC case)

Supply pressure:	75-80 barg
Design pressure:	99 barg
Supply temperature:	15°C
Design temperature:	50°C
Purity:	97.0% mol. O_2 min

Oxygen for Sulphur plant

Supply pressure at IGCC BL:	5.0 barg
Design pressure:	8.0 barg
Supply temperature (min):	15°C
Design temperature:	50°C
Purity:	97% mol. O_2 min

3.4.6. *Electrical System*

The voltage levels foreseen inside the plant area are as follows:

	Voltage level (V)	*Electric Wire*	*Frequency (Hz)*	*Fault current duty (kA)*
Primary distribution	33000 ± 5%	3	50 ± 0.2%	31.5 kA
MV distribution and utilization	10000 ± 5% 6000 ± 5%	3 3	50 ± 0.2% 50 ± 0.2%	31.5 kA 25 kA
LV distribution and utilization	400/230V±5%	3+N	50 ± 0.2%	50 kA
Uninterruptible power supply	230 ± 1% (from UPS)	2	50 ± 0.2%	12.5 kA
DC control services	110 + 10%-15%	2	-	-
DC power services	220 + 10%-15%	2	-	-

3.5. Plant Life

The Plant is designed for 25 years life.

3.6. Codes and standards

The design is of the process and utility units are in general accordance with the main International and EU Standard Codes.

3.7. Software codes

For the design of the plant for the different study cases, three software codes have been mainly used:

- GS: POLIMI's proprietary software, conceived for the prediction of gas turbine performances
- PROMAX v3.2 (by Bryan Research & Engineering Inc.): Process Simulator used for sulphur removal unit simulator.
- Gate Cycle v6.1 (by General Electric): Simulator of Power Island used for HRSG simulation.
- Aspen HYSYS v7.3 (by AspenTech): Process Simulator used for oxy-fuel cycle simulation (excluding the gas turbine) and the CO_2 purification unit.
- Aspen Plus v7.3 (by AspenTech): Process Simulator used for the prediction of gas turbine performances for the NET Power cycle and the high pressure stage of the supercritical CES cycle.

3.7.1. *Gas turbine modelling with GS code*

In the evaluation of the thermal balances of oxy-turbine power plants, a crucial point is the evaluation of the performance of turbomachines, particularly the high temperature expander. In oxy-turbine power plants, compressor (if present) and expander operate on a working fluid that is mainly composed of CO_2 and H_2O (on a wide range of compositions in relation to the different cycles) that has significantly different thermophysical properties (e.g. specific heat, specific heats ratio, heat transfer coefficients) compared to air or combustion products, usually treated in conventional gas turbine engines. This large difference in fluid properties prevents that standard components from the gas turbines industry can be adapted to operate in oxy-turbine power plants, thus a substantial re-design of the turbomachines is required for the specific purpose. As a consequence, the evaluation of the gas turbine performance becomes a quite complicated task because all the effects related to the change of fluid have to be accounted for.

Politecnico di Milano has developed a calculation code, named "GS", conceived for the prediction of gas turbine performance at the design point. It performs the one-dimensional design of the turbine, aimed at establishing all the aerodynamic, thermodynamic, and geometric characteristics of each blade row. Proper correlations are then applied for the evaluation of the efficiency of the stages, while an accurate estimation of the blade cooling flow rates is considered by a model accounting for convective cooling in multi-passage internal channels with enhanced heat transfer surfaces, as well as film and Thermal Barrier Coating (TBC) cooling. Closed-loop cooling circuits can be simulated as well.

The calculation code can in principle be applied to the evaluation of oxy-fuel gas turbines since it is based on general correlations whose validity is independent from the working fluid properties. Two general basic assumptions are the following:

- The thermophysical properties of the working fluids are evaluated according to the ideal gas model (specific heat is calculated by NASA polynomials based on data of the JANAF tables [[1]]). This condition is closely verified for the usual operating range of gas turbine engines.

  Water and steam are treated as real fluid and their equation of state are taken from S.I. tables [[2]].

- Design parameters considered in the model are representative of the geometry employed in current "state of the art" gas turbine engines. Moreover, some critical coefficients have been calibrated to accurately predict the performance indexes of these machines.

Therefore, the current calculation model can be directly applied to the evaluation of the cooled expansion in components featuring conditions similar to those of the current commercial gas turbines (approximately turbine inlet temperature higher than

[1] Stull D.R. and Prophet H., Project Directors, JANAF Thermochemical Tables. 2nd Edition, U.S. National Bureau of Standards, Washington DC, USA, 1971.

[2] Schmidt E., Properties of Water and Steam in S.I. Units, Springer-Verlag, Berlin, Germany, 1982

900°C, inlet pressure below 60 bar). If the above mentioned conditions on temperatures and pressures are satisfied, the turbine calculation model can properly handle any working fluid composition.

Operational limits (e.g. TIT reduction) related to the change of the working fluid composition can be identified and margins deriving from a future technological improvement assessed.

The current cooled expansion model has been implemented in the GS code in 2002 [1]. Since that it has been extensively used for evaluation of gas turbine based plants in many research projects. Among them, six FP7 collaborative projects awarded research teams including Politecnico di Milano (Caesar, Cachet II, Demoys, Democlock, Ascent, Matesa).

Simulations of oxy-fuel, gas turbine based cycles has been carried out in the frame of a project funded by ENEA, the Italian governmental agency for energy and environment, about long-term coal gasification-based power with near-zero emissions [2,3]. In this project the GS model was applied to evaluate the performance of both CO_2-rich and H_2O-rich gas turbine based cycles. The current GS code was also used to perform a study on natural gas fired oxy-fuel combined cycle, on behalf of Edison S.p.A.

A previous version of the GS code [4] was applied for the simulation of oxy-fuel gas turbines in IGCC power plants [5].

3.7.2. Gas turbine modelling with Aspen plus

Direct evaluation by the GS code of cooled expansions is less accurate where the working fluid conditions are far away from those of the current commercial gas turbines (for instance that included in the NET Power's CO_2 cycle and the supercritical CES water cycle both having 300 bar, 1150 °C inlet conditions), for the following main reasons:

- The GS simulation code can handle only, ideal mixtures of ideal gas species and pure water, while it cannot properly model the behaviour of mixtures showing strong real gas effects such as those of the NET Power and supercritical CES turbine (mixtures of CO_2 and H_2O at 300 bar).
- Geometrical and design parameters of the stages (e.g., ratio between blade cooling flow and disk cooling flows, peripheral and axial velocity of each stage, coolant injection angle, ratio between area of cooling holes and total blade surface, Mach number at diffuser inlet, etc) considered in the model described in [1] have been derived from conventional gas turbine expanders. Thus, their direct application to turbine stages operating at significantly different conditions may be inappropriate.

- Critical calibration coefficients related to the performance of the cooling system (convective cooling, film cooling, Thermal Barrier Coating, etc) have been calibrated to accurately predict the performance of current gas turbine expanders.

Therefore, even if an equation of state for real gas mixture were implemented in GS, the expander model would not be suitable for evaluating the above listed cycles for the last two points.

As for that, in order to assess the performance (extracted power, turbine outlet temperature and mass flow rates of the cooling flows) of the NET Power cycle and of the supercritical CES cycle, a new model has been developed, that:

- It is capable of dealing with mixtures of real gases with proper equations of state,
- It is capable of accurately reproducing the results of GS for turbines expanding ideal gases,
- It depends on a few calibration parameters which do not include geometrical details of the stages and cooling systems.

A thorough review of the models available in the literature [6, 7, 8, 9] indicated that one proposed by El-Masri [7] is suitable for the study as it includes all the above listed requirements.

The model is a *continuous expansion* model, meaning that the turbine is treated as an expander whose walls continuously extract work, without distinguishing between stages, as well as stators and rotors. The expansion is divided into infinitesimal steps with extraction of mechanical power (dW), extraction of heat from the walls (dQ), and injection of coolant (dm). The basic assumption is that the specific power flux (W) per unit of wall surface area (Aw) is constant across the expansion and equal to the machine average.

In order to make the model suitable for the purposes of this study, the following modifications have been made:

- The number of expansion steps cannot be infinite as in the original version because, if the real gas behaviour is considered, the model equations cannot be analytically integrated. The expansion needs to be divided into a finite number of (small) steps, and the overall set of equations needs to be solved by means of a numerical algorithm.
- In order to improve the accuracy of the model and make it more similar to the one of GS [1], it is necessary to include a correlation which assesses the total pressure losses caused by the mixing of cooling flows and mainstream.

The model revised by Politecnico di Milano is schematically represented in Figure 2.

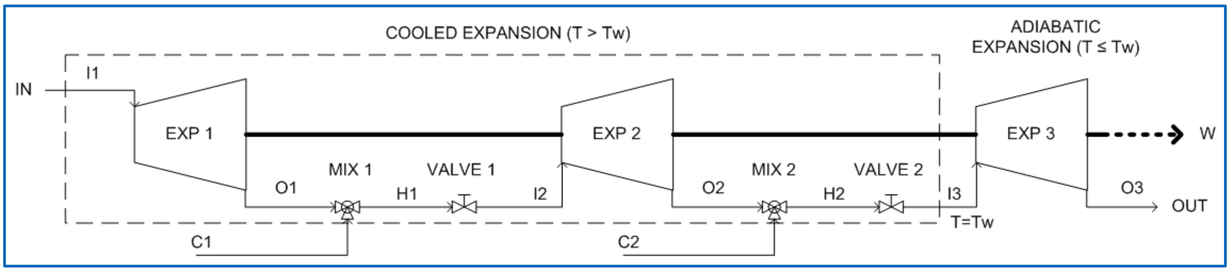


Figure 2. Model of the cooled turbine developed by Politecnico di Milano with N = 2

The expansion is divided into N+1 steps, where the first N steps are cooled, while the last one (N+1) is adiabatic. Each cooled step is made of an adiabatic expander, a mixer, mixing the main stream with the cooling flow, and a valve to introduce pressure losses. N is set to a reasonably large number (16) so as to accurately discretize the expansion. All the expanders have the same isentropic efficiency.

The model and the energy and mass balance equations of each piece of equipment of Figure 2 has been implemented in Aspen Plus. The fluid properties have been modelled with the Peng-Robinson equation of states which, according to [10], is suitable for the considered range of pressures and temperatures.

The turbine model implemented in Aspen Plus was calibrated on the basis of the GS model so as to have a fair comparison, not only between the two cycles modelled with the Aspen code, but also among all the oxy-fuel cycle options investigated in this study.

<u>*References*</u>

[1] Chiesa P., Macchi E., *A Thermodynamic Analysis of Different Options to Break 60% Electric Efficiency in Combined Cycle Power Plants*, ASME paper GT-2002-30663, Best Paper Award of the Electric Power Committee, Journal of Engineering for Gas Turbines and Power, Vol. 126, Issue 4, pp. 770-785, October 2004, DOI: 10.1115/1.1771684.

[2] Romano M.C., Lozza G.G., *Long-term coal gasification-based power plants with near-zero emissions. Part A: Zecomix cycle*, International Journal of Greenhouse Gas Control 4 (2010) 459–468, doi:10.1016/j.ijggc.2009.11.009.

[3] Romano M.C., Lozza G.G., *Long-term coal gasification-based power with near-zero emissions. Part B: Zecomag and oxy-fuel IGCC cycles*, International Journal of Greenhouse Gas Control 4 (2010) 469–477, doi:10.1016/j.ijggc. 2009.11.010.

[4] Consonni S., *Performance Prediction of Gas/Steam Cycles for Power Generation*, MAE Dept. Ph.D. Thesis n. 1983-T, Princeton University, Princeton, NJ, USA.

[5] Chiesa P., Lozza G., *CO_2 Emission Abatement in IGCC Power Plants by Semiclosed Cycles. Part A: with Oxygen-Blown Combustion*, ASME paper 98-GT-384 Journal of Engineering for Gas Turbines and Power, vol. 121, pp. 635-641, October 1999, DOI: 10.1115/1.2818519.

[6] El-Masri, M.A., 1988, *GASCAN - an Interactive Code for Thermal Analysis of Gas Turbine Systems*, Journal of Engineering for Gas Turbines and Power, 110, pp.201–209.

[7] El-Masri, M.A., 1986. *On Thermodynamics of Gas- Turbine Cycles : Part 2 — A Model for Expansion in Cooled Turbines*, Journal of Engineering for Gas Turbines and Power, 108, pp.151–159.

[8] He, F. et al., 2012. *Operation window and part-load performance study of a syngas fired gas turbine*. Applied Energy, 89(1), pp.133–141.

[9] Horlock, J.H., Watson, D.T. & Jones, T. V., 2001, *Limitations on Gas Turbine Performance Imposed by Large Turbine Cooling Flows. Journal of Engineering for Gas Turbines and Power*, 123(3), p.487.

[10] Poling, B.E., Prausnitz, J.M. & O'Connell, J.P., 2000. *The Properties of Gases and Liquids, 5th edition, McGraw-Hill Professional*.

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	OXYFUEL CYCLES REVIEW
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	(*)
CHECKED BY	:	N. FERRARI
APPROVED BY	:	L. MANCUSO

(*) M. Romano, P. Chiesa, E. Martelli, Politecnico di Milano

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

The aim of this analysis is the discussion of the oxyfuel cycles proposed in the literature for the conversion of natural gas and other gaseous fuels. For each cycle, the possible plant arrangements, the main operating parameters, the critical components requiring technological development and the expected performance are discussed. In particular, the following cycles are discussed:

- Semi-closed oxy-combustion combined cycle (SCOC-CC)
- MATIANT cycles
- NET Power cycle
- Graz cycle
- CES cycle
- AZEP cycle
- ZEITMOP cycle

The SCOC-CC, MATIANT and NET Power cycles use recycled CO_2 as moderator of the combustion temperature. Therefore, also the working fluid is a mixture mainly composed of CO_2.

The Graz and CES cycles use water as temperature moderator and thus steam is the main component of the working fluid mixture.

The AZEP and ZEITMOP cycles distinguish from the other cycles because they integrate a high temperature membrane for oxygen production in the power cycle. Since these membranes require a hot pressurized air stream from which O_2 is separated, an externally heated air cycle is also present as main power cycle (AZEP) or as side cycle of the principal CO_2 cycle (ZEITMOP).

2. Oxy-Turbine cycles

2.1. Semi-closed oxy-combustion combined cycle (SCOC-CC)

The SCOC-CC has the simplest arrangement among all the solutions considered.

It closely resembles a conventional combined cycle and hence it is conventionally used as benchmark cycle in most of the comparative analyses on natural gas fired oxyfuel cycles. It is fundamentally based on a combustion Joule-Brayton cycle (Figure 1). The compressor recycles part of the cooled CO_2 resulting from the fuel combustion. The amount of CO_2 recycled is set to achieve the desired combustor outlet temperature and to provide the cooling flows for turbine blades. The hot combustion products are expanded in the turbine and then cooled to nearly ambient temperature in a heat recovery steam generator (HRSG), feeding a bottoming steam cycle. Flue gases are then taken to nearly ambient temperature in a flue gas cooler, where most of the water in the combustion products is condensed. A portion of this stream is recycled to the compressor inlet of the Joule-Brayton cycle. The remainder represents the CO_2 rich stream sent to the purification and final compression units for long term storage.

The working fluid in the topping gas turbine cycle is mainly CO_2. The remaining gases are water, whose amount depends on the composition of the fuel and the conditions (temperature and pressure) of the stream recycled to the compressor inlet, excess O_2 necessary to achieve a complete combustion, Ar and N_2 contained in the not perfectly pure O_2 stream delivered by the ASU and to the N_2 in the fuel. The higher the presence of such non-condensable gases in the CO_2 stream, the more demanding the CO_2 purification process so as to achieve the purity specifications of the pipeline and the storage site.

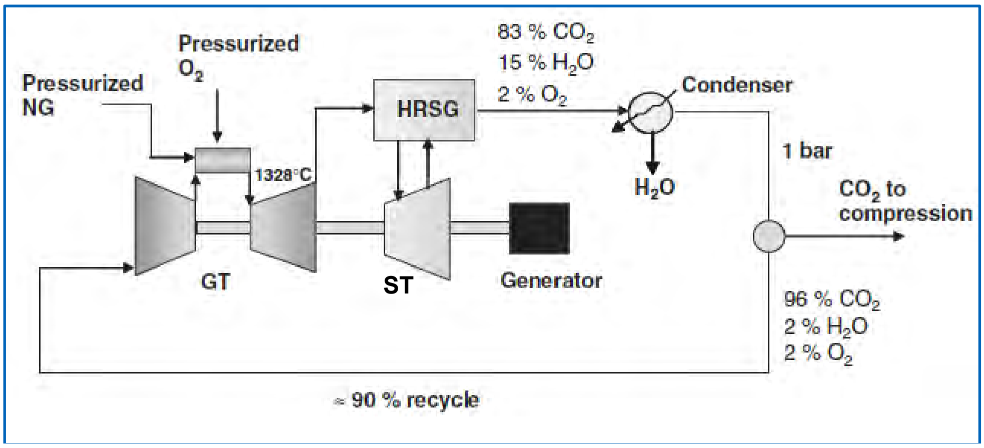


Figure 1. Schematic of the SCOC-CC (Kvamsdal et al., 2007).

The main design parameters of the SCOC-CC cycle partly differ from the ones usually adopted in conventional air-fired combined cycles. These design specs are listed below:

- **Pressure ratio**: as discussed by most of the authors assessing these cycles (Bolland and Mathieu, 1998; Chiesa and Lozza, 1999; Riethman et al., 2009; Romano and Lozza, 2010; Dahlquist et al., 2013) pressure ratio has to be optimized mainly on the basis of the cycle efficiency. In practice, for an assigned turbine inlet temperature (TIT), the combined cycle efficiency reaches a maximum for given temperature increase in the compressor and temperature drops in the expander almost independently of the properties of the working fluid. Since CO_2 has a larger molecular complexity and consequently a lower specific heat ratio than air, a higher pressure ratio is required to obtain the same temperature variation during compression and expansion. In particular, considering a TIT around 1400°C (representing the current technological limit for state of the art, heavy duty machines), an expansion ratio over 30 is required by CO_2 to obtain a turbine outlet temperature (TOT) around 580-620°C, which is the temperature range that guarantees an efficient heat recovery in the bottoming steam cycle which maximizes the combined cycle efficiency. Expansion ratios around 20 are instead required by air to obtain the same TOT.

- **Minimum cycle pressure**: being a semi-closed cycle, the minimum cycle pressure is not necessarily linked to the atmospheric pressure. A positive effect of pressurization on the component design is the reduction of the size of the turbomachines and of the HRSG. On the other hand, excessively high pressures become expensive because of the thickness of the high pressure parts, and the need of developing combustors capable of operating at very high pressure. Effects of pressurization on cycle efficiency have been assessed by Chiesa and Lozza (1999), Riethmann et al. (2009) Romano and Lozza (2010). In these studies, the effect of pressurization on cycle efficiency is reported to be relatively small (the maximum variation of about 0.5 percentage points, is reported Riethmann et al. (2009) when the minimum cycle pressure is increased to 10 bar). The most important effect is related to difficulties in turbine blade cooling when pressure is increased. On one hand, increasing the operating pressure reduces volume flow and hence surfaces to be cooled. On the other hand, it increases the heat transfer coefficients on both sides of the cooling channels and hence the heat flux through the blade walls. This brings about an increase in the temperature drop due to conduction along the wall thickness, which reduces the temperature rise available to the coolant in the inner blade channels. Therefore, larger coolant flows are required to remove the same thermal load. These negative effects might be reduced by cooling the CO_2 used for blade cooling as proposed by

Romano and Lozza (2010), following an approach today considered in Alstom GT24 and GT26 commercial gas turbines.

- **Flue gas cooling temperature**: Dahlquist et al (2013) assessed the effect of increased temperatures of flue gas recycle. In the analysis, pressure ratio has been varied to keep the target TOT of 620°C, which means that rather small variations were made (between 28.5 at and 32.6 for recycle temperatures of 80°C and 40°C respectively) as consequence of the modified working fluid composition (richer of steam at increased recycle temperatures). In general, it was demonstrated that higher recycle temperatures lead to higher cycle efficiencies, since lower heat is discharged to the environment. On the other hand, it results in a larger compressor size due to the lower recycled gas density, a lower specific work and more critical turbine blade cooling due to the higher temperature of the compressed CO_2. The authors claim that, considering performance, manufacturability, availability and working fluid composition, the preferred choice of the recirculated working fluid is around 60°C.
  Sundkvist et al (2014) also assessed a plant configuration with high temperature (98°C) flue gas recycle from the outlet of the HRSG. An intercooled compression has been considered in this case, with condensate separation and heat recovery by steam generation at the intercooler. In this case, an efficiency of 41.9%, about 6% points less than the reference SCOC-CC with cold recycle, is reported, most likely due to the inefficiencies associated to the intercooled compression.

In the literature, net efficiency of natural gas-fired SCOC-CC power plants with firing temperatures of 1300-1400°C and pressure ratio between 30 and 45 is reported to be in the 45-48% range (Bolland and Mathieu, 1998; Kvamsdal et al., 2007; Riethmann et al., 2009; Lozza et al., 2009; DOE/NETL, 2010; Dahlquist et al., 2013, Sundkvist et al., 2014).

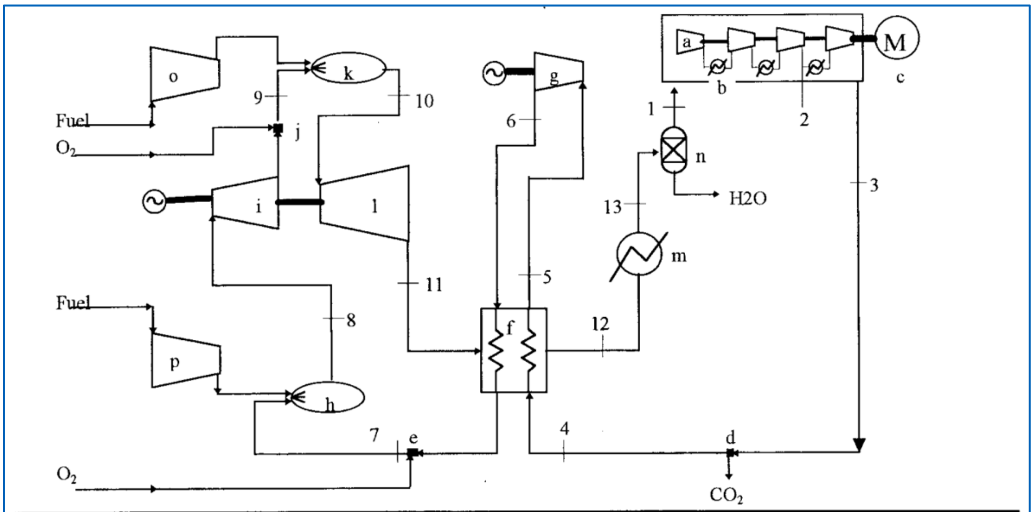
The SCOC-CC plant is the simplest oxyfuel cycle configuration proposed in the literature for gaseous fuels. However, due to the unusual working fluid in the gas turbine cycle with respect to conventional air-fired machines, turbomachines of the SCOC-CC plant require to be developed from scratch. In particular, the CO_2 turbine requires a re-design to optimize the blade geometry and the cooling channels on the properties of the CO_2-rich working fluid. For the development of the CO_2 gas turbine, no real technological barrier is foreseen, since it can be designed with the same criteria today used by gas turbine manufacturers. However, the establishment of a market for such machines is needed to justify the cost for the development of a completely new combustion turbine model, requiring important R&D efforts.

2.2. MATIANT cycles

The MATIANT cycles are named after the two designers Mathieu and Iantovski, who proposed the first cycle of this series (Iantovski and Mathieu, 1997). The original MATIANT cycle has been assessed by Mathieu and Nihart (1999a, 1999b), who investigated the effects of the main design parameters on cycle performance. A similar cycle, the COOPERATE cycle (CO_2 Prevented Emissions Recuperative Advanced turbine Energy), was earlier assessed by Yantovski et al. (1995) and Yantovski (1996). Such a cycle slightly differs from the MATIANT cycle in the regenerator, as mentioned in the following. However, the COOPERATE cycle was assessed with a lower level of accuracy and hence only the MATIANT cycle is considered in this section. Main results and considerations are however valid for both cycles.

The configuration of the MATIANT cycle and its representation on a temperature-entropy chart are shown in Figure 2 and Figure 3 respectively. The cycle features a triple expansion and uses a CO_2-based mixture as working fluid. CO_2 is compressed to very high pressure (300 bar) in a multi-stage intercooled compression section (component 'a-b' in Figure 2). It is then heated up to 700°C in a recuperator ('f') and expanded to around 40 bar in a high pressure uncooled turbine ('g'). After reheating in the regenerator, CO_2 is heated up to 1300°C by oxy-combustion in the first combustor ('h'). Hot CO_2 is then expanded in the MP cooled turbine ('i'), then reheated in a second oxy-fuel combustor ('k') and finally expanded in the LP cooled turbine to around atmospheric pressure. Hot CO_2 released from the last turbine represents the hot stream of the regenerator ('f'). Further cooling by discharging heat to the environment is performed in the final cooler ('m'), where also water originated from the combustion is condensed and separated.

The main difference in the COOPERATE cycle previously mentioned is related to the regenerator. In the COOPERATE cycle, the stream exiting the high pressure turbine is sent directly to the first combustor, without a second heating passage in the regenerator.



a : CO_2 compressor	b : intercooling	c : driving motor
d : splitter	e : mixer	f : recuperator
g : uncooled HP expander	h : cooled combustion chamber 1	i : MP expander with internal cooling
j : mixer	k : cooled combustion chamber 2	l : LP expander with internal cooling
m : cooler	n : water separator unit	o and p : fuel compressors

Figure 2. Configuration of the MATIANT cycle (Mathieu and Nihart, 1999b).

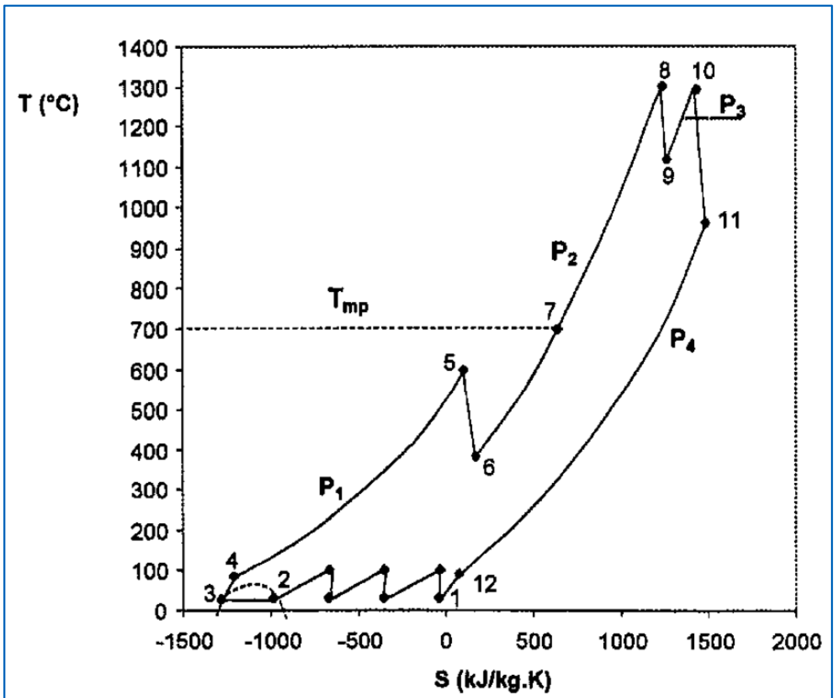


Figure 3. Representation of the MATIANT cycle on a temperature-entropy chart (Mathieu and Nihart, 1999b).

Mathieu and Nihart (1999b) assessed the effect of the main design parameters on the cycle performance. In their analysis, they did not calculate MP and LP expansions as cooled expansions, but simply applied an efficiency penalty of 2.5 percentage points to take the penalties of a cooled expansion into account. In their parametric analysis, they found that the CO_2 compressor isentropic efficiency and the MP turbine inlet pressure are the parameters that most affect the cycle efficiency. This is shown in Figure 4, where efficiency gains of about 5.5 percentage points can be achieved by increasing the compressor isentropic efficiency from 75% to 90% and variations of over 3 percentage points are observed by varying the MP turbine inlet pressure from 40 to 100 bar. In each of the cases reported in Figure 4, the MP turbine outlet pressure has been optimized. However, a limited dependence on this parameter is reported by the authors. A rather limited dependency is also reported for the maximum cycle pressure, i.e. at the inlet of the HP turbine. Mathieu and Nihart (1999b) report an efficiency drop of about one percentage point by reducing the maximum pressure from 300 to 150 bar.

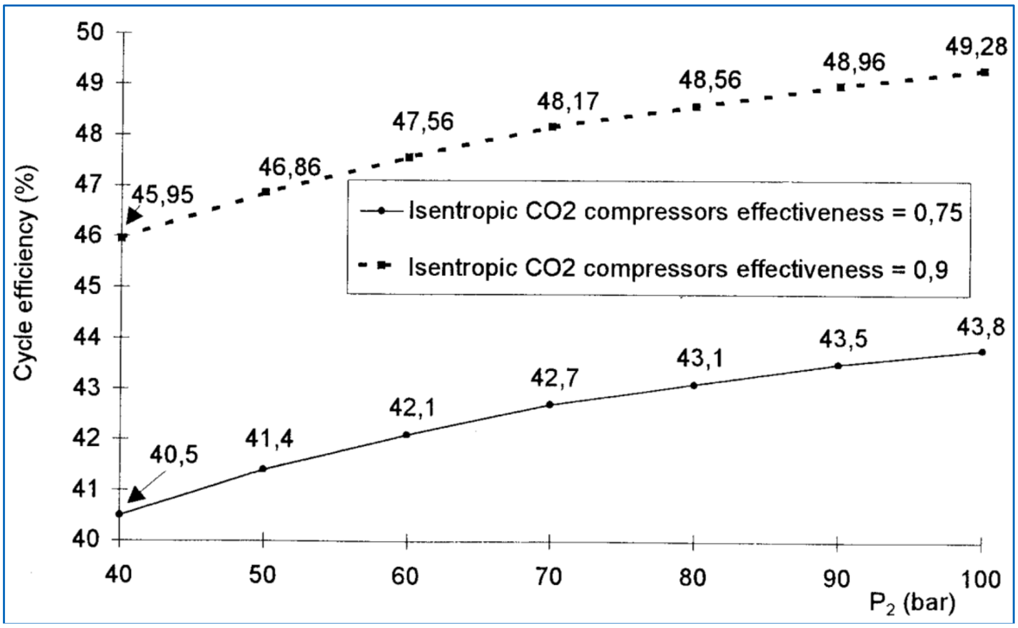


Figure 4. Effect of CO_2 compressor efficiency and MP turbine inlet pressure P_2 on cycle efficiency (Mathieu and Nihart, 1999b). A consumption of 280 kWh/t_{O2} for oxygen production at 5 bar is assumed by the authors.

The following critical components, unusual for today power plants can be identified for the MATIANT cycle:

- **CO_2 turbines**. For the high temperature MP and LP turbines, a technology similar to the conventional cooled gas turbine is needed. Similarly to the SCOC-CC turbine, a new design of the blades geometry and of the cooling path is needed to account for the different properties of the working fluid. The high pressure turbine would not require blades cooling if a suitable material is used and operates at pressures typical of conventional steam turbines. It is hence expected that a design somewhat derived from steam turbines can be adopted for this machine, with proper modifications to the stage geometry. For all the three turbines, re-design is hence required. As for the SCOC-CC turbine, no real technological barrier is foreseen, but a proper market justifying the high cost of development is needed.

- **Regenerator**. Large surface and high cost materials (e.g. high Nickel alloys) are expected for this component. The cold gas is at very high pressure (around 300 bar) and the maximum temperatures are notably high, with the hot gas entering at 900-1000°C and the cold gases heated up to 600-700°C. No similar heat exchanger is today employed in the industrial practice. Just a few relatively small size gas turbines (e.g. Solar Turbines Mercury 50) and micro GTs employ a regenerator, which however operate at much lower temperatures (of the order of 600-650°C for the hot gas inlet) and lower pressure differences (3-15 bar). Therefore, it is expected that this component, whose performance affects significantly the plant performance, will contribute significantly to the cost of the plant.
- **CO_2 condenser**. CO_2 compression and heat rejection are performed in a region where strong real gas effects occur and CO_2 may even condense (or is cooled at pressure over the critical one) before the final compression to the maximum cycle pressure. It is expected that consumption for CO_2 compression and final cycle efficiency are strongly influenced by the minimum CO_2 temperature, i.e. by the cooling medium used in the plant. Performance of the cycle should hence be evaluated in case cold cooling water is not available.

A second version of the cycle, named E-MATIANT cycle, was also proposed by Mathieu et al. (2000). The E-MATIANT cycle is an Ericsson-like cycle, including an intercooled compression, a regenerator which preheats the compressed CO_2 against the hot gas from the turbine, an oxyfuel combustor and an adiabatic expansion in a cooled CO_2 turbine (Figure 5). Minimum pressure is kept at a near-atmospheric value, while the maximum pressure can be optimized. The result of the optimization performed by Mathieu et al. (2000) is shown in Figure 6. Assuming a TIT of 1300°C and a minimum temperature difference of 20°C in the regenerator, an optimal efficiency around 47% was found, with a maximum cycle pressure of about 60 bar. It has to be noted that the CO_2 compressor isentropic efficiency assumed in this work is between 85% (first three stages) and 80% (last stage), which is an intermediate value with respect to the analysis performed for the reference MATIANT cycle (Figure 4).

As for the cycle complexity and the presence of critical components, the E-MATIANT cycle is significantly simplified with respect to the reference MATIANT cycle. In particular, expansion is greatly simplified, since it is performed in a single high temperature cooled turbine. Despite the regenerator is still expected to be a critical component from the techno-economic point of view, relevant simplifications are expected with respect to the MATIANT cycle, associated to the lower maximum cycle pressure and to the lower temperatures of the streams on the hot side.

Compared to the original MATIANT cycle, where the working fluid is condensed toward the end of the intercooled compression, the E-MATIANT cycle always operates in the single phase region. The former configuration certainly reduces the compression work but in the meantime it entails a series of serious issues.

In relation to the temperature of the cooling medium used in the plant, temperature at the end of the last intercooling stage (point 3 in Figure 3) may alternatively be either below or above the critical point. This has a decisive influence on the heat transfer pattern of the condenser, making the design and operation of this component difficult. The design and operation of the pump performing the last compression stage are difficult too, because of the variation of fluid density and compressibility induced by the variation of the temperature.

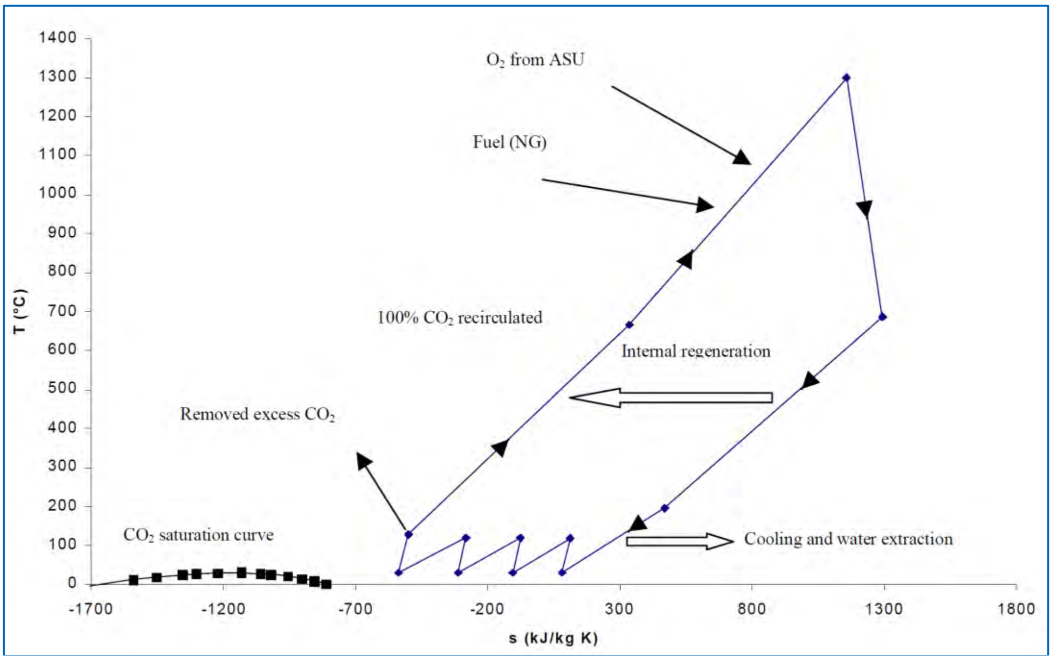


Figure 5. Temperature-entropy diagram of the E-MATIANT cycle (Mathieu et al., 2000).

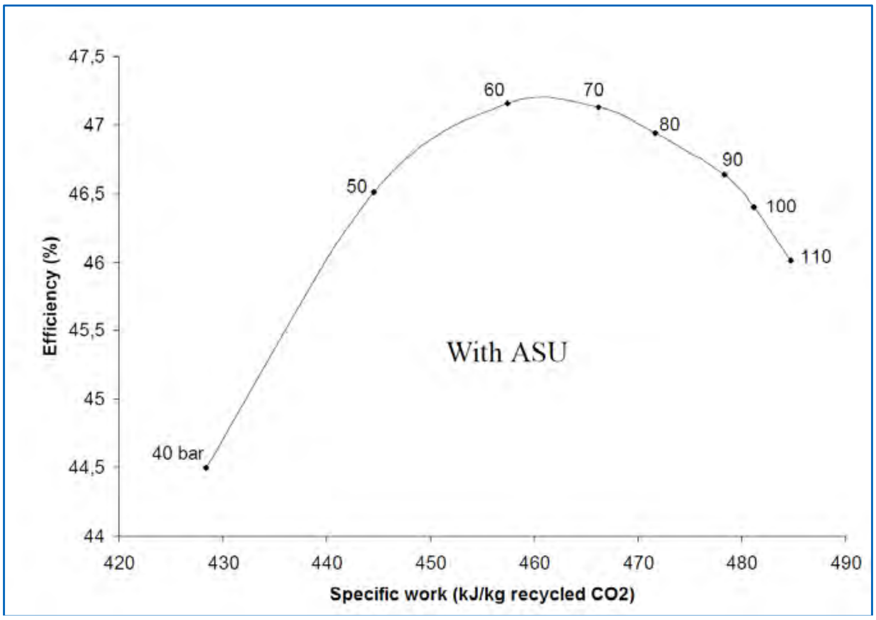


Figure 6. Effect of the maximum pressure of the E-MATIANT cycle on efficiency and specific work (Mathieu et al., 2000).

A further version of the MATIANT cycle, the CC-MATIANT cycle, was finally presented by Houyou et al. (2000). In this case, an adiabatic compression is assumed and heat of the stream at turbine outlet is recovered by a bottoming steam cycle (Figure 7). The difference between this cycle and the SCOC-CC is related to the partial regeneration. As a matter of fact, the high temperature heat in the turbine exhaust is used to preheat the compressed CO_2 before being cooled in the heat recovery steam generator. Houyou et al. (2000) compared the three versions of the MATIANT cycle, reporting the highest efficiency for the CC-MATIANT cycle (49%), followed by the E-MATIANT cycle (47%). The lowest efficiency was instead reported for the reference MATIANT cycle, presumably calculated with consistent assumptions, for which an efficiency of 44% is reported.

Despite the CC-MATIANT cycle was reported as the most efficient of the series, it is highly doubtful that the complication introduced by the high temperature regenerator is justified on the economic side, with respect to the similar and much simpler SCOC-CC plant. For this reason, and because of the lack of additional studies confirming its performance, the CC-MATIANT cycle was not considered further in this study.

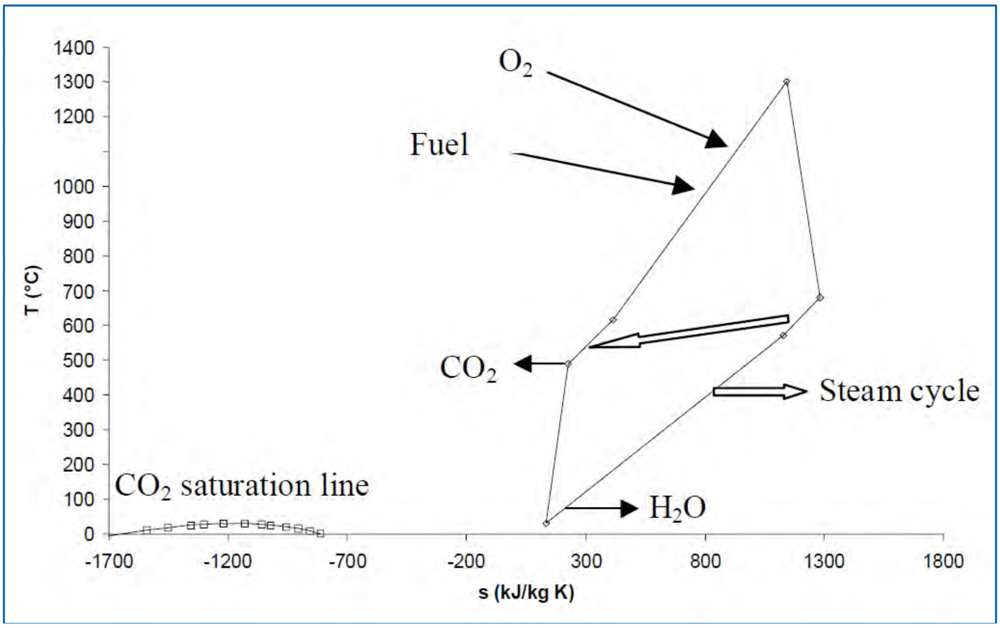


Figure 7. Temperature-entropy diagram of the CC-MATIANT cycle (Houyou et al., 2000).

2.3. NET Power cycle

The NET Power cycle is under development by a consortium constituted by NET Power LLC, CB&I, Exelon Corporation and the Toshiba Corporation. Also named “Allam” cycle, after Rodney Allam, who patented the concept (Allam et al., 2010), it is a regenerative supercritical CO_2 cycle.

A simplified cycle layout is shown in Figure 8. In Figure 9, the same cycle is represented on a pressure-enthalpy diagram. CO_2 compression is performed by a 1-2-stage intercooled compressor (up to around 80 bar) and a pump (up to 200-400 bar). CO_2 is then preheated up to 700-750°C in a regenerator (named ‘economizer’ heat exchanger) by the hot CO_2 from the turbine. Low temperature heat available from intercooling of the ASU compressors is also used in this heat exchanger for CO_2 preheating. This low temperature heat integration with the ASU (or other available sources) is important to compensate for the imbalance of heat capacity between the hot and the cold CO_2 streams in the regenerator. A portion of the preheated CO_2 is then sent to the oxy-combustor, where hot gases at temperature higher than 1100°C are produced. A smaller fraction provides the cooling flow to the gas turbine blade (not shown in the schematic diagram in Figure 8). The remaining flow is combined (not shown in the schematic diagram in Figure 8) with high pressure oxygen from the ASU before being pre-heated, providing the oxidant stream to the combustor.

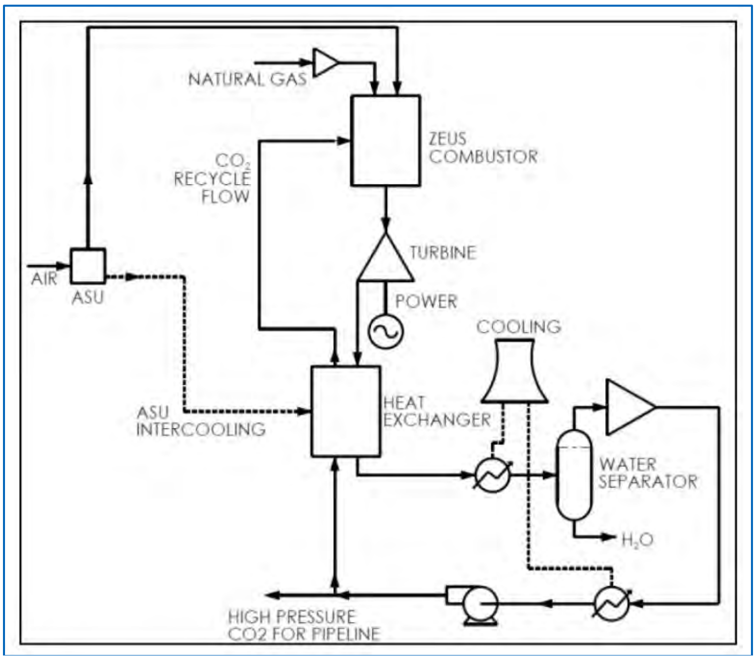


Figure 8. Simplified scheme of the NET Power cycle (Allam et al. 2013b, Allam et al. 2014).

The flow rate of CO_2 recycled to the combustor as temperature moderator is determined to achieve the target turbine inlet temperature. This results in a working fluid composed by more than 90% of CO_2. A cooled turbine expands the gases to below the CO_2 critical pressure of 73 bar (e.g. to 20-30 bar) and the expanded CO_2 is then cooled in the regenerator.

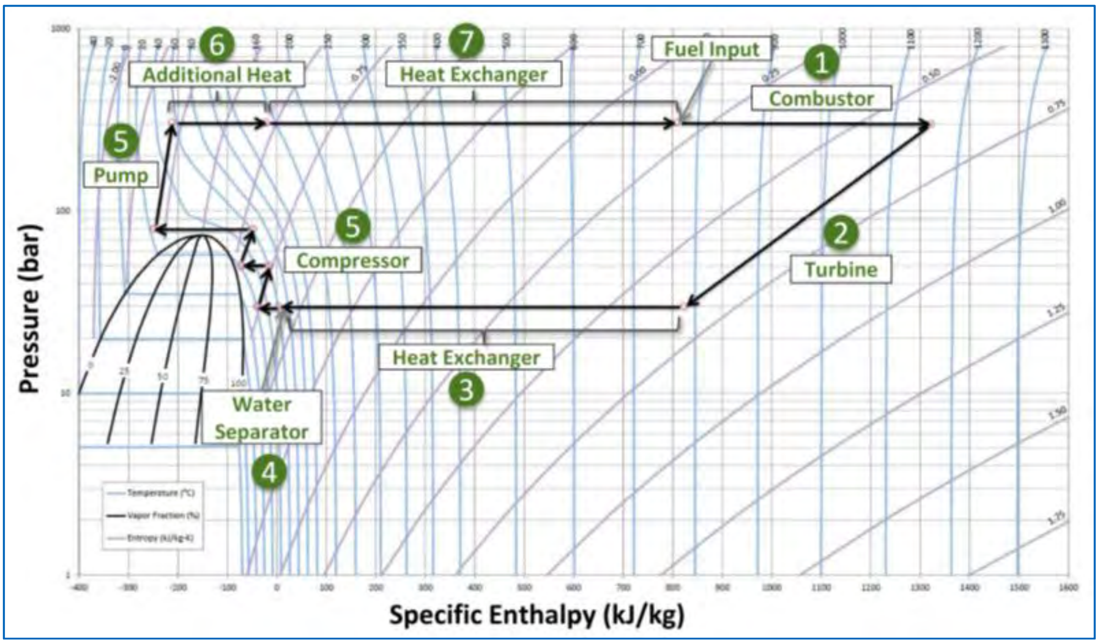


Figure 9. Representation of the NET Power cycle on a pressure-enthalpy chart (Allam et al., 2013b).

With such a plant configuration, an efficiency of 53.9% is reported (Allam et al., 2013a) for a turbine inlet and outlet conditions of 300 bar/1189°C and 50 bar/885°C respectively and for a high pressure CO_2 combustor inlet temperature of 860°C.

The most updated publication (Allam et al., 2014) claims an efficiency of around 59% for a turbine inlet and outlet conditions of 300 bar/1150°C and 30 bar/775°C respectively and for a high pressure CO_2 combustor inlet temperature of 750°C. The 59% efficiency figure is claimed to be achieved using NET Power proprietary know-how, although 55% is the figure estimated based on information that NET Power disclose publicly.

A more complex configuration featuring the reheating of the working fluid in a second combustor has been also proposed (Figure 10). In this plant, the intermediate pressure CO_2 released from the high pressure turbine (component 3) is reheated in a low pressure combustor (2) and then expanded to nearly atmospheric pressure in a low pressure turbine (4). High temperature and low pressure CO_2 is then cooled in regenerators (14, 13) by preheating the high pressure CO_2 recycle. Low temperature heat is also provided in a parallel heat exchanger (15) by cooling air compressed adiabatically in compressor 16, then delivered to the ASU (20). Oxygen is delivered at two pressure levels. The high pressure oxygen (stream 27) is compressed at the maximum cycle pressure and mixed with the cold CO_2 before preheating and use in the high pressure combustor (1). The intermediate pressure O_2 (34) is also preheated and used in the low pressure combustor. Similarly, methane is used at two different pressure levels (high pressure stream 44 and intermediate pressure stream 40) to match the pressure of the two combustors. Intermediate pressure oxygen and methane are also preheated before combustion in heat exchanger 22 against a portion of the low pressure CO_2. Outstanding efficiencies between 58.6 and 58.9% are reported for this plant when fired with methane, for turbine inlet temperatures of 1150°C and 1350°C respectively, and recycled CO_2 preheating temperature of 735°C. This efficiency value includes the consumption for O_2 production (Allam et al. 2013a), while exporting the net CO_2 product at atmospheric pressure.

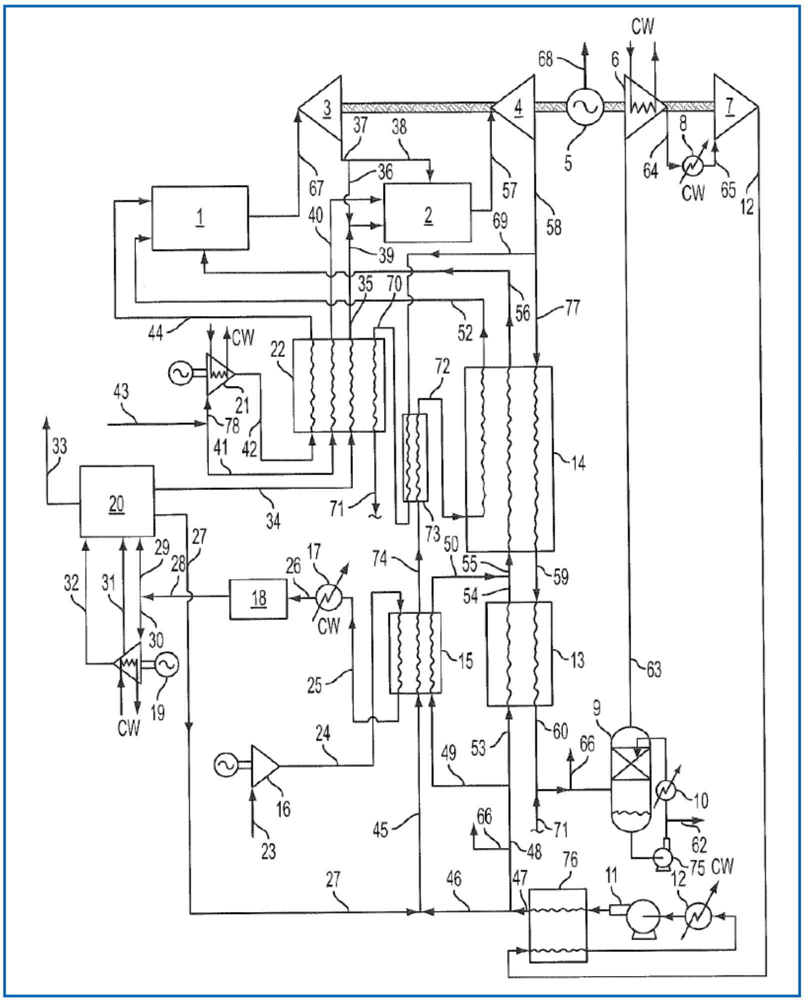


Figure 10. Configuration of the double combustor NET Power cycle (Allam et al., 2013a).

In addition to the extremely high efficiencies, a very compact design is expected for the cycle, with footprint of 1/3 of a conventional combined cycle (Allam et al. 2013b). This also leads to relatively low expected specific investment costs of 800-1200 $/kW (Allam et al., 2014).

In the NET Power cycle, the following critical components require discussion:

- **The turbine**. It operates at very high inlet pressures (200-400 bar), typical of current supercritical steam cycles and high inlet temperatures (higher than 1100°C), similar to E-class commercial gas turbines. Turbine blades hence require cooling and a new design which cannot completely rely on the current steam turbines and gas turbine technology. Toshiba is the partner in the consortium in charge of the development of the turbine.
  In case of the reheated configuration, the second turbine operates at inlet and outlet temperatures and pressures comparable with the state-of-the-art gas turbines. Similar considerations as for the SCOC-CC turbine can be made for this machine.
- **Combustor**. It combines high combustion temperature and pressure, which are conditions far from the common practice. Successful ignition and initial operation of a test combustor is reported by Toshiba and NET Power (2013a).
- **Economizer heat exchanger**. It may be a compact multi-channel plate-fin design or printed circuit in Ni-alloy (e.g. Alloy617) (Allam et al. 2013b). The cost of the economizer is expected to strongly impact on the final plant cost, due to its large size and the high maximum temperatures, requiring high cost materials. NET Power proposes a design with rather small temperature difference at the hot end, where the hot gas enters at 700-885°C and the cold CO_2 is heated to 675-860°C. It is expected that heat exchanger design and minimum temperature approach needs to be properly optimized due to the important impact on plant cost and efficiency.
- **Condenser/heat rejection**. NET Power claims that the cycle is suitable for hot climates and dry cooling (NET Power, 2013b), though it is believed that reduction of efficiency and power output will result from the increase of the condensing/heat rejection temperature, as affecting the power demand of the recycle stream recompression. However, it has to be noted that, the impact of hot ambient conditions is less significant in the semi-closed oxy-fuel cycle as the gas turbine performance are not affected by ambient temperature, which only affects the performance of the equipment impacted by the achievable heat rejection value (i.e. cooling medium temperature level) such as the steam turbine and the intercooled compressors.

2.4. Graz cycle

The original idea of the Graz cycle was presented by Herbert Jericha (1985), who then developed the concept along with his research group at the Graz University of Technology. It is a medium pressure and high temperature oxy-fuel Joule-Brayton cycle, using steam as combustion temperature moderator.

Figure 11 shows the configuration of the S-Graz cycle (i.e. "high steam content" Graz cycle). This configuration was first proposed at the ASME Turbo Expo conference in 2004 (Sanz et al., 2005) and distinguishes from the previous version of the Graz cycle because steam is used as temperature moderator and the working fluid is hence mainly constituted by steam. In previous versions of the Graz cycle, CO_2 recycled after water condensation was used as temperature moderator. However, such a configuration demonstrated to be less efficient, did not provide particular technological advantages and was hence abandoned since then and not discussed in this work.

With reference to the S-Graz cycle shown in Figure 11, fuel is burned in a medium pressure, steam moderated oxy-combustor, where hot gases at around 40 bar and 1400°C are produced. Hot gases are then expanded to atmospheric pressure in the high temperature turbine (HTT) and are cooled in a HRSG to around 180°C. In the HRSG, high pressure steam (180 bar, 550°C) is produced, then expanded in a high pressure turbine (HPT) down to around 40 bar. Steam exiting the HPT is used as temperature moderator in the combustor and as cooling flow of the HTT blades. Working fluid from the HRSG is partly recycled to the combustor by an intercooled compressor (C1/C2) to provide additional steam for combustion temperature control. This recycle is needed since the steam raised in the HRSG and expanded in the HPT is not enough to control the combustion temperature. The cooled fluid from the HRSG is then expanded in a low pressure steam turbine (LPT) down to vacuum pressure. Steam is then condensed by means of cooling water and liquid water is pumped back to the HRSG. CO_2 saturated with steam is extracted from the condenser and compressed for final storage.

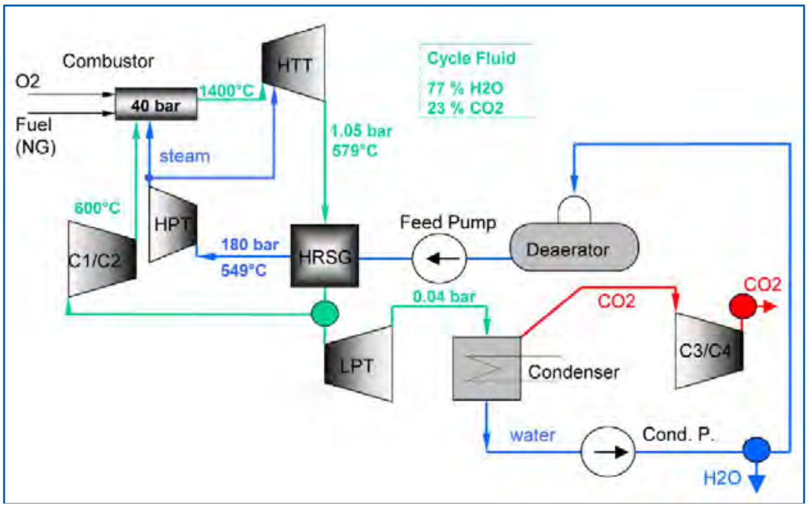


Figure 11. Configuration of the S-Graz cycle (Jericha et al., 2008a)

A "modified S-Graz" cycle was proposed at the ASME Turbo Expo 2006 conference (Jericha et al., 2008a) and considered since then for the following studies (Figure 12).

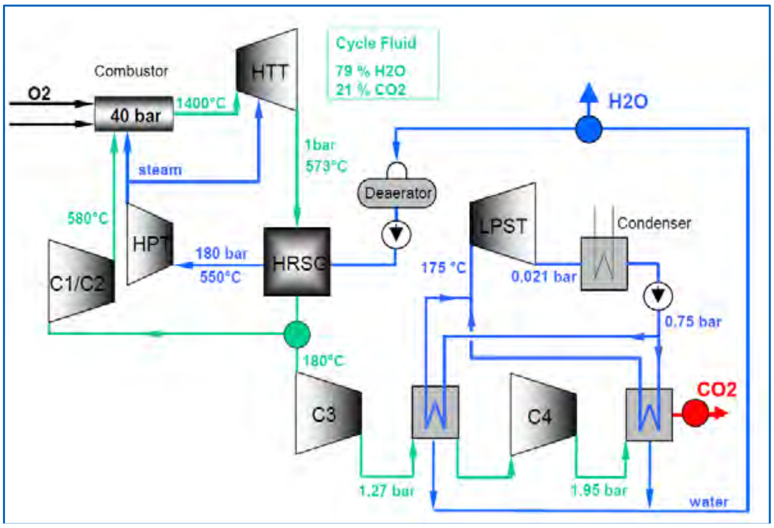


Figure 12. Configuration of the modified S-Graz cycle (Jericha et al., 2008a)

In this cycle, the low pressure section was modified to solve the problem of the very large condenser size. As a matter of fact, the presence of CO_2 leads to low heat transfer coefficients in the condenser due to additional mass transfer resistance. Being also at vacuum pressures, very large heat exchange area and cross-sections are expected. In the new configuration, working fluid from the HRSG is slightly compressed by two intercooled compressors (C3 and C4) up to 1.95 bar. In the intercooler and the after cooler, most of the water condenses at around 100°C, allowing producing VLP superheated steam at 0.75 bar, to be expanded in a LP steam turbine. The temperature-heat diagram of the LP condensers/evaporators is shown in Figure 13.

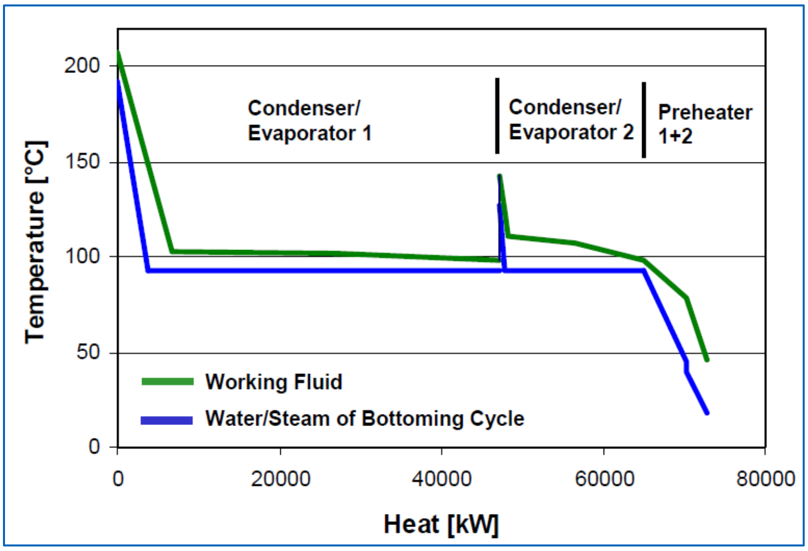


Figure 13. Temperature-heat diagram of the condenser/evaporator of the modified S-Graz cycle (Jericha et al., 2008a)

Diverse results of process simulations using natural gas or syngas as fuel have been reported by the research group of the Graz University of Technology between 2002 and 2011. Results differ due to progressive update of the calculation assumptions, due to modification in the fuel composition and due to the modifications introduced in the cycle configuration.

At the ASME Turbo Expo 2007 conference, a comparison between a 400 MW_e modified S-Graz cycle and a SCOC-CC plant was presented (Sanz et al., 2007). For a combustion temperature of 1400°C, a net efficiency of 53.09% was predicted for the Graz cycle (including a consumption of 250 kWh/t_{O2} for oxygen production), to be compared with 49.75% of the SCOC-CC. Authors claim that the higher efficiency of the Graz cycle is mainly due to the lower requirement of cooling flows, equal to 13.7% of the total HTT inlet flow, compared to 30.5% of the turbine inlet flow in the SCOC-CC case. The different cooling flow rates are due to the different number of turbine stages that require cooling, to the better heat transfer properties of H_2O with respect to CO_2 and to the lower temperature of the H_2O used for cooling in the Graz cycle.

At ASME Turbo Expo 2008 conference (Jericha et al., 2008b), a 600 MW_e Graz cycle was presented with firing temperature increased to 1500°C. An efficiency improvement by 1 percentage point was calculated with respect to the 1400°C case, resulting in a net efficiency of 54.14%. The detailed flowsheet of this case with the main thermodynamic properties of the streams is reported in Figure 14.

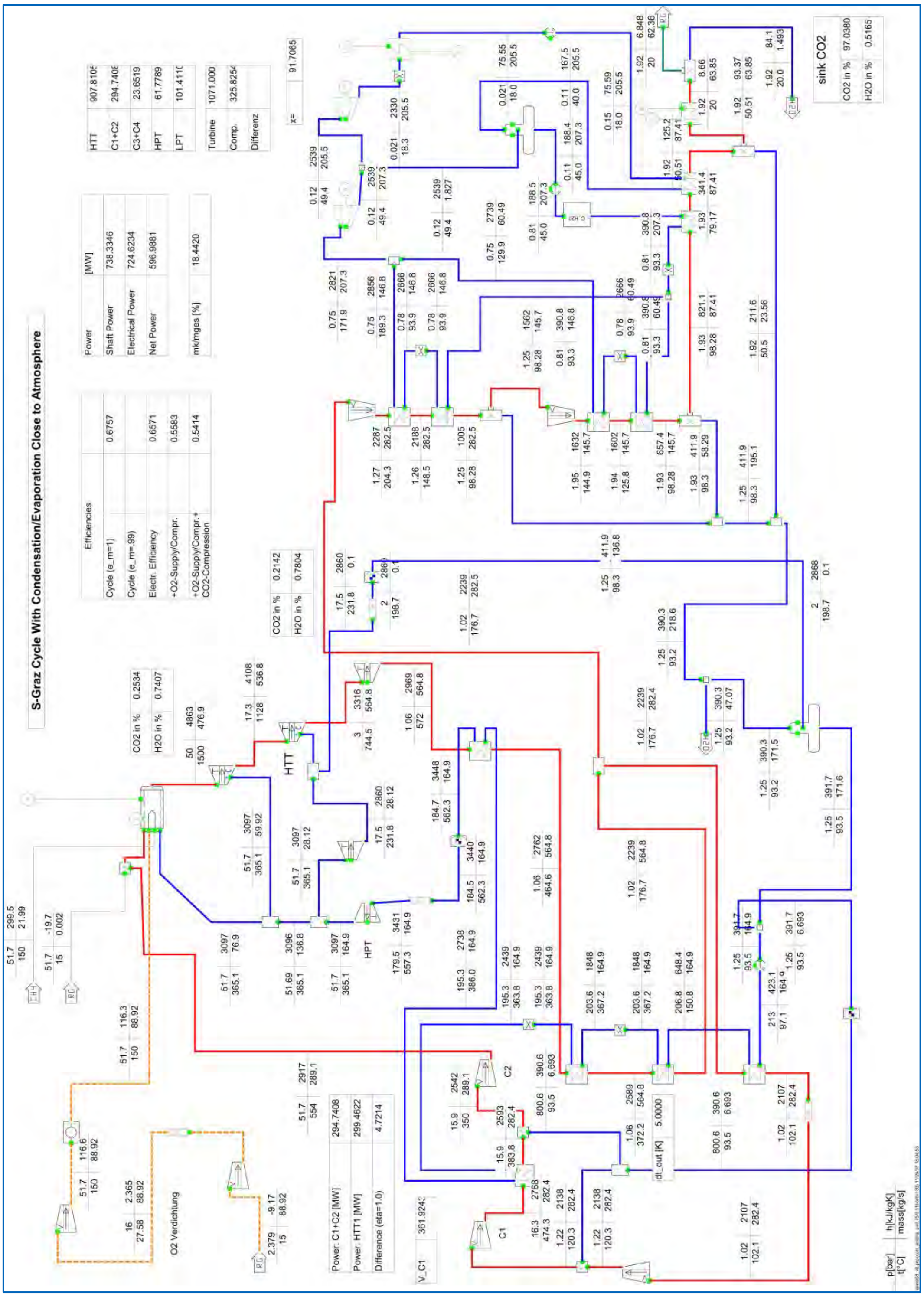


Figure 14. Pressure, temperature, enthalpy and mass flow rate of the streams of the 600MW$_e$ Graz cycle proposed by Jericha et al. (2008)

The performance of the Graz cycle has been also assessed by Kvamsdal et al. (2007), who predicted a net efficiency of 48.6%, to be compared with 47.0% of the benchmark SCOC-CC plant.

Particular attention has been given by the Graz cycle developers to the design of the turbomachines, for which a preliminary sizing has been performed. The layout of the turbomachines of the 400 MW_e cycle is reported in Figure 15 (Sanz et al., 2007).

The compression shaft consists of the cycle compressors C1 and C2, which are driven by the first part of the high temperature turbine HTT (the compressor turbine HTTC). This shaft runs free at its optimal speed of 8500 RPM. The second part of the HTT (the power turbine HTTP) delivers the main output to the generator at 3000 RPM. A further elongation of the shaft is done by coupling the four-flow LPST at the opposite side of the generator. The HPT can be coupled to the far end of the LPST or can drive a separate generator. In (Sanz et al., 2007), the basic geometrical data of the turbomachines is also reported. The different rotational speed of the HTTC and the HTTP allows designing turbine stages with favorable size, in line with the practice of current machines. Mean diameters of 1.06-1.14 m and blade heights of 0.10-0.177 m for the HTTC and mean diameters of 1.77-3.11 m and blade heights of 0.25-0.65 m for the HTTP are for example reported.

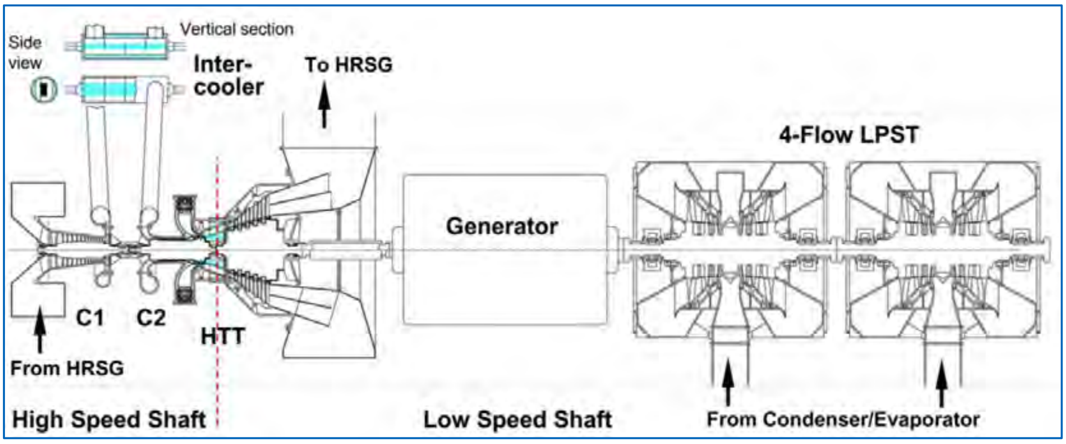


Figure 15. Layout of the turbomachines of the 400 MW_e Graz cycle (Sanz et al., 2007)

The following considerations can be made on the technological hurdles associated with the development of the main Graz cycle components:

- **Turbomachines**. Especially the high temperature turbine needs a completely new design, being substantially a steam turbine operating at conditions typical of a gas turbine. As for most of the previous high temperature cooled turbines, no real technological barrier is foreseen for its development, but a market driver is needed to justify the high cost of development. The same is true for the steam compressors, with lower uncertainties being non-cooled machines. No issue is foreseen for the HP and the LP turbines, for which current commercial steam turbines can be used.

An important issue to be understood for the high temperature turbine is related to corrosion phenomena in presence of the H_2O-rich high temperature stream. Increased corrosion with respect to conventional gas compositions have been observed in material tests for the high temperature turbine of the CES cycle discussed below (Anderson et al., 2010). The operating conditions and gas composition in the CES cycle turbine are not far from the ones of the Graz cycle, so that similar conclusions on the corrosion side are expected.

- **Combustor**. Operating pressure and temperature lie in the range of conventional GT combustors, so that no real technological challenge is foreseen. A dedicated design is of course needed due to the unusual nature of the combustion gases.
- **Condenser/Evaporator**. Condensing and evaporation pressure has to be optimized on a techno-economic basis. In particular, evaporation at sub-atmospheric pressure can lead to economically sub-optimal designs due to the large volume flows of VLP steam. However, no technological barrier is foreseen for these components.

2.5. CES cycle

This cycle was proposed by Clean Energy Systems, Inc. since early 2000s. Also named “water cycle” by some researchers, it uses water in vapor and liquid phases as combustion temperature moderator. This cycle is conceptually similar to the hydrogen-fuelled oxycombustion cycle assessed by Westinghouse researchers in the ‘90s (Bannister et al., 1999), who proposed this high efficiency internal combustion steam cycle, assessed the cycle performance and performed preliminary dimensioning of the turbines.

Different versions, with multiple variations of the original CES cycle, have been proposed through the years. The configuration proposed in the most recent works as candidate for large scale commercial applications is shown in Figure 16 (Anderson et al., 2008).

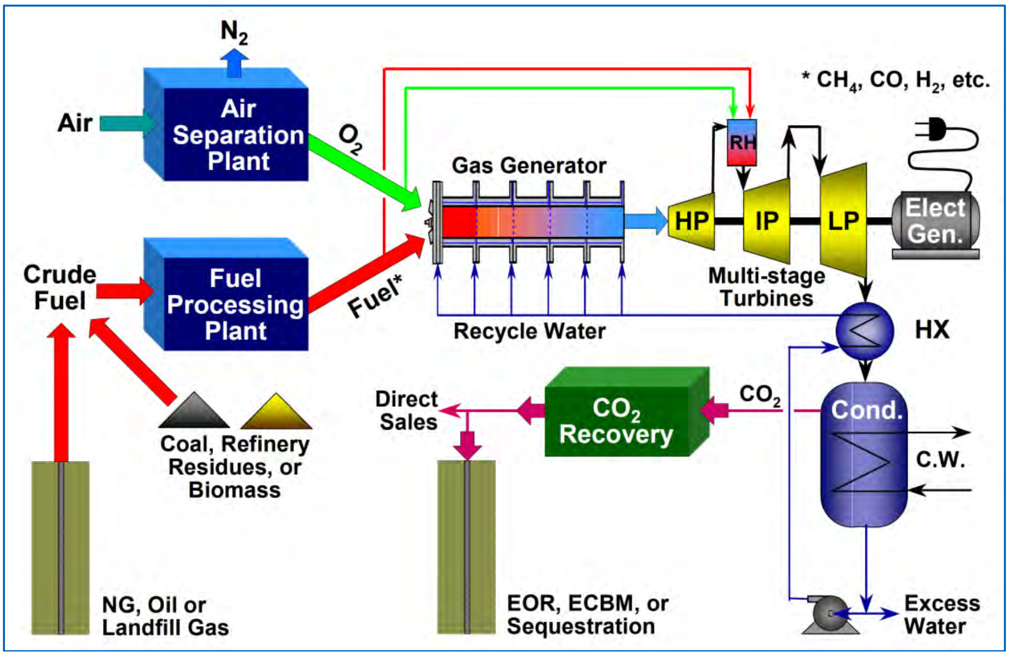


Figure 16. Third generation CES cycle for large scale applications (Anderson et al., 2008).

A high pressure oxyfuel combustor (the “Gas generator”) is used for conversion of most of the fuel, utilizing preheated liquid water as temperature moderator. This combustor, whose design is derived by rocket combustors, operates at pressures around 80-100 bar and with an outlet temperature of about 760°C. Hot gas produced in the gas generator is expanded in an uncooled HP steam turbine. Working fluid discharged at around 40 bar, it is reheated to very high temperature (around 1760°C) by a supplementary oxyfuel combustion, and expanded in an IP turbine. Part of the working fluid from the HP turbine outlet bypasses the reheater and is used for IP turbine blades cooling. A LP turbine finally expands the working fluid from IP turbine outlet (1 bar and 760°C) to the condensing pressure. Heat at LP turbine outlet is recovered by preheating the high pressure water recycled to the gas generator. CO_2 is extracted with water vapor from the vacuum condenser and compressed to the final storage pressure.

The development of the CES cycle is planned to be performed through three turbine generations. In the first generation plant, a commercial GE J79 gas turbine is used as high temperature turbine, with inlet temperatures of 760-927°C, so that turbine cooling can be avoided. The resulting 60-70 MWe plant would have an efficiency of 30-34%. The second generation plant will use the SGT-900 gas turbine, with inlet temperature of 1180°C. In this case, the 100-200 MW_e plant would have a 40-45% electric efficiency. For the third generation cycle would then be based on a turbine with a dedicated design and inlet temperature of 1760°C. A net efficiency in the 50% range is foreseen for the resulting 400 MW_e plant.

The performance of the CES cycle has been assessed by other authors. With HPT and HTT inlet temperatures of 900°C and 1328°C respectively and a condensing pressure of 0.045 bar, Kvamsdal et al. (2007) predicted an efficiency of 44.6%, vs. 47% of the benchmark SCOC-CC.

More recently, in a report from DOE/NETL (2010), an efficiency of 49.6% is reported, vs. 46.2% of a SCOC-CC plant. It is not easy to justify such a different result with respect to the work by Kvamsdal et al. (2007). Differences might be due to the different TIT (about 80 and 100°C higher for the HP and the IP turbine respectively, in the DOE/NETL work), different cooled turbine models and in general different cycle parameters. The DOE/NETL plant configuration and the main flows properties are shown in Figure 17 and Table 1. High pressure turbine inlet conditions of 982°C and 150 bar, high temperature turbine inlet conditions of 1426°C and 27 bar, condensing pressure and temperature of 0.1 bar and 38°C and moderating water at gas generator inlet at 170 bar and 353°C have been assumed in this study. On the other hand, the highest investment cost and the highest cost of electricity have been estimated for the CES cycle, compared to SCOC-CC and to other competitive pre-combustion and post-combustion capture plants.

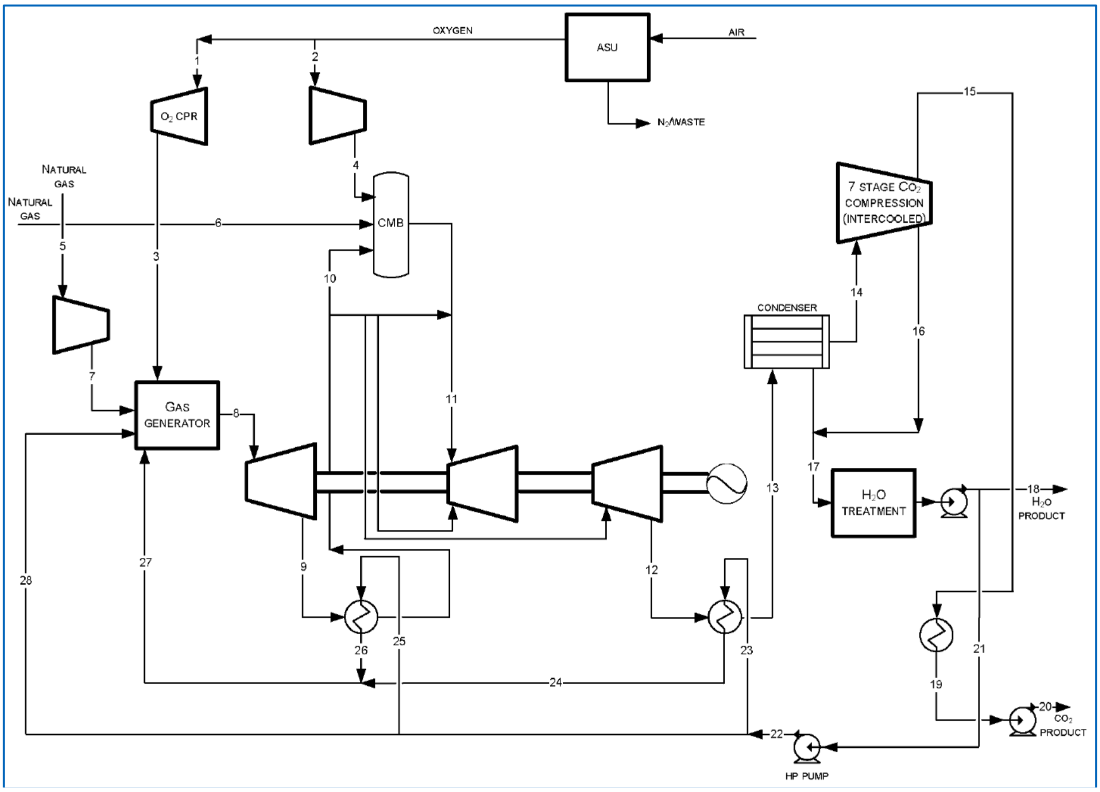


Figure 17. Configuration of the CES cycle proposed by DOE/NETL (2010)

Table 1. Properties of the main flows of the CES cycle reported by DOE/NETL (2010).

	1	2	3	4	5	6	7	8	9	10	11	12	13	14
V-L Mole Fraction														
Ar	0.0020	0.0020	0.0020	0.0020	0.0000	0.0000	0.0000	0.0002	0.0002	0.0002	0.0004	0.0004	0.0004	0.0019
CH_4	0.0000	0.0000	0.0000	0.0000	0.9310	0.9310	0.9310	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
C_2H_6	0.0000	0.0000	0.0000	0.0000	0.0320	0.0320	0.0320	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
C_3H_8	0.0000	0.0000	0.0000	0.0000	0.0070	0.0070	0.0070	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
C_4H_{10}	0.0000	0.0000	0.0000	0.0000	0.0040	0.0040	0.0040	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
CO	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
CO_2	0.0000	0.0000	0.0000	0.0000	0.0100	0.0100	0.0100	0.0514	0.0514	0.0514	0.1075	0.1048	0.1048	0.4776
H_2O	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.9463	0.9463	0.9463	0.8876	0.8905	0.8905	0.5010
N_2	0.0030	0.0030	0.0030	0.0030	0.0160	0.0160	0.0160	0.0011	0.0011	0.0011	0.0023	0.0022	0.0022	0.0102
O_2	0.9950	0.9950	0.9950	0.9950	0.0000	0.0000	0.0000	0.0010	0.0010	0.0010	0.0021	0.0020	0.0020	0.0093
Total	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
V-L Flowrate (kg_{mol}/hr)	2,961	4,463	2,961	4,463	1,433	2,160	1,433	29,045	29,045	26,431	33,988	35,731	35,731	7,841
V-L Flowrate (kg/hr)	94,765	142,815	94,765	142,815	24,839	37,433	24,839	562,947	562,947	512,282	709,419	743,196	743,196	240,744
Solids Flowrate (kg/hr)	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Temperature (°C)	32	32	129	149	38	38	117	982	669	316	1,426	422	59	38
Pressure (MPa, abs)	0.21	0.21	17.24	2.90	3.10	3.10	17.24	14.82	2.76	2.69	2.62	0.01	0.01	0.01
Enthalpy (kJ/kg)[A]	28.97	28.97	103.20	135.37	48.09	48.09	168.64	-6,445,108.64	3,461.26	2,722.28	4,801.97	2,661.09	2,026.47	778.47
Density (kg/m³)	2.6	2.6	157.0	26.2	21.8	21.8	92.5	27.2	6.9	11.3	3.9	0.1	0.1	0.2
V-L Molecular Weight	32.003	32.003	32.003	32.003	17.327	17.327	17.327	19.382	19.382	19.382	20.873	20.800	20.800	30.703
V-L Flowrate (lb_{mol}/hr)	6,528	9,838	6,528	9,838	3,160	4,763	3,160	64,034	64,034	58,271	74,931	78,773	78,773	17,287
V-L Flowrate (lb/hr)	208,921	314,852	208,921	314,852	54,760	82,525	54,760	1,241,087	1,241,087	1,129,389	1,564,001	1,638,466	1,638,466	530,751
Solids Flowrate (lb/hr)	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Temperature (°F)	90	90	264	300	100	100	243	1,799	1,236	600	2,599	792	139	100
Pressure (psia)	30	30	2,500	420	450	450	2,500	2,150	400	390	380	2	2	2
Enthalpy (Btu/lb)[A]	12.5	12.5	44.4	58.2	20.7	20.7	72.5	-2,770,898.0	1,488.1	1,170.4	2,064.5	1,144.1	871.2	334.7
Density (lb/ft³)	0.163	0.163	9.804	1.637	1.359	1.359	5.774	1.698	0.429	0.704	0.241	0.003	0.006	0.010

A - Reference conditions are 32.02 F & 0.089 PSIA

	15	16	17	18	19	20	21	22	23	24	25	26	27	28
V-L Mole Fraction														
Ar	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
CH_4	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
C_2H_6	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
C_3H_8	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
C_4H_{10}	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
CO	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
CO_2	1.0000	0.0001	0.0000	0.0000	1.0000	1.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
H_2O	0.0000	0.9999	1.0000	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
N_2	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
O_2	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
Total	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
V-L Flowrate (kg_{mol}/hr)	3,744	3,918	31,808	7,199	3,744	3,744	24,609	24,609	11,217	11,217	9,936	9,936	21,153	3,456
V-L Flowrate (kg/hr)	164,788	70,599	573,050	129,689	164,788	164,788	443,343	443,343	202,080	202,080	179,006	179,006	381,086	62,257
Solids Flowrate (kg/hr)	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Temperature (°C)	119	38	36	38	27	28	38	39	39	353	39	353	353	39
Pressure (MPa, abs)	14.48	0.04	0.01	0.34	14.20	15.27	0.34	17.93	17.93	17.24	17.93	17.24	17.24	17.93
Enthalpy (kJ/kg)[A]	12.18	153.68	153.62	159.40	-230.40	-228.52	159.40	180.23	180.23	2,510.83	180.23	2,504.21	2,679.93	176.95
Density (kg/m^3)	253.3	663.8	993.6	993.2	757.3	761.2	993.2	1,000.3	1,000.3	126.3	1,000.3	127.1	108.6	747.6
V-L Molecular Weight	44.010	18.017	18.016	18.016	44.010	44.010	18.016	18.016	18.016	18.016	18.016	18.016	18.016	18.016
V-L Flowrate (lb_{mol}/hr)	8,255	8,639	70,125	15,870	8,255	8,255	54,254	54,254	24,729	24,729	21,906	21,906	46,635	7,619
V-L Flowrate (lb/hr)	363,295	155,644	1,263,359	285,915	363,295	363,295	977,405	977,405	445,511	445,511	394,641	394,641	840,152	137,253
Solids Flowrate (lb/hr)	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Temperature (°F)	246	100	98	100	80	83	100	102	102	668	102	668	668	102
Pressure (psia)	2,100	6	2	50	2,060	2,215	50	2,600	2,600	2,500	2,600	2,500	2,500	2,600
Enthalpy (Btu/lb)[A]	5.2	66.1	66.0	68.5	-99.1	-98.2	68.5	77.5	77.5	1,079.5	77.5	1,076.6	1,152.2	76.1
Density (lb/ft^3)	15.816	41.440	62.028	62.006	47.275	47.519	62.006	62.449	62.449	7.882	62.449	7.932	6.782	46.672

An alternative "revised" CES cycle has been also proposed, shown in Figure 18 (Hustad, 2009). The main differences of this configuration with respect to the conventional one are: (i) the low pressure turbine (10-GS03) discharge at near atmospheric pressure, (ii) the presence of a two stage intercooled compression (10-CX01, 10-CX02) of the working fluid for temperature control in the reheater (10-HG01) and (iii) the presence of a low pressure condensing steam turbine expanding to sub-atmospheric pressure the low pressure steam generated from heat recovery from compressor inter- and after-cooler (10-HS01, 10-HS03) and a portion of the exhaust steam from the high pressure turbine, after reheating.

For this configuration, a net cycle efficiency of 47% (with unclear reference to either LHV or HHV) is reported.

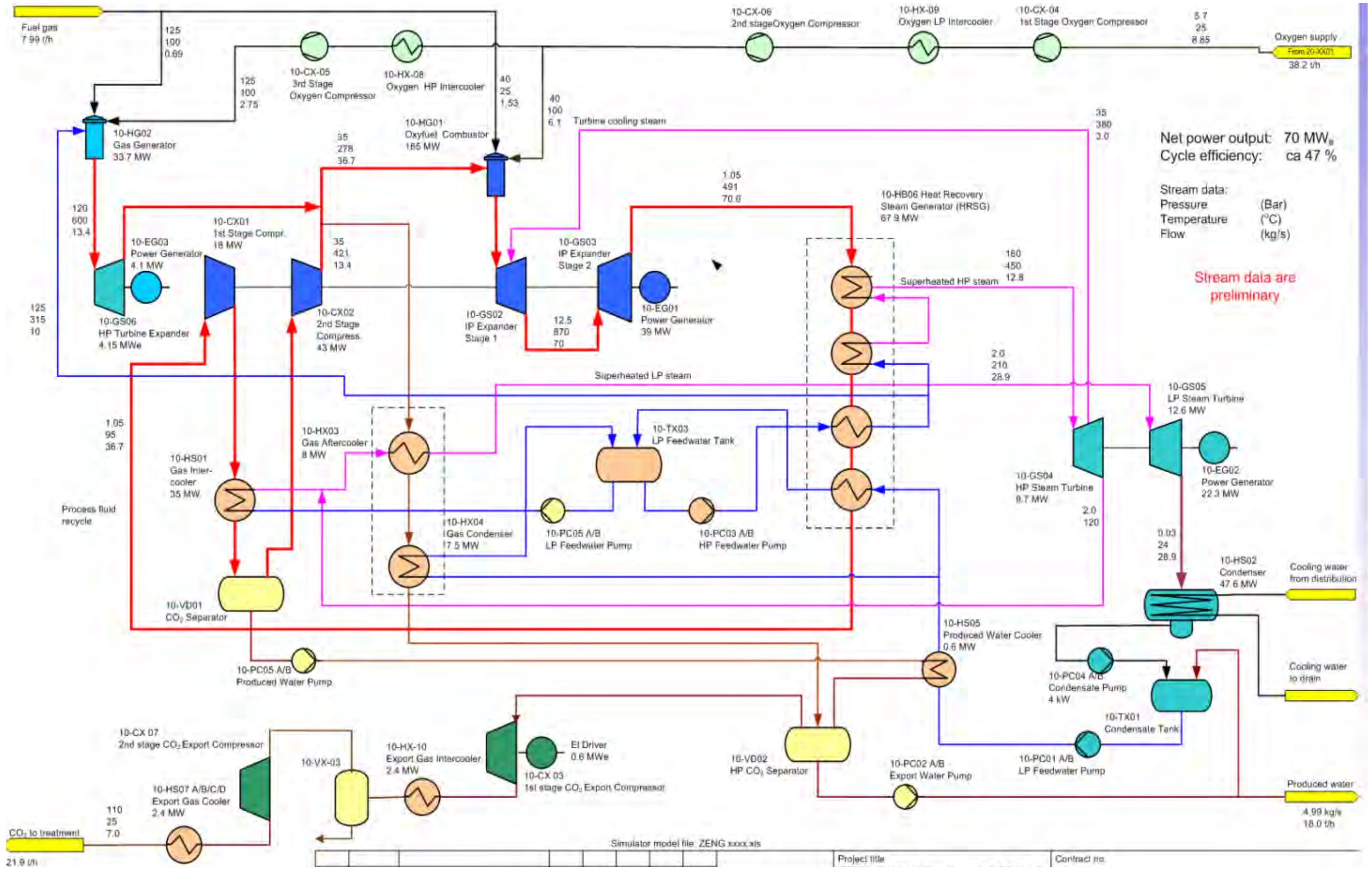


Figure 18. Configuration of the revised CES cycle Hustad (2009)

The following main components of the CES cycle require discussion:

- **Turbines**. Commercial gas turbines have been adapted for the HTT of the CES cycle. In particular, operations of the OFJ-79 (i.e. the modified GE J79) and of the OFT-900 (i.e. the modified Siemens SGT-900) turbines at the CES Kimberlina (CA) test facility are reported by CES (2013). Operating pressures, temperatures and expansion efficiencies achieved in these tests are however not available at the author's knowledge. As far as efficiency is concerned, it is expected to be rather low, due to the much different nature of the steam-based working fluid with respect to air combustion gases, for which the machines are designed. Therefore, it is believed that for commercial plants, a dedicated design is needed to have acceptable expansion efficiencies. Similarly to previous cycles, no real technological barrier is foreseen for this machine considering the operating conditions similar to commercial gas turbines, but a market driver is again needed to justify the cost of development. It must be remarked that in this case, the target TIT (1760°C) of the full scale plant proposed by CES is significantly higher than TIT of current technology gas turbines. Therefore, a reduced value might be considered, compatible with the status of gas turbine technology. Particular attention needs also to be given to material selection, considering the corrosive environment in presence of high steam concentrations, as highlighted in lab tests (Anderson et al., 2010).

  Lower uncertainties and easier adaptability of current design seems possible for the high pressure turbine. A proper material selection is however needed, considering the inlet temperature of 760°C, significantly higher than the inlet temperature of state-of-the-art steam turbines.

  Similar considerations can be made for the low pressure turbine, for which a design adapted from commercial steam turbines seems to be possible. Also in this case, the unusually high inlet (760°C) and outlet (430°C) temperatures require proper material selection.

  A few details on the turbine design are discussed in the works on CES cycle, some of these are reported in Figure 19.

- **Gas generator**. This component is very unusual for power plants, being water cooled and derived from rocket technology. Significant tests has been performed by CES in the Kimberlina test facility, where a 20MW_{th} gas generator has been successfully tested for over 2000 h and a 200 MW_{th} gas generator was tested for over 500 starts. Combustion efficiency, maximum temperature and pressure achieved in these tests are however not reported in the open literature at the authors' knowledge.

- **Reheater**. The reheater operates at conditions similar to combustors of conventional turbines. Therefore, despite a dedicated design is needed, no real technological challenge is foreseen for this component. Tests on a 20 MW_{th} reheater have been performed (Anderson et al., 2010) at the CES Kimberlina test facility. Stable and high efficiency combustion is reported for that test campaign, which however was performed at conditions (inlet pressures of 6.4-8.2 bar and exhaust temperatures of 600-720°C) far from the expected pressure and temperatures of the final full scale plant.

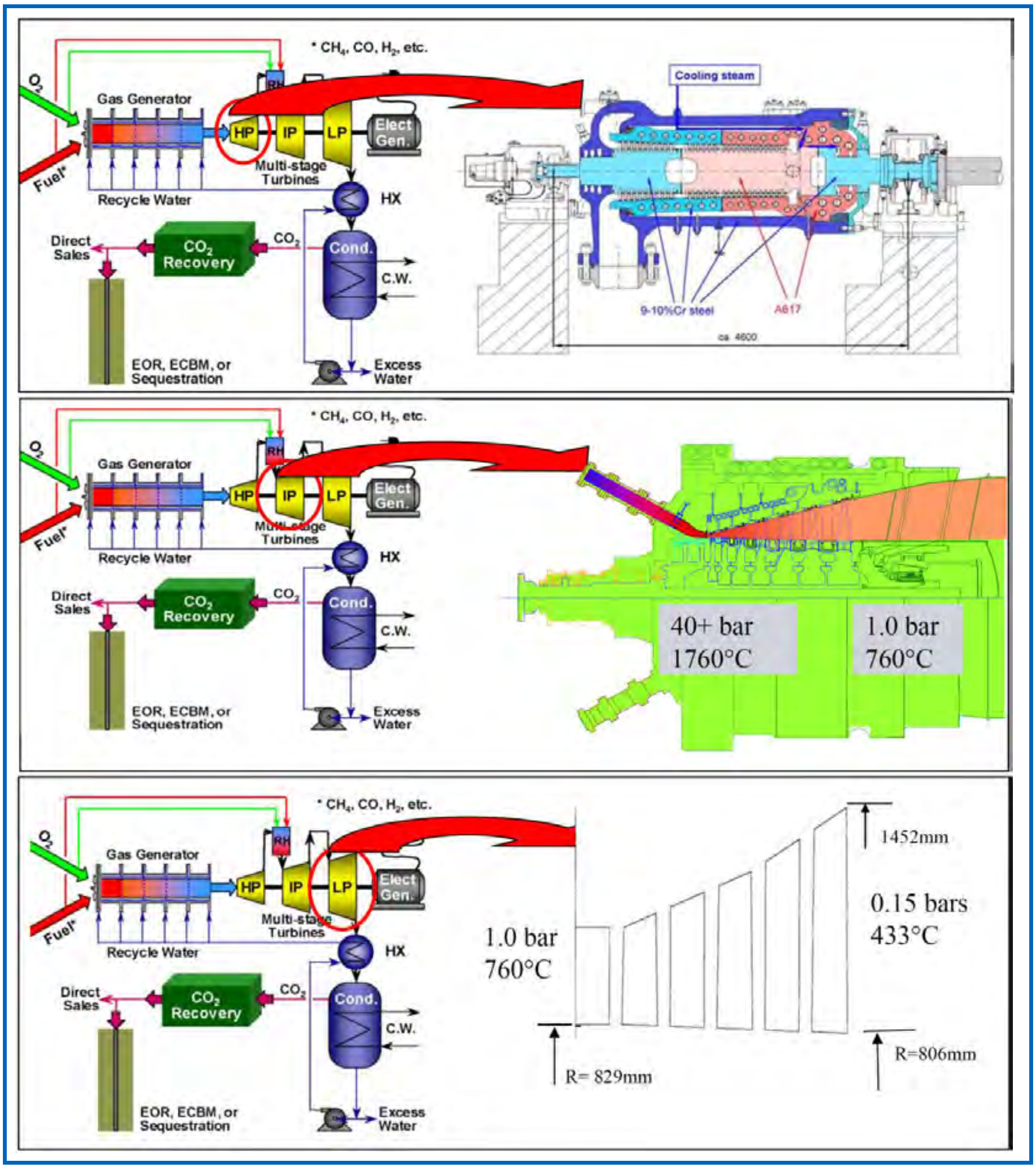


Figure 19. Basic design of the turbines of the CES cycle (Anderson et al., 2008)

2.6. AZEP cycle

AZEP (advanced zero emission power plant) cycle is based on an externally fired gas turbine cycle using an oxygen transport membrane (OTM) for oxygen separation, as shown in Figure 20. Air exiting the compressor of the gas turbine is sent to the mixed conducting membrane reactor (MCM). Here, most of the flow rate is heated up to about 700-900°C in a lower temperature heat exchanger (LTHX) and oxygen is separated with an OTM, where a high temperature recycled CO_2/H_2O-based stream is used as sweep gas. The oxygen-rich stream obtained is then partly cooled in the lower temperature heat exchanger, compressed with an ejector or a blower and used to burn natural gas under near-stoichiometric conditions. High temperature combusted gas is then partly recycled to heat the oxygen depleted air up to the maximum temperature in a high temperature heat exchanger (HTHX), before entering the membrane module as sweep gas, facilitating in this way the oxygen permeation. The remaining part of the gas from the combustor is cooled with a fraction of air from the gas turbine compressor in another high temperature heat exchanger and then sent to the CO_2 compression section. High temperature compressed air and O_2-depleted air are then mixed and expanded in the gas turbine. Heat from CO_2 cooling and from gas turbine exhaust is recovered by the bottoming steam cycle.

With TITs in the range of 1140-1275°C, efficiencies of 48.9-50.0% are reported in the literature, corresponding to penalties of 6.1-8.3% points with respect to a reference combined cycle without CO_2 capture (Möller et al., 2006; Kvamsdal et al., 2007; Anantharaman et al., 2009; Mancini and Mitsos, 2011). In order to increase the TIT, layouts with post-combustion of the high temperature air with NG have been proposed. In this way, by accepting higher CO_2 emission (which are virtually zero in the base configuration), plant efficiency can be greatly increased. Net efficiencies of 51.1-53.4% were for example obtained for an 85% carbon capture ratio (Möller et al., 2006; Kvamsdal et al., 2007; Anantharaman et al., 2009; Mancini and Mitsos, 2011).

Replacement of membrane permeators with membrane reactors, where permeating O_2 is consumed by oxidation of natural gas, is also possible. This layout was initially proposed for the AZEP concept by Sundkvist et al. (2001) and recently reconsidered by Mancini and Mitsos (2011), who recommend a hybrid layout with a membrane reactor followed by a membrane permeator (i.e. with no reactions on the permeate side). Effects on efficiency of this layout are limited, but lower membrane surface are expected for an optimized configuration.

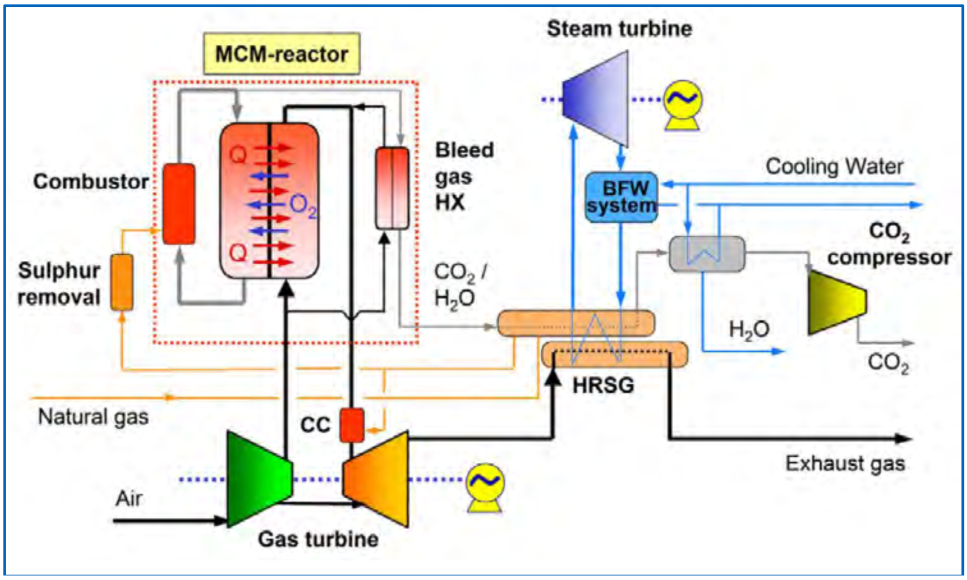


Figure 20. Configuration of the AZEP cycle with post-combustion (Anantharaman et al., 2009)

Turbomachines do not represent critical components for this cycle and a commercial gas turbine can be quite easily adapted for the AZEP cycle conditions. On the other hand the high temperature heat exchanger and the OTM permeator are novel components requiring significant R&D efforts for material selection and component design. Sundkvist et al. (2007), report successful testing of the MCM reactor components. Monolith-based modules composed of membrane and heat exchangers, all made of ceramic materials, have been developed and manufactured. The modules have been tested at up to 10 bar and around 850°C for relatively low operation times. In these tests, oxygen permeations in line with models predictions and good heat transfer coefficients were obtained. Despite these encouraging preliminary results and the recent advancements in oxygen membrane technology, the uncertainties about a successful development of the MCM reactor from both the technological and the economic sides remain very high.

2.7. ZEITMOP cycle

Another natural gas-fired oxy-fuel plant, based on OTM for oxygen production is the ZEITMOP cycle (Figure 21 and Figure 22), proposed by Yantovski et al. (2004).The cycle is based on a supercritical CO_2 cycle, where CO_2 is compressed in an intercooled compressor (component 11) up to over 200 bar, heated in a recuperative heat exchanger (10) and expanded in a high pressure turbine (9) to about 15 bar. It is then used as sweep gas in an OTM (4), where it is enriched with O_2 separated from a stream of compressed air. CO_2/O_2 flow is used as oxidant in a NG combustor (7) which produces high temperature oxidized gas to be expanded to nearly ambient temperature in a high temperature and medium pressure turbine (6). Air used as source of O_2 is compressed in an air compressor (1) and heated up to the OTM operating temperature in a heat exchanger (3) using the hot expanded CO_2 as hot stream. The oxygen depleted air from the membrane is then expanded to ambient pressure in a turbine (5).

Efficiencies of 50.4-52.0% with virtually zero CO_2 emissions are reported for this cycle (Yantovski et al., 2004; Foy and McGovern, 2007). A configuration with reactive membrane can also be adopted to reduce the membrane surface. In this case, cycle efficiency strongly depends on the maximum tolerable membrane temperature, which coincides with the maximum cycle temperature. As an example, by increasing OTM outlet temperature form 1000°C to 1400°C, efficiency improves from 46% to 51% (Foy and McGovern, 2007).

Most of the technological uncertainties of this cycle lie in the oxygen membrane. The high temperature heat exchangers used for air and high pressure CO_2 heating are other critical components from the techno-economic point of view. CO_2 turbines also require dedicated design and development, but with a lower level of uncertainty with respect to the oxygen membrane.

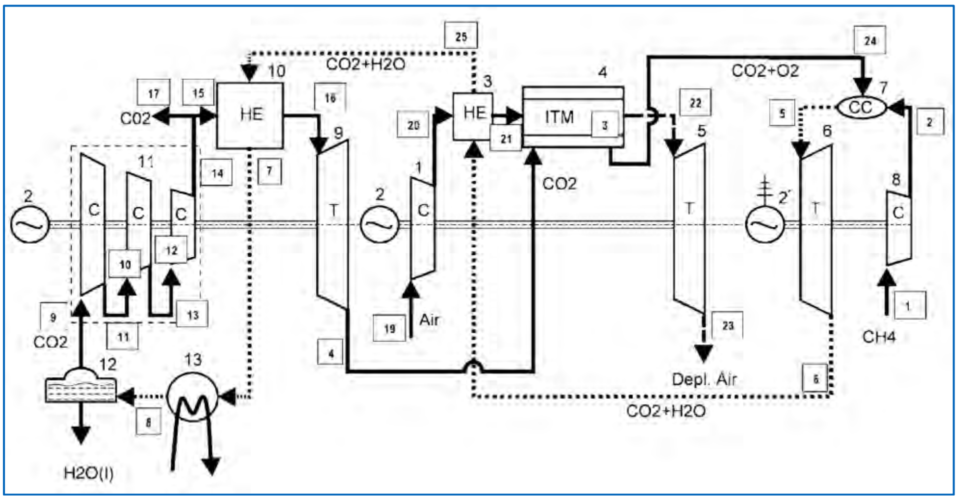


Figure 21. Configuration of the ZEITMOP cycle (Yantovski et al., 2004)

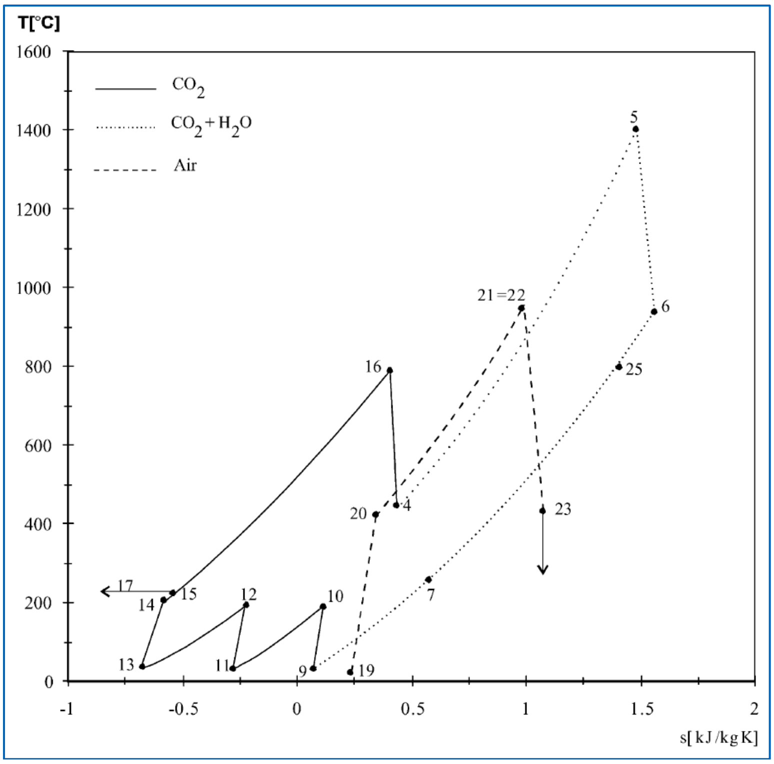


Figure 22. The ZEITMOP cycle on a temperature-entropy chart (Yantovski et al., 2004)

3. Selection of Oxyfuel cycles

From the literature review presented in the previous sections, the oxyfuel cycles can be evaluated on the basis of the expected efficiency and on the technological development required for their key components. For this second task, an index with value between 1 and 4 can be attributed to each plant component, according to the expected efforts required for its commercial development. The lowest development level is required for units with index 1, being already available on the market or easily adaptable from commercial components. The highest index of required development is 4, indicating highly immature components, still at a status of lab scale material testing. The complete list of the indexes used to evaluate the development level of innovative components is reported in Table 2.

These indexes, which have a qualitative nature and reflect the sensibility of the authors, are used to evaluate the class of components (turbomachines, combustors, heat exchangers and oxygen production technology) of each oxyfuel cycle assessed, as shown in Table 3. In Table 3, the range of efficiencies reported in the literature, which is the other primary criteria used to select the cycles, is also resumed.

Table 2. Index of the level of development of the components of the oxyfuel power cycles reviewed.

Development index	Comment
1:	Commercial components or technology very close to the state-of-the-art can be used. Modifications to existing components or new design activities are moderate and relatively low-cost.
2:	The component requires dedicated design, but component development seems feasible in short-mid term with the current knowledge. Limited modifications to the process parameters are possible, considering techno-economic limitations on component design.
3:	The component requires dedicated design and operates at conditions far from current commercial components. Significant modifications to the process parameters might be needed on the basis of possible techno-economic limitations on component design.
4:	The component requires a technology breakthrough and the successful techno-economic development is highly uncertain.

Operating flexibility of the plants would be another important element to rank the different cycles. However, evaluating the plants flexibility is challenging in this case, since this would require a detailed design of the cycle components, which is considered premature with the current knowledge of such innovative plants. Therefore, flexibility aspects are not considered in the present evaluation, but briefly discussed in Chapter F.2.

Table 3. Resume of component development index as per Table 2 and cycle efficiencies of the reviewed oxyfuel gas cycles.

Cycle	Turbomachines	Combustion system	Heat exchangers	Oxygen production	Cycle efficiency, %
SCOC-CC	Compressor: 1 Turbine: 2	GT combustor: 1	HRSG: 1	ASU: 1	45-48
MATIANT	Compressors: 1 HP turbine: 2 MP turbine: 2 LP turbine: 2	MP combustor: 1 LP combustor: 1	Regenerator: 2	ASU: 1	40-49
E-MATIANT	Compressors: 1 Turbine: 2	Combustor: 1	Regenerator: 2	ASU: 1	46-47
NET Power	Compressors: 1 Turbine: 3	Combustor: 2	Regenerator: 2	ASU: 1	55-59
Graz cycle	Compressors: 2 HT turbine: 2 HP turbine: 1 LP turbine: 1	Combustor: 1	HRSG: 1 Condenser/ Evaporator: 1	ASU: 1	49-54
CES cycle	HP turbine: 1 HT turbine: 2 LT turbine: 1	Gas generator: 2 Reheater: 1	HRSG: 1	ASU: 1	45-50
AZEP	Compressor: 1 Turbine: 1	Combustor: 1	HT heat exchangers: 3 HRSG: 1	OTM: 4	49-53
ZEITMOP	Air compressor: 1 N_2 turbine: 1 CO_2 compressors: 1 HP CO_2 turbine: 2 LP CO_2 turbine: 2	Combustor: 2	Air heater: 2 HP CO_2 heater: 2	OTM: 4	46-51

The values of the component development index and of the cycle efficiencies have been used to attribute an overall score to each cycle. An initial score between 0 and 10 was initially given on the basis of the cycle efficiency. A score of 10 was assigned to the NET Power cycle, for which an outstanding efficiency higher than 55% is reported. Scores of 9, 8 and 7 have been assigned to cycles with expected efficiencies above 50% (Graz and AZEP cycles) around 50% (CES and ZEITMOP) and below 50% (SCOC-CC, MATIANT and E-MATIANT) respectively. A penalty was then added to this initial score, on the basis of the development indexes. For each component with a development index of 2, a penalty of 1 point was given. Similarly, for each component with a development index of 3 and 4, penalties of 2 and 4 points were given. So, for example, a final score of 6 was obtained for the NET Power cycle by subtracting from the cycle efficiency score (10), 2 points associated to the two components with development index of 2 (the combustor and the regenerator) and 2 points for the component with development index of 3 (the turbine).

The results of this procedure applied to each cycle are shown in Table 4. The highest overall score was obtained by the Graz cycle (7), followed by the SCOC-CC, the NET Power and the CES cycles (6). Therefore, these are the four cycles selected for detailed analysis in this work. Additional motivations for the selection of these cycles are related to the abundance of literature available, with information on possible turbine design (SCOC-CC and Graz cycle), the involvement of companies in the development of the cycle components (NET Power and CES cycles) and the presence of significant past and planned experimental activities (NET Power and CES cycles).

Table 4. Score for development index, cycle efficiency and total score for the cycles examined.

Cycle	Cycle efficiency score	Development index penalty	Total score
SCOC-CC	7	1	**6**
MATIANT	7	4	**3**
E-MATIANT	7	2	**5**
NET Power	10	4	**6**
Graz cycle	9	2	**7**
CES cycle	8	2	**6**
AZEP	9	6	**3**
ZEITMOP	8	9	**-1**

4. References

[1] Allam J.R., Fetvedt J.E., Forrest B.A., Freed D.A., 2014. *The oxy-fuel, supercritical CO_2 Allam cycle: new cycle developments to produce even lower-cost electricity from fossil fuels without atmospheric emissions*. ASME Paper GT2014-26952, Proceedings of ASME Turbo Expo 2014, Düsseldorf, Germany.

[2] Allam J.R., Palmer M., Brown G.W., 2010. *System and Method for High Efficiency Power Generation Using a Carbon Dioxide Circulating Working Fluid*. Palmer Labs, LLC, assignee. Patent US 2011/0179799 A1.

[3] Allam J.R., Palmer M.R., William Brown G., Fetvedt J., Forrest B.A., 2013a. *System and method for high efficiency power generation using a carbon dioxide circulating working fluid*. 8 Rivers Capital, LLC, Palmer Labs, LLC, assignees. Patent US2013/0213049 A1.

[4] Allam J.R., Palmer M.R., William Brown G., Fetvedt J., Freed D., Nomoto H., Itoh M., Okita N., Jones C., 2013b. *High efficiency and low cost of electricity generation from fossil fuels while eliminating atmospheric emissions, including carbon dioxide*. Energy Procedia 37, 1135-1149.

[5] Anantharaman R., Bolland O., Åsen K.I., 2009. *Novel cycles for power generation with CO_2 capture using OMCM technology*. Energy Procedia 1, 335-342.

[6] Anderson R.E., MacAdam S., Viteri F., Davies D.O., Downs J.P., Paliszewski A., 2008. *Adapting Gas Turbines to Zero Emission Oxy-Fuel Power Plants*. ASME paper GT2008-51377, Proceedings of ASME Turbo Expo 2008, Berlin, Germany.

[7] Anderson R., Viteri F., Hollis R., Keating A., Shipper J., Merrill G., Schillig C., Shinde S., Downs J., Davies D., Harris M., 2010. *Oxy-Fuel Gas Turbine, Gas Generator and Reheat Combustor Technology Development and Demonstration*. ASME paper GT2010-23001, Proceedings of ASME Turbo Expo 2010, Glasgow, UK.

[8] Bannister R.L., Newby R.A., Yang W.C., 1999. *Final Report on the Development of a Hydrogen-Fueled Combustion Turbine Cycle for Power Generation*. Journal of Engineering for Gas Turbines and Power 121, 38-45.

[9] Bolland O., Mathieu P., 1998. *Comparison of two CO_2 removal options in combined cycle power plants*. Energy Conversion and Management, 39, 1653-1663.

[10] Clean Energy Systems, Inc., 2013. *Oxy-Turbine Technology Update*. Natural Gas CCS Networking Meeting, 31 May 2013, London, UK.

[11] Chiesa P., Lozza G., 1999. *CO_2 Emission Abatement in IGCC Power Plants by Semiclosed Cycles: Part A – With Oxygen-Blown Combustion*. Journal of Engineering for Gas Turbines and Power 121, 635-641.

[12] Dahlquist A., Genrup M., Sjoedin M., Jonshagen K., 2013. *Optimization of an Oxyfuel Combined Cycle Regarding Performance and Complexity Level*. ASME paper GT2013-94755, Proceedings of ASME Turbo Expo 2013, San Antonio, Texas, USA.

[13] DOE/NETL, 2010. *Carbon Capture Approaches for Natural Gas Combined Cycle Systems*. Report DOE/NETL-2011/1470, Revision 2, December 20, 2010.

[14] Foy K., McGovern J., 2007. *Analysis of the effects of combining air separation with combustion in a zero emissions (ZEITMOP) cycle*. Energy Conversion and Management 48, 3046-3052.

[15] Houyou S., Mathieu P., Nihart R., 2000. *Techno-economic comparison of different options of very low CO_2 emission technologies*. Proceedings of the Fifth International Conference on Greenhouse Gas Control Technologies, Cairns, Australia.

[16] Hustad C.W., 2009. *CO_2 Compression for Advanced Oxy-Fuel Cycles*. Workshop on Future Large CO_2 Compression Systems, Gaithersfield, MD, March 30-31, 2009.

[17] Iantovski E., Mathieu P., 1997. *Highly efficient zero emission CO_2-based power plant*. Energy Conversion and Management 38, S141-S146.

[18] Jericha, H., 1985. *Efficient Steam Cycles With Internal Combustion of Hydrogen and Stoichiometric Oxygen for Turbines and Piston Engines*. CIMAC Conference Paper No. T13, Oslo, Norway.

[19] Jericha H., Sanz W., Göttlich E., 2008a. *Design Concept for Large Output Graz Cycle Gas Turbines*. Journal of Engineering for Gas Turbines and Power 130, 011701-1-10.

[20] Jericha H., Sanz W., Göttlich E., Neumayer F., 2008b. *Design Details of a 600 MW Graz Cycle Thermal Power Plant for CO_2 Capture*. ASME paper GT2008-50515, Proceedings of ASME Turbo Expo 2008, Berlin, Germany.

[21] Kvamsdal H.M., Jordal K., Bolland O., 2007. *A quantitative comparison of gas turbine cycles with CO_2 capture*. Energy 32, 10-24.

[22] Lozza G., Chiesa P., Romano M., Valenti G., 2009. *CO_2 Capture from Natural Gas Combined Cycles*. Proceedings of the 1st International Conference on Sustainable Fossil Fuels for Future Energy – S4FE 2009, Rome, Italy.

[23] Mancini N.D., Mitsos A., 2001. *Conceptual design and analysis of ITM oxy-combustion power cycles*. Physical Chemistry Chemical Physics 13, 21351-21361.

[24] Mathieu P., Nihart R., 1999a. *Zero-Emission MATIANT Cycle*. Journal of Engineering for Gas Turbines and Power 121, 116-120.

[25] Mathieu P., Nihart R., 1999b. *Sensitivity analysis of the MATIANT cycle*. Energy Conversion and Management 40, 1687-1700.

[26] Mathieu P., Dubuisson R., Houyou S., Nihart R., 2000. *New concept of CO_2 removal technologies in power generation, combined with fossil fuel recovery and long term CO_2 sequestration*. ASME paper 2000GT160, Proceedings of ASME Turbo Expo 2000, Munich, Germany.

[27] Möller B.F., Torisson T., Assadi M., 2006. *AZEP Gas Turbine Combined Cycle Power Plants - Thermo-economic Analysis*. International Journal of Thermodynamics 9, 21-28.

[28] NET Power, 2013a. *The NET Power Process - Truly Clean, Substantially Cheaper Energy*. Natural Gas CCS Networking Meeting, 30-31 May 2013, London, UK.

[29] NET Power, 2013b. *The NET Power Process - Truly Clean, Substantially Cheaper Energy*. 4th Annual Global CCUS Summit, 22-23 October 2013, Beijing, China.

[30] Riethmann T., Sandera F., Span R., 2009. *Modelling of a supercharged semi-closed oxyfuel combined cycle with CO_2 capture and analysis of the part-load behavior*. Energy Procedia 4, 415-422.

[31] Romano M.C., Lozza G.G., 2010. *Long-term coal gasification-based power with near-zero emissions. Part B: Zecomag and oxy-fuel IGCC cycles*. International Journal of Greenhouse Gas Control 4, 469-477.

[32] Sanz W., Jericha H., Moser M., Heitmeir F., 2005. *Thermodynamic and Economic Investigation of an Improved Graz Cycle Power Plant for CO_2 Capture*. Journal of Engineering for Gas Turbines and Power 127, 765-772.

[33] Sanz W., Jericha H., Bauer B., Göttlich E., 2007. *Qualitative and Quantitative Comparison of Two Promising Oxy-Fuel Power Cycles for CO_2 Capture*. Journal of Engineering for Gas Turbines and Power 130, 031702-1-11.

[34] Sundkvist S.G., Griffin T., Thourshaug N.P., 2001. AZEP – *Development of an integrated air separation membrane gas turbine*. Second Nordic Minisymposium on Carbon Dioxide Capture and Storage, Göteborg, Sweden.

[35] Sundkvist S.G., Dahlquist A., Janczewski J., Sjödin M., Bysveen M., Ditaranto M., Langørgen Ø., Seljeskog M., Siljan M., 2014. *Concept for a Combustion System in Oxyfuel Gas Turbine Combined Cycles*. Journal of Engineering for Gas Turbines and Power 136, 101513.

[36] Yantovski E.I., 1996. *Stack downward Zero Emission Fuel-Fired Power Plants Concept*. Energy Conversion and Management 37, 867-877.

[37] Yantovski E., Gorski J., Smyth B., Elshof J., 2004. *Zero-emission fuel-fired power plants with ion transport membrane*. Energy 29, 2077-2088.

[38] Yantovski E.I., Zvagolsky K.N., Gavrilenko V.A., 1995. *The COOPERATE – Demo power cycle*. Energy Conversion and Management 36, 861-864.

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	PERFORMANCE EVALUATION OF OXYFUEL GAS TURBINE CYCLE
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	(*)
CHECKED BY	:	N. FERRARI
APPROVED BY	:	L. MANCUSO

(*) P. Chiesa, M. Romano, E. Martelli, Politecnico di Milano

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

Oxy-fuel combined cycles, being the most efficient configuration realizable by direct application of the current technology, will be assessed as reference benchmark plants of the oxy-combustion technology.

Politecnico di Milano has carried out a comparative thermodynamic analysis with conventional air-blown combined cycles based on the same technology level in order to assess achievable performance and identify critical aspects inherent to their technical development.

2. Calculation

2.1. Methodology

Heat and mass balance and overall performance of the cases considered in this section have been evaluated by a computer code (named GS) developed by the authors' research group at Politecnico di Milano and thoroughly described in reference [1].

The code is conceived for prediction of gas turbine performance at the design point and includes the one-dimensional design of the turbine, functional to establish all the aerodynamic, thermodynamic, and geometric characteristics of each blade row required for an accurate estimation of the cooling flows and the evolution of the cooled expansion. The model accounts for convective cooling in multi-passage internal channels with enhanced heat transfer surfaces, as well as film and TBC cooling.

GS code has been extensively used for evaluation of gas turbine based plants in many research projects. Among them, six FP7 collaborative projects awarded to research teams including Politecnico di Milano (Caesar, Cachet II, Demoys, Democlock, Ascent, Matesa). Simulations of oxy-fuel, gas turbine based cycles has been carried out in the frame of a project funded by ENEA about long-term coal gasification-based power with near-zero emissions [2, 3].

In this project the GS model was applied to evaluate the performance of both CO_2-rich (similar to NET Power proposal) and H_2O-rich (similar to CES proposal) gas turbine based cycles. A previous version of the GS code [4] was applied for the simulation of oxy-fuel gas turbines in IGCC power plants [5].

Distinctive features of the calculation code are the following:

- Efficiency of the turbomachines (notably compressors and turbines) are evaluated by correlations accounting for the size of the machines and the main operating parameters (volume flow rate, enthalpy drop, rotational speed) of each stage on the basis of the similarity rules. Correlations have been calibrated to make the code capable to correctly predict the performance of many recent plants used as a benchmark.
- Model for the evaluation of the cooling flow rate takes into account all the parameters such as pressure and thermophysical properties of the fluids flowing on the outer (hot gas stream) and inner side (coolant stream), geometry of the cooling circuit, effect of thermal barrier coating and film cooling, etc. Closed loop cooling circuit can also be considered.
- The sequence of the heat transfer banks in the heat recovery steam generator is automatically defined so as to reduce the gap between the temperature

profiles of the exhaust gases and water/steam stream and consequently the power output of the steam cycle.

- The calculation code is limited to operation with mixtures of ideal gases, except for water/steam that are calculated according to proper equations of state.

The GS code was used just to simulate the power section. The ancillary units as the air separation unit, CO_2 purification and compression units were simulated by Foster Wheeler with proper methodologies and results included in the overall mass and energy balances.

2.2. Air blown combined cycle

The plant flow diagram of the reference combined cycle is reported in Figure 1. Conditions of significant streams are reported in Table 1. Calculation assumptions have been set in order to represent the average performance of current, state of the art combined cycles as defined by the IEAGHG report *CO_2 capture at gas fired power plants* [[6]].

A power plant composed of two gas turbines, two heat recovery gas generators and one steam turbine has been considered.

For site conditions and natural gas characteristics reference shall be made to chapter B of the present report. The plant performances results are summarized hereafter:

Gas Turbine

Compressor pressure ratio = 17
Number of turbine stages = 4
Total temperature at first turbine rotor inlet (TIT) = 1430°C
Number of cooled rows = 6
Number of TBC cooled rows = 4
Number of film cooled rows = 3
Fuel temperature at the combustor inlet = 117°C
Inlet filter pressure loss = 7 mbar
HRSG pressure loss = 30 mbar
Air mass flow rate at compressor inlet = 656.94 kg/s = 2365 t/h
Natural gas mass input = 16.515 kg/s
Natural gas thermal input (LHV) = 768.0 MW
Electric gross power = 295.5 MW
Electric gross efficiency = 38.48 %
Turbine outlet temperature = 633.6 °C

3 pressure level + RH steam cycle

Condensation temperature = 29 °C (40 mbara)

Heat losses = 0.3 % of the heat transferred

HP pressure level:

steam temperature at SH outlet = 152.7 bar
steam pressure at SH outlet = 600.6 °C
pinch point ΔT = 9 °C
subcooling ΔT = 0 °C

IP pressure level:

steam temperature at SH outlet = 36.8 bar
steam pressure at SH outlet = 600.9 °C
pinch point ΔT = 9 °C
subcooling ΔT = 0 °C

LP pressure level:

steam temperature at SH outlet = 5.52 bar
steam pressure at SH outlet = 299.5 °C
pinch point ΔT = 9 °C
subcooling ΔT = 10 °C

Flue gas exit temperature = 80.3 °C

Steam turbine electric gross power = 336.5 MW

Heat released to condenser = 486.6 MW

Auxiliaries

Gas turbines = 2 x 1.25 MW

Boiler feed water pumps = 4.75 MW

Electric consumption for heat rejection = 4.87 MW = 1 % of the heat released

Miscellaneous = 11.6 MW

Overall electric consumption = 23.7 MW

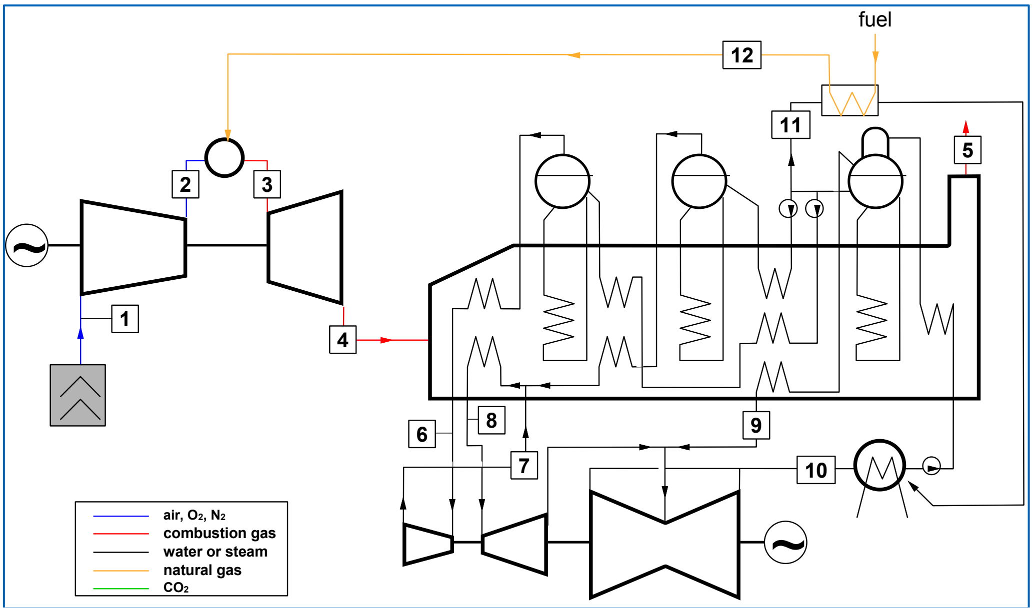


Figure 1. Plant layout of the reference combined cycle.
For sake of simplicity just one gas turbine and one HRSG are represented in the scheme.

Table 1. Combined cycle H&MB.

	T, °C	p, bar a	G, (1) kg/s	Q, (1) kmol/s	Ar, %mol	CO_2 %mol	H_2O (v) %mol	H_2O (l) %mol	N_2 %mol	O_2 %mol
1	9.0	1.01	2 x 656.9	2 x 22.759	0.922	0.030	0.906	0.000	77.382	20.760
2	393.7	17.1	2 x 500.0	2 x 17.322	0.922	0.030	0.906	0.000	77.382	20.760
3	1533.3	16.6	2 x 516.5	2 x 18.281	0.873	5.465	11.064	0.000	73.366	9.231
4	633.6	1.04	2 x 673.5	2 x 23.718	0.884	4.219	8.736	0.000	74.287	11.874
5	80.3	1.01	2 x 673.5	2 x 23.718	0.884	4.219	8.736	0.000	74.287	11.874
6	600.6	152.7	173.1	9.6063	0.000	0.000	100.000	0.000	0.000	0.000
7	391.1	40.0	171.3	9.5103	0.000	0.000	100.000	0.000	0.000	0.000
8	600.9	36.8	193.6	10.744	0.000	0.000	100.000	0.000	0.000	0.000
9	299.5	5.52	23.59	1.3096	0.000	0.000	100.000	0.000	0.000	0.000
10	29.0	0.04	217.2	12.054	0.000	0.000	90.010	9.990	0.000	0.000
11	158.8	6.0	2 x 7.04	2 x 0.3906	0.000	0.000	0.000	100.000	0.000	0.000
12	117.0	70.0	2 x 16.52	2 x 0.9165	Natural gas composition as assigned					

(1) Steam flow rates refer to the production of both the HRSG. Gas flow rates refer to a single gas turbine unit.

In the following Table 2, the performance results of the plant here considered are reported and compared to the ones of the reference combined cycle considered in [6].

Table 2. Performance comparison between reference combined cycle considered here and that in IEAGHG report [6].

	Data from [6]	Simulated plant
Air mass flow rate, kg/s	656.9	656.9
Turbine outlet temperature, °C	639.5	633.6
GT gross power output, MW	2 x 295.2	2 x 295.5
ST gross power output, MW	343.5	336.5
Overall gross power output, MW	934.0	927.5
Auxiliaries, MW	23.7	23.7
Net power output, MW	910.3	903.8
Fuel input, MW	1546.2	1536.0
Plant net efficiency	0.5887	0.5884
GT gross efficiency	0.3819	0.3848

Additional relevant results regarding the efficiency of the turbomachines included in the simulated plant are summarized in the table below. These values are calculated by the calibrated code keeping into account penalties related to dimensional effects and not optimal specific speed.
In gas turbine expansion, stage efficiency relates to an equivalent total-to-total efficiency of each stage referred to an adiabatic process without any coolant mixing. Efficiency of the last steam turbine section includes leaving losses.

Gas turbine compressor	
Average isentropic efficiency	88.758%
Average polytrophic efficiency	92.179%
Gas turbine expander	
1st stage efficiency	89.689%
2nd stage efficiency	90.999%
3rd stage efficiency	92.069%
4th stage efficiency	92.103%
Steam turbine	
section from 152.72 to 40 bar (6 stages)	88.811%
section form 36.8 to 5.52 bar (10 stages)	93.810%
section from 5.52 to 1.512 bar (6 stages)	94.108%
section from 1.512 to 0.04 bar (6 stages)	86.749%

2.3. Semi-closed oxy-combustion combined cycle (SCOC-CC)

Plant flow diagram adopted for the SCOC-CC arrangement is illustrated in Figure 2. Conditions of significant streams are reported in Table 3.

The rationale for the selection of the main design parameters are discussed in the following:

- A conventional low pressure double-column ASU plant is considered, as typically foreseen also for modern large IGCC plants when only oxygen production is required. Air compressor is electrical driven. Oxygen is separated in liquid phase from bottom of the LP cryogenic distillation column, pumped to the required pressure and next vaporized. The energy consumption for ASU are calculated by Foster Wheeler and accounted for in the final energy balance.
- In the configuration considered for this study case, the oxygen stream is directly fed into the combustion chamber, as shown in Figure 2.

  Alternatively, the O_2 stream in gaseous phase can be mixed at the compressor inlet. This latter option could lead to some benefits as: i) simplification of the plant layout; ii) better efficiency of the O_2 compression; iii) higher inlet temperature of the O_2 in the combustor with beneficial effects on conversion efficiency. However, this solution is negatively affected by the open loop cooling technology adopted for the gas turbine. The cooling flows that bypass the combustor represent about 35% of the flow at the compressor inlet. If O_2 is mixed at the compressor inlet, the O_2 entrained in the cooling flows is released in the exhaust gas leading to an increase of the O_2 flow rate from the ASU and consequently more energy is required for CO_2 purification to achieve the required oxygen purity in the CO_2 product, due to the higher O_2 concentration in the CO_2 rich stream delivered to the purification unit.
- 97% purity of the oxygen stream from the ASU has been assumed as a project design bases. To ensure fuel complete combustion, oxidant excess is provided in the amount of the 3% of the O_2 required to stoichiometrically burn the natural gas. This assumption, coherent with [7], leads to a 0.52% O_2 molar concentration at the combustor outlet (wet basis) that is significantly higher the one quoted by the NET Power [8] where a concentration "greater than about 0.1% molar" is indicated.
- Exhaust gas from the HRSG are cooled in an Indirect Contact Cooler (ICC). Cooling water cooled water distributed at the top moves down by gravity and cools the CO_2 stream rising inside the tower. Most of the water vapor entrained in the CO_2 stream is condensed and mixes to the liquid stream. Temperature of the stream recirculated to the compressor inlet is 28°C.
- Incondensable gases are partly removed the CO_2 rich stream in a low temperature process not represented in Figure 2. Separation efficiency and energy consumptions for separation and compression have been calculated by Foster Wheeler and results introduced in the overall heat and mass balance.
- To increase plant efficiency, thermal integrations with the ancillary units has been included in the plant scheme. In particular, the oxygen stream is warmed up by a heat duty available from the aftercooler of the LP CO_2 compressor

section while the fuel stream by heat from the aftercooler of the HP section. A 1.4 MW heat duty, required by the TSA regeneration within the CO_2 purification unit, is provided by the two heat recovery steam generators to the plant.

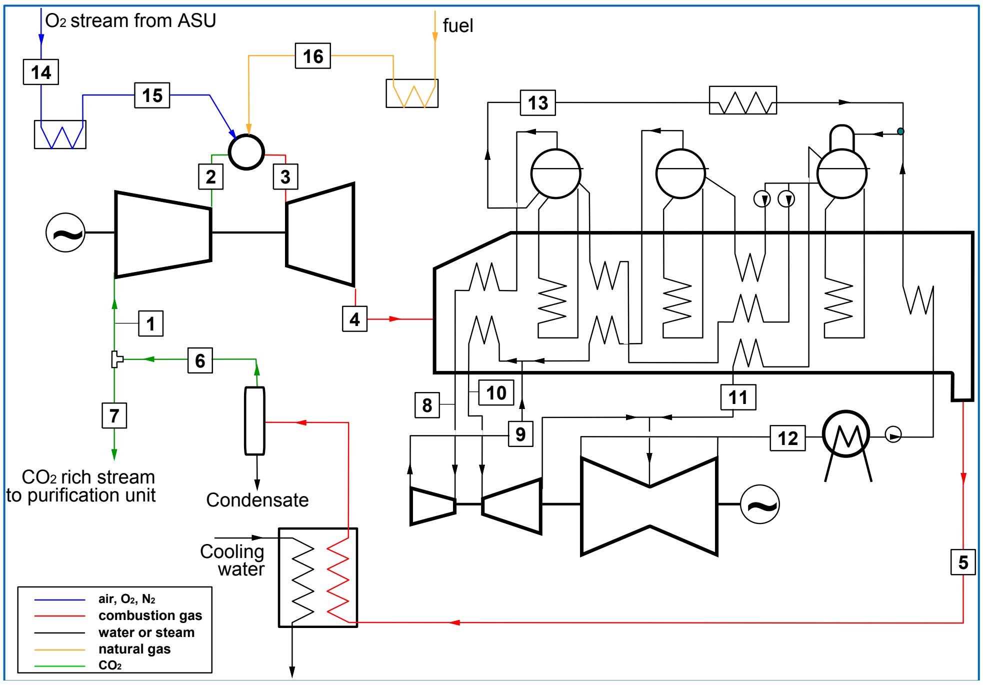


Figure 2. Plant layout of the SCOC-CC configuration.
For sake of simplicity just one gas turbine and one HRSG are represented in the scheme.

Table 3. SCOC-CC H&MB.

	T, °C	p, bar a	G, (1) kg/s	Q, kmol/s	Ar, %mol	CO_2 %mol	H_2O (v) %mol	H_2O (l) %mol	N_2 %mol	O_2 %mol
1	28.0	1.03	2 x 638.6	2 x 15.046	3.563	89.732	3.668	0.000	2.518	0.518
2	412.7	45.8	2 x 409.5	2 x 9.6469	3.563	89.732	3.668	0.000	2.518	0.518
3	1533.3	44.4	2 x 489.4	2 x 12.580	3.046	76.714	17.644	0.000	2.153	0.443
4	619.2	1.07	2 x 718.5	2 x 17.978	3.201	80.623	13.447	0.000	2.262	0.466
5	66.9	1.03	2 x 718.5	2 x 17.978	3.201	80.623	13.447	0.000	2.262	0.466
6	28.0	1.03	2 x 685.7	2 x 16.153	3.563	89.732	3.668	0.000	2.518	0.518
7	28.0	1.03	2 x 47.02	2 x 1.1076	3.563	89.732	3.668	0.000	2.518	0.518
8	589.7	152.7	181.6	10.079	0.000	0.000	100.000	0.000	0.000	0.000
9	381.5	40.0	179.8	9.978	0.000	0.000	100.000	0.000	0.000	0.000
10	593.1	36.8	204.0	11.323	0.000	0.000	100.000	0.000	0.000	0.000
11	299.5	5.52	23.32	1.2944	0.000	0.000	100.000	0.000	0.000	0.000
12	29.0	0.04	227.3	12.618	0.000	0.000	89.810	10.190	0.000	0.000

13	350.3	166.0	1.48	0.082	0.000	0.000	0.000	100.000	0.000	0.000
14	25.0	46.7	2 x 63.38	2 x 1.9734	2.000	0.000	0.000	0.000	1.000	97.000
15	200.0	45.8	2 x 63.38	2 x 1.9734	2.000	0.000	0.000	0.000	1.000	97.000
16	117.0	70.0	2 x 16.52	2 x 0.9165	Natural gas composition as assigned					

(1) Steam flow rates refer to the production of both the HRSG. Gas flow rates refer to a single gas turbine unit.

2.3.1. *Selection of the design parameters*

The most sensible design parameters have been selected after an optimization process. A discussion about the final choice is presented in the following:

- As qualitatively discussed in chapter C.1, the pressure ratio of the SCOC-CC is expected to be higher than a corresponding air blown commercial plant. Pressure ratio of 44.5 has been selected to bring about the same temperature increase (384.7°C) across the compressor of the reference gas turbine.
- The same temperature at the combustor outlet of 1533°C of the reference gas turbine has been assumed. Expansion line and cooling flow rates are calculated accordingly by maintaining the same design parameters of the cooling system (e.g. blade metal temperature, blade wall thickness, TBC thickness, coolant passage cross section) assumed for the reference air breathing gas turbine. The resulting TIT (total temperature at the 1st rotor inlet) is 1352°C that is 78 °C less than the value of the air breathing machine working with the same combustor outlet temperature. The lower TIT actually denotes a higher cooling flow rate in the first stator than the standard turbine that actually increases from 9.1 to 17% of the flow rate at the compressor inlet. This is essentially due to the higher heat flux on the outer blade surface (+123%) resulting from the higher pressure ratio and the change in composition of the working fluid.
- The absolute size of the plant has been chosen in order to maintain the same natural gas input of the reference combined cycle.
- As discussed in chapter C.1, minimum cycle pressure at the compressor inlet can be selected arbitrarily. This choice does not affect significantly the thermodynamics of the cycle but is crucial for the design of the turbomachines. A pressure slightly higher than the ambient pressure (to avoid leakages into the CO_2 loop) has been prudentially selected so as to keep the design of turbomachines closer to the current standards.
- To keep the same rotational speed (3000 RPM), acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine, the number of

stages of the gas turbine expander has been increased to 5. Further details of the machine design is reported below:

Pressure at the compressor inlet: 1.03 bar
Compressor pressure ratio: 44.47
Number of turbine stages: 5
Total temperature at first turbine rotor inlet (TIT) = 1351.8°C
Number of cooled rows: 7
Number of TBC cooled rows: 6
Number of film cooled rows: 5
Fuel temperature at the combustor inlet = 117°C
HRSG + gas cooler pressure loss = 40 mbar
Natural gas mass input = 16.515 kg/s
Natural gas thermal input (LHV) = 768.0 MW
Mass flow rate at compressor inlet = 638.64 kg/s = 2299.1 t/h
97% purity O_2 mass flow rate: 63.38 kg/s
Electric gross power = 305.9 MW
Electric gross efficiency = 39.84 %
Turbine outlet temperature = 619.2 °C

3. Results

The CO_2 purification unit has been calculated by Foster Wheeler for the inlet CO_2 rich stream resulting from the calculation of the power plant. CO_2 concentration in the stream is increased from 89.73% (wet basis) to 99.55%, with 100 ppm O_2 content in agreement with the project design bases. The resulting CO_2 capture rate is 90.6%.

Table 4 draws a comparison between the SCOC-CC and the reference combined cycle. Difference in net electric output is mainly due to the power absorbed by the auxiliaries that in the SCOC-CC accounts for almost 215 MW on a gross power of 968 MW. The net electric efficiency reduces from 58.8 to 49% (LHV basis).

Table 4 shows also the SPECCA (Specific Primary Energy Consumption for CO_2 Avoided) index, which identifies the amount of thermal energy required to avoid the emission of one kg of CO_2. The SPECCA is defined as follows:

$$SPECCA = \frac{HR - HR_{REF}}{E_{REF} - E} = \frac{3600\left(\frac{1}{\eta} - \frac{1}{\eta_{REF}}\right)}{E_{REF} - E}$$

where:

- η is the net electrical efficiency of the SCOC-CC plant;
- η_{REF} is the net electrical efficiency of the reference combined cycle without carbon capture;
- E is the CO_2 emission [kg_{CO2}/MWh_{EL}] of the SCOC-CC plant;
- E_{REF} is the CO_2 emission [kg_{CO2}/MWh_{EL}] of the reference combined cycle without carbon capture.

Additional relevant results regarding the efficiency of the turbomachines included in the SCOC-CC plant are summarized in the table below.

Gas turbine compressor	
Average isentropic efficiency	88.651%
Average polytropic efficiency	91.891%
Gas turbine expander	
1st stage efficiency	87.635%
2nd stage efficiency	88.984%
3rd stage efficiency	90.808%
4th stage efficiency	92.753%
5th stage efficiency	91.382%
Steam turbine	
section from 152.72 to 40 bar (6 stages)	89.028%
section form 36.8 to 5.52 bar (10 stages)	93.867%
section from 5.52 to 1.512 bar (6 stages)	94.096%
section from 1.512 to 0.04 bar (6 stages)	86.573%

Table 4. Performance comparison between reference combined cycle and SCOC-CC plant.

		Reference combined cycle	SCOC-CC
Net electric power output, MW		903.8	754.4
Gas turbine gross output	x 2	295.5	305.9
Steam turbine gross output		336.5	356.2
Auxiliaries			
Gas turbine	x 2	1.25	1.27
Feed water pumps		4.75	5.12
Heat rejection auxiliaries		4.87	8.66
Air separation unit			147.1
CO_2 purification and compression			39.8
Total thermal power output, MW		486.6	866.0
Condenser		486.6	508.5
Recirculated gas direct contact cooler	x 2		115.0
Air separation unit			98.6
CO_2 purification and compression			28.8
Fuel thermal power input, MW		1536.0	1536.0
Net electric efficiency, %		58.84	49.04
Fossil CO_2 produced, kg/s		87.49	87.49
Fossil CO_2 captured, kg/s		0.0	79.30
Carbon capture ratio, %		0.0	90.64
CO_2 emission factor, kg_{CO2}/MWh		348.5	39.14
CO_2 avoided, %			88.77
SPECCA, MJ/kg_{CO2}			3.95

4. References

[1] Chiesa P., Macchi E.: *A Thermodynamic Analysis of Different Options to Break 60% Electric Efficiency in Combined Cycle Power Plants*, ASME paper GT-2002-30663, Best Paper Award of the Electric Power Committee, Journal of Engineering for Gas Turbines and Power, Vol. 126, Issue 4, pp. 770-785, October 2004, DOI: 10.1115/1.1771684.

[2] Romano M.C., Lozza G.G.: *Long-term coal gasification-based power plants with near-zero emissions. Part A: Zecomix cycle*, International Journal of Greenhouse Gas Control 4 (2010) 459–468, doi:10.1016/j.ijggc.2009.11.009.

[3] Romano M.C., Lozza G.G.: *Long-term coal gasification-based power with near-zero emissions. Part B: Zecomag and oxy-fuel IGCC cycles*, International Journal of Greenhouse Gas Control 4 (2010) 469–477, doi:10.1016/j.ijggc.2009.11.010.

[4] Consonni S., 1992, *Performance Prediction of Gas/Steam Cycles for Power Generation*, MAE Dept. Ph.D. Thesis n. 1983-T, Princeton University, Princeton, NJ, USA.

[5] Chiesa P., Lozza G.: *CO_2 Emission Abatement in IGCC Power Plants by Semiclosed Cycles. Part A: with Oxygen-Blown Combustion*, ASME paper 98-GT-384 Journal of Engineering for Gas Turbines and Power, vol. 121, pp. 635-641,October 1999, DOI: 10.1115/1.2818519.

[6] IEAGHG, *CO_2 Capture at Gas Fired Power Plant*, report 2012/08, July 2012.

[7] Jericha H., Sanz W., Göttlich E., Neumayer F.: *Design Details of a 600 MW Graz Cycle Thermal Power Plant for CO_2 Capture*, ASME paper GT2008-50515, Proceedings of ASME Turbo Expo 2008, Berlin, Germany.

[8] R.J. Allam, M.R. Palmer, G.W. Brown JR, J.E. Fetvedt, B.A. Forrest: *System and Method for High Efficiency Power Generation Using a Carbon Dioxide Circulating Working Fluid*, US Patent No. US 2013/0213049 A1, Aug. 22, 2013.

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	BASIC INFORMATION ON OXY-TURBINE POWER PLANT
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	N. FERRARI
CHECKED BY	:	L. MANCUSO
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

LIST OF ATTACHMENT

1. Introduction

The oxy-turbine power plant is a combination of several process units. The main process blocks of the plant are the following:

- Oxy-turbine power island;
- Air Separation Unit;
- CO_2 purification and compression.

Other ancillary utilities, such as cooling water, plant and instrument air, demineralised water support the operation of these basic blocks.

The focus of this Chapter D is to provide a general description of the major blocks of the power plant, which are included in the study base cases, while Chapter D.1 through D.6 of the report give basic engineering information for each alternative, with the support of specific heat and mass balances, utility consumption summaries, etc.

The study cases assessed in the study are listed in the following Table 1.

Table 1. Main study cases

Case	Chapter	Description
Case 1	D.1	Semi-closed oxy-combustion combined cycle (SCOC-CC)
Case 2	D.2	NET Power cycle – single combustor
Case 3a	D.3	S-Graz cycle
Case 3b	D.4	Modified S-Graz cycle
Case 4a	D.5	Basic CES cycle
Case 4b	D.6	Revised CES cycle
Case 4c	D.7	Supercritical CES cycle

2. Basic information of main process units

2.1. Power Island

The following section describes the plant configuration schemes proposed for the selected oxy-turbine cycle. Specific design features, main process parameters and the integration with the other process units are described in the case-specific chapter.

2.1.1. *Semi-closed oxy-combustion combined cycle (SCOC-CC)*

The SCOC-CC resembles a conventional combined cycle and hence it is conventionally used in literature as a benchmark cycle in most of the comparative analyses on natural gas fired oxy-fuel cycles.

The simplified scheme is shown in the below Figure 1.

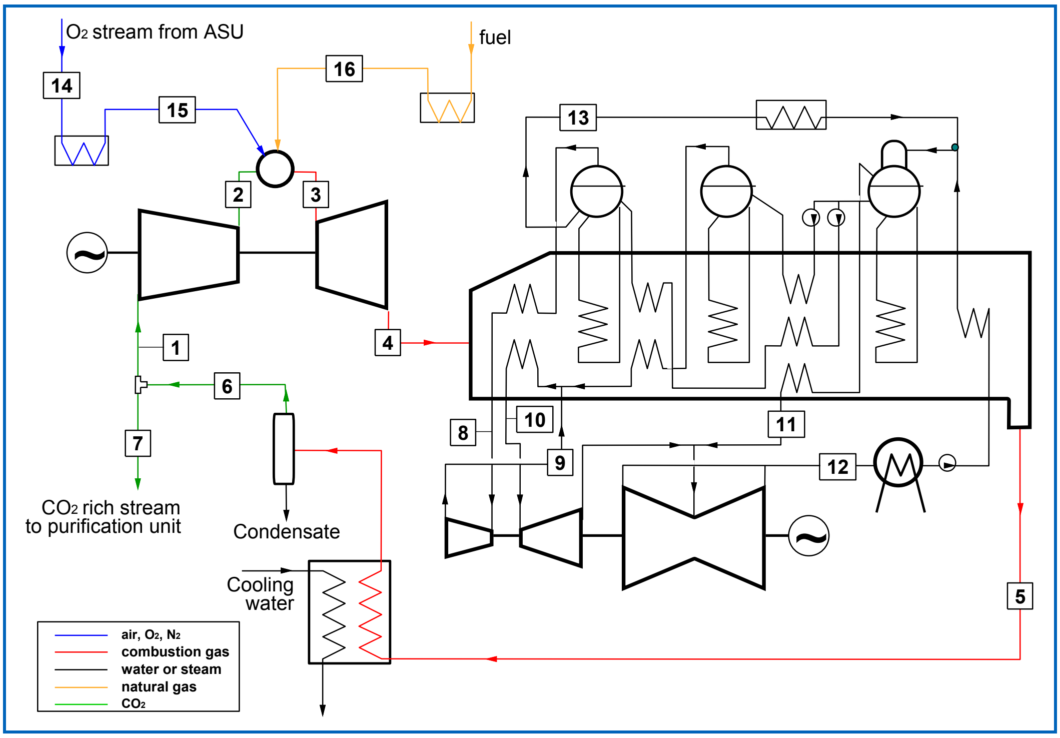


Figure 1. Simplified SCOC-CC plant scheme

The gas turbine compressor recycles part of the cooled CO_2 resulting from the fuel combustion. The hot combustion products are expanded in the turbine and then cooled to nearly ambient temperature in a heat recovery steam generator (HRSG), feeding a bottoming steam cycle. Flue gases are then taken to nearly ambient temperature in a flue gas cooler, where most of the water in the combustion products is condensed.

A portion of this stream, not recycled to the compressor inlet, represents the CO_2 rich stream sent to the purification and final compression unit for long term storage. The higher the presence of such non-condensable gases in the CO_2 stream, the more demanding the CO_2 purification process to achieve the purity specifications of the pipeline and the storage site.

Oxy-turbine

The gas turbine compressor recycles part of the cooled CO_2 resulting from the fuel combustion. The amount of CO_2 recycled is set to achieve the desired combustor outlet temperature (COT) and to provide the cooling flows for turbine blades. After being pre-heated, both fuel and oxygen from the air separation unit are injected in the gas turbine combustion chamber, mixed with the recycle stream from the compressor and finally expanded in the gas turbine. The compression pressure ratio is higher than in the conventional air-fired combined cycle so to have the similar temperature profile in the gas turbine.

The working fluid in the gas turbine cycle is mainly CO_2. The remaining gases are water, whose amount depends on the conditions (temperature and pressure) of the stream recycled to the compressor inlet, along with the fuel composition, excess oxygen necessary to achieve a complete combustion, argon and nitrogen contained in the oxygen stream from the ASU and the nitrogen content in the fuel.

The specific data selected for SCOC-CC study case are reported in chapter D.1.

HRSG

The exhaust gases from the gas turbine are conveyed to the Heat Recovery Steam Generator, located downstream of the machine and connected by means of an exhaust duct.

The simplified process flow diagram of the HRSG is shown in Figure 2.

The HRSG is a natural circulation type, with horizontal flue gas flow arrangement and vertical tubes generating steam at three pressure levels, including integral deaerator for BFW production.

Exhaust gases coming from the gas turbine enter the HRSG casing through the inlet duct, flow counter-current to steam/water and meet in sequence the following coils, before being sent to the recycle duct:

- HP super-heater (2nd section) / MP re-heater (2nd section) (in parallel arrangement);
- HP super-heater (1st section) / MP re-heater (1st section) (in parallel arrangement);
- HP evaporator;
- HP economizer (2nd section) / MP super-heater (in parallel arrangement);

- MP evaporator;
- HP economizer (2nd section) / MP economizer / LP super-heater (in parallel arrangement);
- LP evaporator, with integral deaerator;
- Condensate preheater.

The above sequence of steam/water coils is typical for a standard natural gas-fired combined cycle with a size same as the one selected for the study.

The exhaust flue gas downstream the heat recovery section is sent to a conventional contact cooler. The flue gas is sent to a Venturi scrubber for first quench with water from the bottom of the contact column. In the column, the flue gases are cooled down to 28°C by contact with condensate that has been cooled against cooling water.

Most of the flue gases from the top of the contact column is recycled back to the gas turbine compressors. The remaining stream is sent to the downstream CO_2 purification and compression unit.

During start-up, when the gas turbine is operated in air firing mode the flue gas is discharged to the atmosphere through the start-up stack.

Condensate from the steam turbine condenser, after being treated in the polishing unit, is fed to the condensate pre-heater; a portion of the condensate stream at the pre-heater outlet is recycled back to the inlet by means of a dedicated recirculation pump in order to raise the inlet temperature to the minimum value of 55°C required at HRSG inlet to avoid exhaust gases acid corrosion on HRSG coils.

The pre-heated condensate is then fed to the LP steam drum, equipped with a degassing tower to generate the Boiler Feed Water (BFW).

Degassed BFW for HP and MP services is directly taken from the deaerator and delivered to the relevant sections by means of dedicated BFW pumps.

HP BFW from the deaerator is delivered to the HP economizer coils by means of the HP BFW pumps (one in operation and one in hot stand-by); flows through the HP economizer coils and then feeds the HP Steam Drum. Saturated water from the steam drum is used for the regeneration heater in the CO_2 processing unit.

The generated steam is finally superheated to the maximum possible temperature level in two sections of HP super-heating coils and then sent to the HP module of the steam turbine. To control the maximum value of the HP superheated steam final temperature, a de-superheating station, located between the two HP super-heater coils, is provided. The cooling medium is HP BFW taken from the HP BFW pumps discharge and adjusted through a dedicated temperature control valve.

MP BFW from the deaerator is delivered to the MP economizer coils of the HRSG by means of the MP BFW pumps (one operating and one in stand-by); it flows through the MP economizer coils and feeds the MP steam drum.

The generated MP steam, superheated in the dedicated coils and mixed with the exhaust steam coming from the HP module of the steam turbine, is reheated in the two sections of MP re-heating coils and then enters the MP module of the steam turbine. To control the reheated steam final temperature, a de-superheating station, located between the re-heater coils, is provided. The cooling medium is MP BFW, taken out from the MP BFW pumps discharge and adjusted through a dedicated temperature control valve.

The HP superheated steam and MP reheated steam temperature is selected in order to respect the most severe of the following design criteria:

- Minimum approach temperature between steam and exhaust gas temperature of 25°C to have an adequate heat transfer coefficient and limit the requirement of the surface.
- Maximum steam temperature of 600°C, to use material ASME A 335, 9Cr-1Mo-V, Grade P91 and avoid the use of more exotic materials.

The saturated LP steam from the LP steam generator, after superheating, is mixed with the exhaust of the MP module and then flows to the steam turbine LP module.

Continuous HP and MP blow-down flowrates from the HRSG are manually adjusted by means of dedicated angle valves; they are sent to the dedicated blow-down drum to flash and recover LP steam, which is fed to the deaerator. The remaining flashed liquid is cooled down against cooling water by means of a dedicated blow-down cooler and delivered to the atmospheric blow-down drum, which also collects the possible overflows coming from HRSG's steam drums and the intermittent HP and MP blow-down flowrates, which are manually adjusted by means of dedicated angle valves.

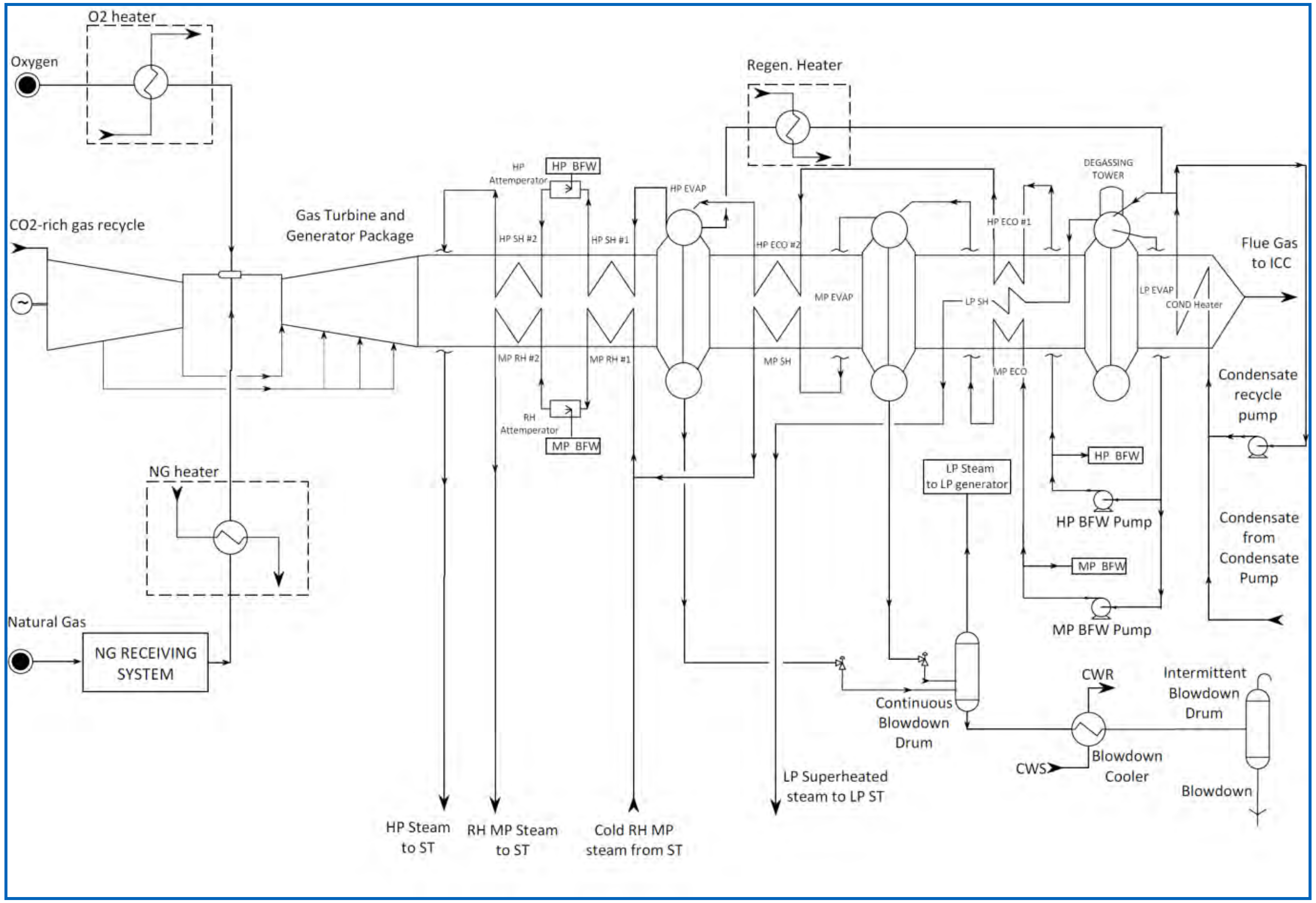


Figure 2. HRSG simplified process flow diagram

Steam turbine

The following process description makes reference to the simplified process flow diagram shown in Figure 3.

The Steam Turbine consists of an HP section, MP section and a double-flow LP section, all connected to the generator by a common shaft. Depending on the alternative, the last stage bucket length of the LP section is selected to have an exhaust annulus velocity in the range of 220-300 m/s.

The superheated HP steam from each HRSG is combined in a header and then enters the HP section of the steam turbine. The exhaust steam from the HP module of the steam turbine is split between the HRSG's, mixed with the MP saturated steam coming from the relevant HRSG section, and reheated. The reheated steam from the HRSGs is combined in a header and then enters the MP section of the steam turbine. The exhaust steam from the MP module of the steam turbine is mixed with the superheated LP steam and delivered to the LP module.

The wet steam at the outlet of the LP module is routed to the steam condenser at 4.0 kPa, corresponding to 29°C. The cooling medium in the tube side of the surface condenser is cooling water from the cooling tower.

The condensate stream, extracted from the steam condenser by means of two, motor-driven and vertical condensate pumps (one operating and one in stand-by), is mixed with demineralised water makeup and sent to condensate pre-heater in the HRSG.

In case of steam turbine trip, live HP steam is bypassed to the MP manifold by means of a dedicated let-down station, while MP steam and excess of LP steam are also let down and then sent directly into the condenser neck.

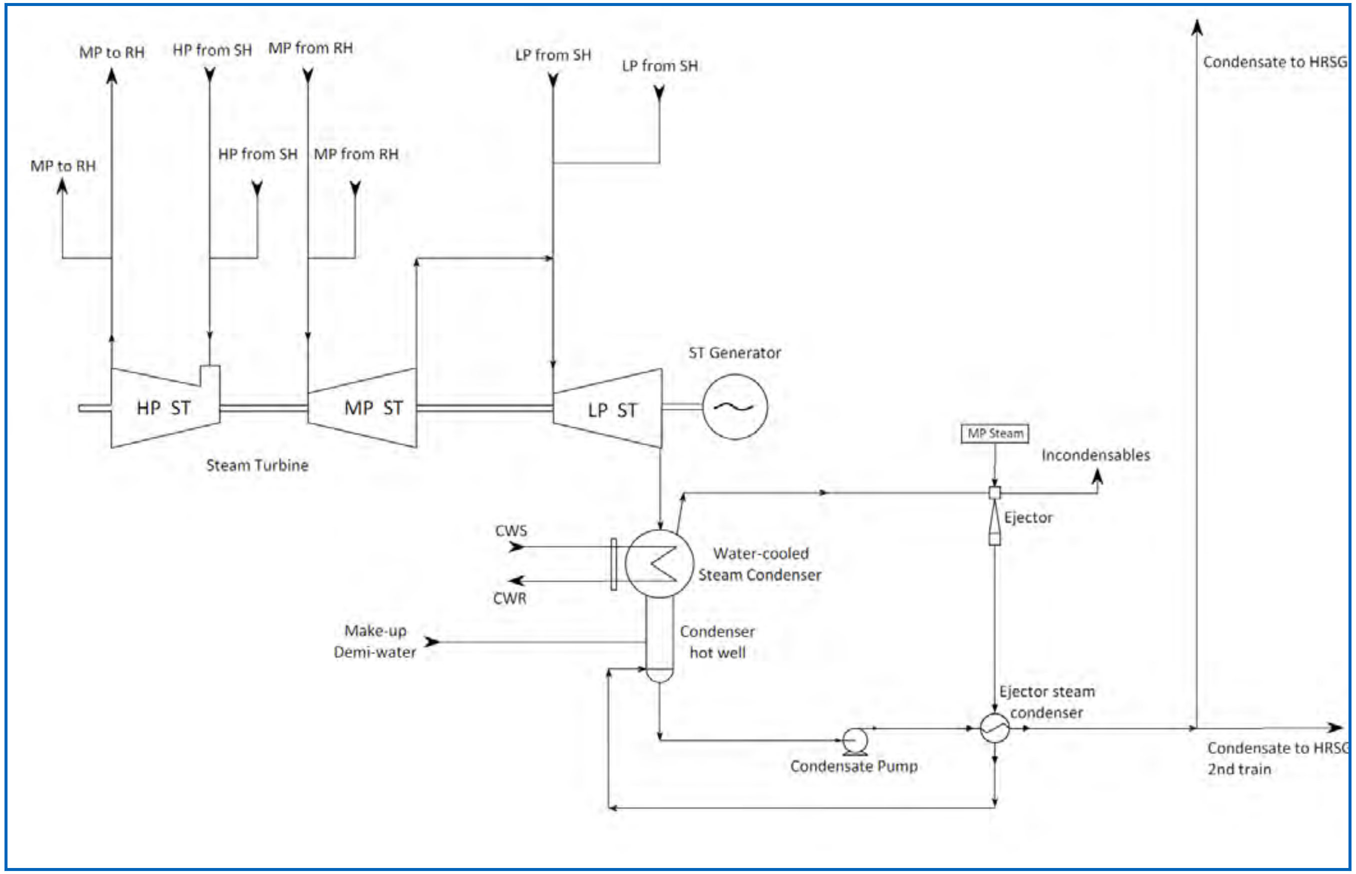


Figure 3. Steam Turbine simplified process flow diagram

2.1.2. NET Power

The NET Power cycle utilizes carbon dioxide as the working fluid in a high-pressure, low-pressure-ratio Brayton cycle, operating with a single turbine that has an inlet pressure in the range of 200 bar to 400 bar and a pressure ratio of 6 to 12. The cycle includes a high pressure oxy-fuel combustor that burns natural gas with a high-purity oxygen stream in a CO_2 environment to provide a high pressure feed stream to a power turbine.

An economizer heat exchanger transfers heat from the high temperature turbine exhaust flow to a high pressure CO_2 recycle stream that flows into the combustor, diluting the combustion products and lowering the combustor outlet temperature.

Heat recovery from the flue gases is maximised in order to enhance the cycle efficiency and reduce the cooling water stream required for the flue gas final cooling. Liquid water derived from hydrogen combustion is separated, and the remaining stream of predominantly carbon dioxide is compressed to the required high pressure. The inert content in the fuel and in the oxygen stream determine the extent to which the recycled stream can to be compressed as affecting the critical pressure of the CO_2-rich stream above which the compression can be performed by means of pumps.

The recycle stream is then reheated in the economizer heat exchanger before returning to the combustor. Part of this recycle stream is used as cooling stream to lower the blade metal temperature in the turbine.

The net CO_2 product derived from the combustion is removed from the recycled stream to be sent to the CO_2 purification unit.

The different schemes developed for this cycle are listed in previous chapter C.1. The present study is developed based on the configuration schematically shown in Figure 4 prepared by Foster Wheeler with the input from NET Power.

The cycle is based on a high pressure gas turbine with inlet pressure around 300 bar and discharge pressure in the range of 30-50 bar.

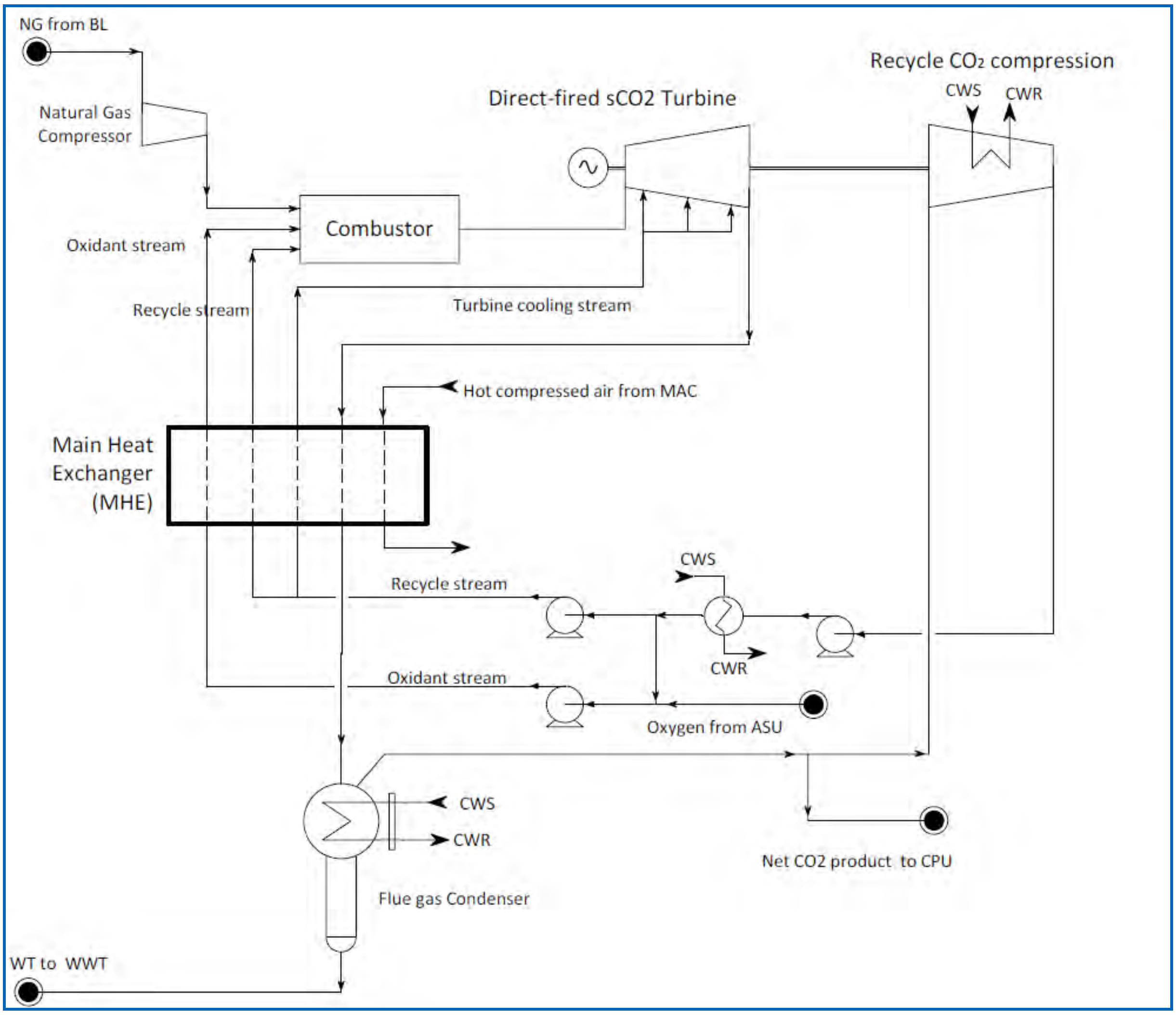


Figure 4. NET Power cycle simplified scheme

2.1.3. GRAZ

The S-Graz Cycle consists of a high-temperature cycle, including the gas turbine and associated compressors and combustion chamber, the HRSG, a high pressure steam turbine (back-pressure type) and a low temperature cycle, whose design parameters vary depending on the different cycle configuration proposed for the GRAZ cycle, substantially including a low pressure turbine and condenser.

The fuel along with the nearly stoichiometric mass flow of oxygen is fed to the combustion chamber, which is operated at a pressure of 40-50 bar. The steam generated in the HRSG and the recycled gas mixture of water and carbon dioxide are used to cool the burners and the turbine blades.

The working fluid, mainly composed of steam (85-89%mol) and carbon dioxide (10-14%mol), is expanded to a pressure slightly above the atmospheric pressure. The exhaust flue gas is sent to a single pressure level HRSG, generating high pressure steam (around 170-180 bar) to be expanded in a back pressure steam turbine down to the pressure level required for steam injection in the gas turbine expander for blade metal temperature control.

The cooled gas from the HRSG is divided into two separate streams: part is recycled back to the combustion chamber for combustion temperature control, after being further cooled and compressed, while the remaining portion is sent to the low temperature section.

The low temperature section depends on the selected configuration; two different schemes are analysed in the present study:

- **Option a)** is based on the original S-GRAZ scheme shown in Figure 5, as proposed during the ASME Turbo Expo 2005 [1]
- **Option b)** is based on the modified S-GRAZ scheme shown in Figure 6, as proposed during the ASME Turbo Expo 2008 [2]

In the S-GRAZ cycle, the fraction of flue gas from the HRSG not recycled back to the combustor through gas turbine compressors is expanded down to the vacuum conditions and condensed against cooling water in a flue gas condenser. The heat recovery steam generation is installed downstream the expansion stages to atmospheric pressure, upstream the last expansion stage for several reasons:

- The HRSG is conventional, with the flue gas at a pressure level slightly above the atmosphere.

[1] W. Sanz, H. Jericha et al, *A further step towards a Graz cycle power plant for CO_2 capture*, ASME Turbo Expo 2005, Reno-Tahoe, Nevada, US

[2] H. Jericha, W. Sanz et al., Design details of a 600 MW Graz cycle thermal power plant for CO_2 capture, ASME Turbo Expo 2008, Berlin, Germany

- Expanding the flue gas directly to vacuum conditions with no heat recovery would lead to a discharge temperature not reasonable for water condensation, but not high enough for the generation of the super-heated high pressure steam required for the back-pressure steam turbine.
- Only the fraction of flue gas not required as recycle stream is expanded further down to the condenser pressure, reducing the power production but also the power consumption required for re-compression and the condenser size and cooling water requirement.

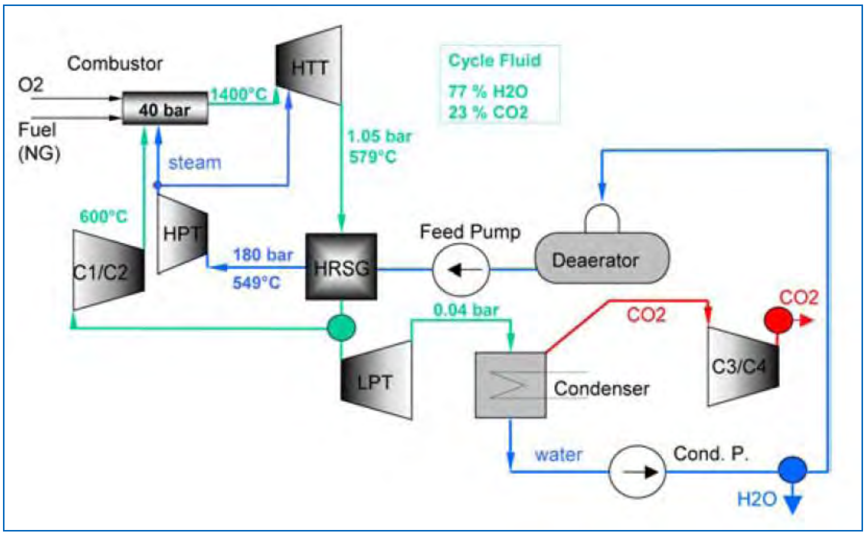


Figure 5. Option a) S-GRAZ simplified scheme [1]

In the modified S-GRAZ cycle, the fraction of flue gas from the HRSG not recycled back to the combustor through the gas turbine compressors is compressed up to around 2 bar, while recovering the compression heat for low pressure (below atmosphere) super-heated steam generation. Steam is expanded in a steam turbine, condensing type, increasing power generation (refer to below section 0).

In this cycle, the low pressure section is modified mainly to reduce the very large condenser size, both in terms of heat exchange area and cross-sections, mainly related to the huge flue gas flow to the condenser and the reduced low heat transfer coefficients due to the CO_2 presence. In this configuration the condenser is not critical equipment, but is a conventional water cooled steam condenser.

[1] W. Sanz, H. Jericha et al, *A further step towards a Graz cycle power plant for CO_2 capture*, ASME Turbo Expo 2005, Reno-Tahoe, Nevada, US

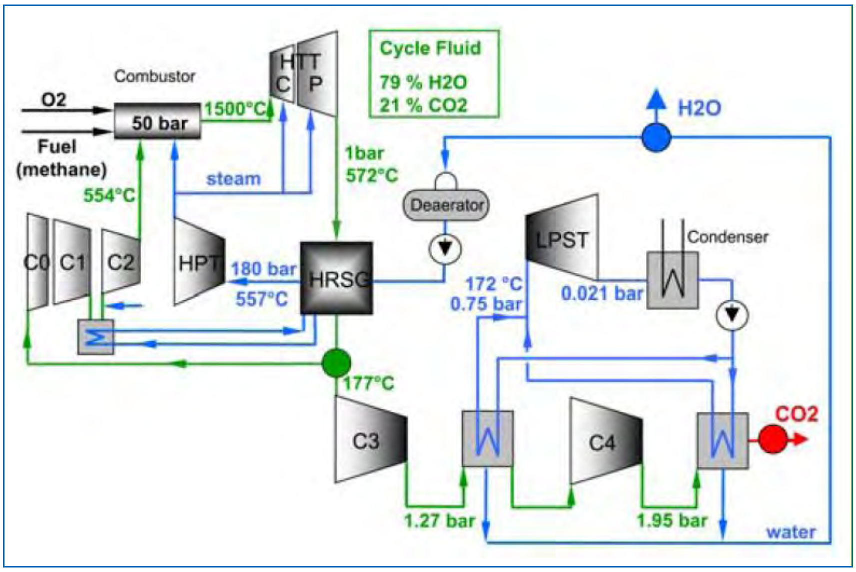


Figure 6. Option b) Modified S-GRAZ simplified scheme [1]

For more detailed information on the S-Graz cycle configuration and equipment, reference shall be made to ATTACHMENT A.1.

[1] H. Jericha, W. Sanz et al., Design details of a 600 MW Graz cycle thermal power plant for CO_2 capture, ASME Turbo Expo 2008, Berlin, Germany

2.1.4. *CES*

This cycle, proposed by Clean Energy Systems (CES) is based on the use of water both in vapour and liquid phases as combustion temperature moderator.

Even though different variations of the CES cycle have been proposed, the cycle mainly consists of a high pressure oxy-fuel combustor where part of the fuel and oxidant are combusted, utilizing preheated liquid water and/or steam as temperature moderator. Hot gas produced in the gas generator is expanded in an uncooled HP steam turbine.

Working fluid, discharged at around 30-60 bar, is reheated to very high temperature by a supplementary oxy-fuel combustion, and expanded in an LP turbine.

The several cycle variations proposed through the years of technology development, varying the condition of the HP combustion moderator stream, number and pressure conditions of the combustors, the cooling medium for the LP turbine (steam or a fraction of the flue gas from the first combustor) and/or the discharge pressure from the LP turbine (atmospheric or condenser pressure).

Three different schemes are analysed in the present study, the latter (Option c) being the long-term solution for which CES claims the best efficiency:

- **Option a)** is based on the scheme proposed by DOE/NETL [1] shown in Figure 7, based on a LP turbine discharging at condenser pressure (0.1 bar) and using a fraction of flue gas as cooling stream
- **Option b)** is based on the revised scheme proposed by CO_2-Global at a Workshop on Future Large CO_2 Compression Systems [2] shown in Figure 8, based on a LP turbine discharging at atmospheric pressure and related HRSG where steam to be used as cooling stream is generated.
- **Option c)** is based on the configuration suggested by CES, based on three combustors gas turbine and steam at supercritical conditions as moderator stream for the combustor and HP turbine coolant stream, as shown in Figure 9. The LP turbine discharge is at vacuum pressure condition as per case 4a. The cooling stream for second and third combustor is part of the flue gas from the upstream turbine section (respectively HPT and MPT).

[1] DOE/NETL, 2010. Carbon Capture Approaches for Natural Gas Combined Cycle Systems. Report DOE/NETL-2011/1470, Revision 2, December 20, 2010

[2] Carl-W. Hustad, CO_2-Global, CO_2 Compression for Advanced Oxy-Fuel Cycles At Workshop on Future Large CO2 Compression Systems, DOE Office of Clean Energy Systems, EPRI, and NIST, 2009 Gaithersfield, MD

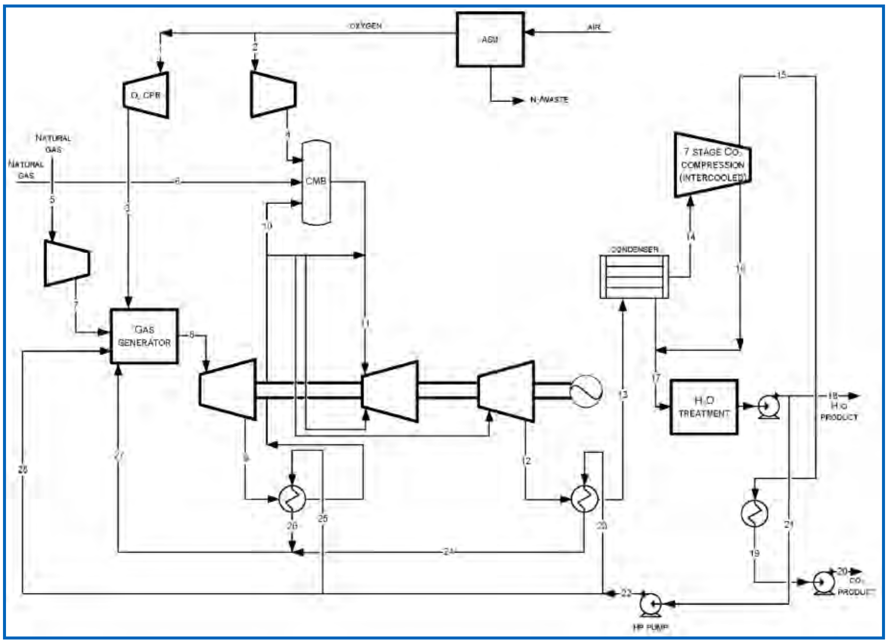


Figure 7. Option a) CES cycle scheme [1]

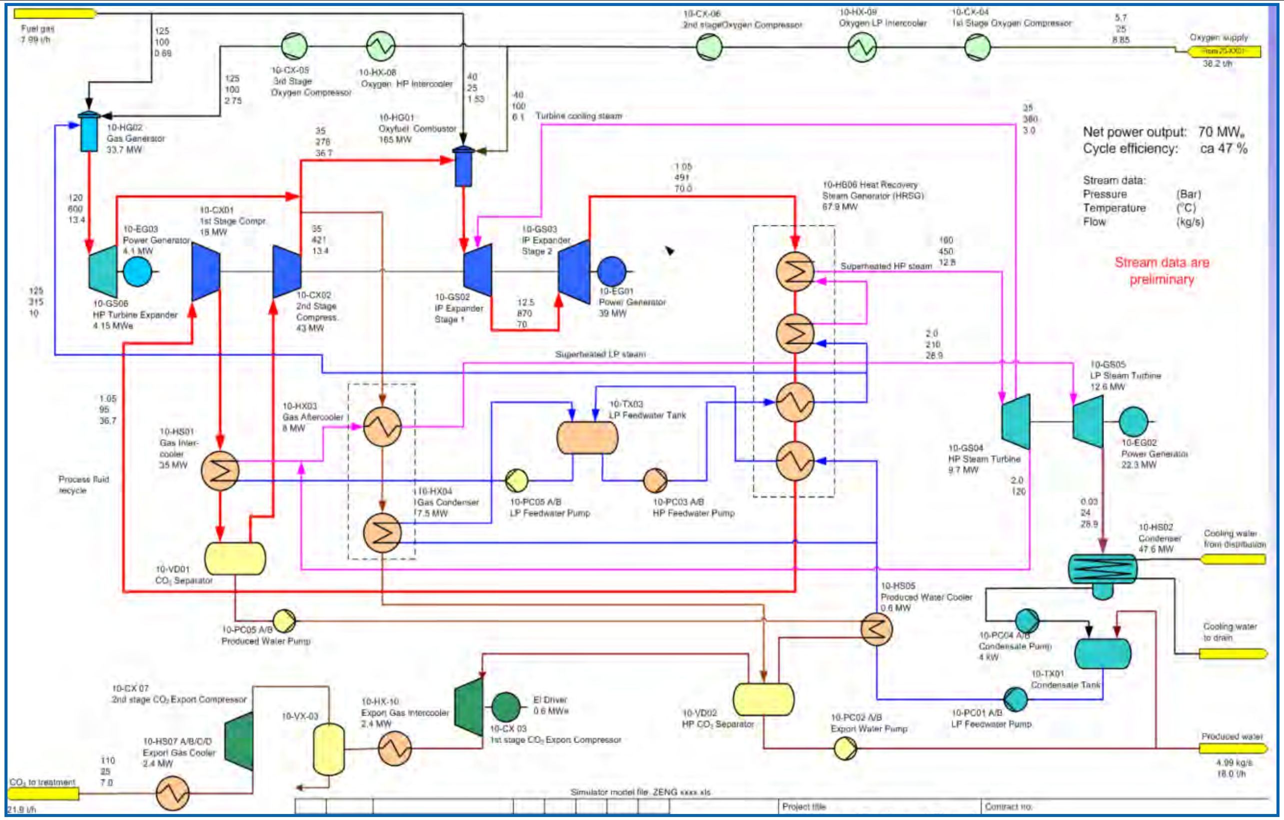


Figure 8. Option b) Revised CES cycle scheme [1]

[1] Carl-W. Hustad, CO_2-Global, CO_2 Compression for Advanced Oxy-Fuel Cycles At Workshop on Future Large CO_2 Compression Systems, DOE Office of Clean Energy Systems, EPRI, and NIST, 2009 Gaithersfield, MD

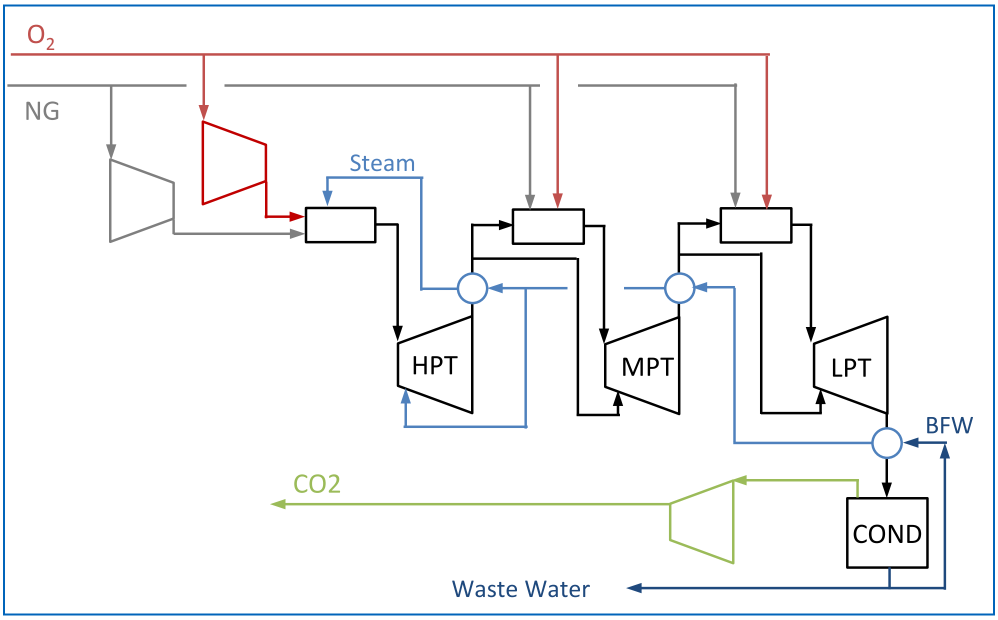


Figure 9. Option c) Supercritical CES cycle scheme

2.2. Low pressure steam cycle

In some of the proposed configurations, heat at low temperature (in the range of 220-270°C) is made available at the end of the flue gas heat recovery section downstream the expander.

In order to enhance the overall plant efficiency and reduce the cooling water consumption, this low value heat is used to generate steam at low pressure level (below atmosphere), to be expanded in a steam turbine.

This low pressure steam cycle is mainly composed of:

- Steam turbine, water-cooled and condensing type, and related condenser;
- Condensate preheater, boiler feed water economiser, steam generator, steam super-heaters;
- Deaerator;
- Boiler feed water and condensate pumps.

Depending on the oxy-cycle, the above listed heat-exchangers are installed either in the warm end of the heat recovery section downstream the expander or used as intercoolers in the flue gas re-compression section.

2.3. Carbon dioxide compression and purification

The purpose of this section is to cool, dry, compress and purify to the required level the product CO_2 stream from the indirect contact cooler before sending it to the pipeline, outside plant battery limits.

The CO_2 purification and compression unit consists of the following main sections:

- Raw gas compression.
- TSA unit.
- Auto-refrigerated Inerts Removal, including distillation column to meet the required oxygen specification in the CO_2 product.
- Final compression up to 110 bar.

Raw gas compression

The following description refers to the simplified scheme shown in Figure 10.

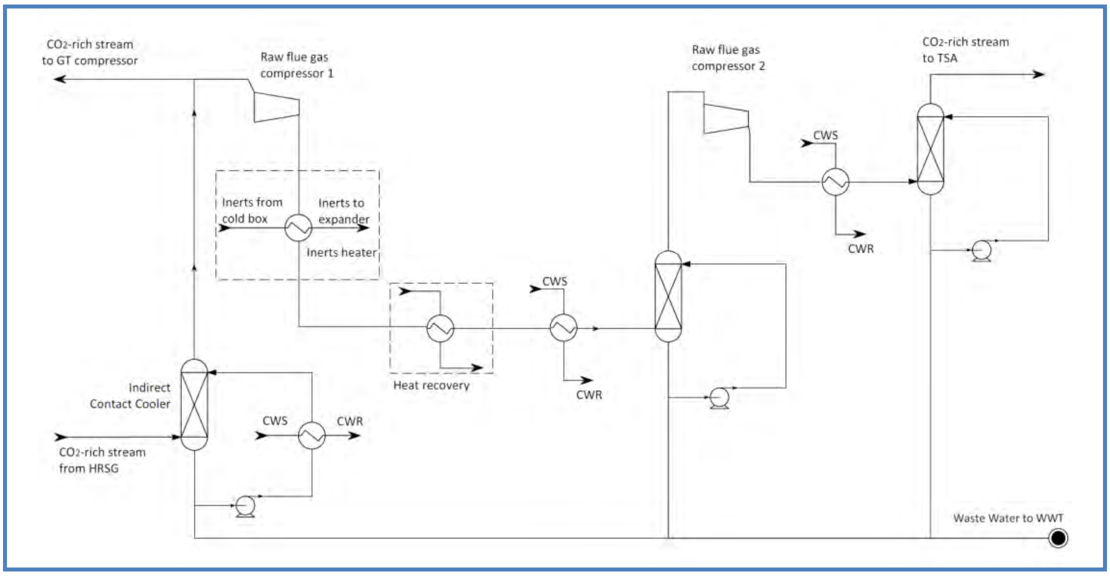


Figure 10. Raw gas compression scheme to 30 bar

The CO_2 stream entering the CPU is compressed adiabatically to 15 bar, producing a stream of compressed impure carbon dioxide at about 300°C. Such stream is used to preheat the vent stream from the downstream inerts purification section and, depending on the oxy-turbine cycle, for fuel, oxygen and boiler feed water pre-heating. Final cooling is made against a stream of cooling water to produce a stream of CO_2 at about 26°C.

The CO_2 stream is fed to the bottom of a first contacting column, where it ascends and contacts counter-currently a stream of descending acid water. The column is designed to provide sufficient contact time between the ascending gas and the descending liquid to convert part of NO_x to nitric acid. Thus, a stream of carbon dioxide is removed from the top of the column and a stream of flue gas condensate that also contains some nitric acid is removed from the column bottom. The liquid is then pumped and split into two: part of the liquid is cooled down and recycled to the same contacting column as reflux, whereas the excess of liquid is sent to the Waste Water Treatment unit.

The stream of carbon dioxide from the top of the first contacting column is compressed to about 30 bar by an integrally geared centrifugal compressor. Heat of compression generated in the compression stage is removed by means of a cooling water exchanger in order to produce a stream of cooled, compressed carbon dioxide, which is fed to the bottom of the second contacting column.

The gas stream ascends the column and contacts counter-currently a stream of aqueous nitric acid solution. The column is designed to provide sufficient contact time between the ascending gas and the descending liquid to almost completely convert the remaining NO_x contaminant to produce nitric acid. The lean carbon dioxide stream is removed from the top of the column and a stream of aqueous nitric acid is removed from the column bottom. The liquid is then pumped and divided into two: part of the liquid is cooled down and recycled to the same contacting column as reflux, whereas the excess of liquid is sent to the Waste Water Treatment unit. A stream of fresh water is injected into the top of the column to increase NO_x conversion and to ensure that no acid droplets are entrained in the gas stream leaving the column top.

TSA system

The raw CO_2 gas passes through a thermally regenerated dual bed desiccant dryer to lower the dew point below -55°C before entering the auto-refrigerated inerts removal section. This desiccant dryer system prevents ice formation which could cause a blockage in the cold box as well as causing corrosion in the pipeline.

Auto-refrigerated inerts removal

The inerts removal process is based on the principle of phase separation between condensed liquid CO_2 and insoluble inerts gas at a temperature of –55°C, which is very close to the triple point, or freezing temperature, of CO_2.

The actual CO_2 pressure levels and the configuration selected for the separation are fixed by the CO_2 purity and recovery specification requirements. The inerts removal process configuration is mainly affected by the oxygen specification in the CO_2 product of 100 ppm, which implies the installation of a distillation column.

The following description refers to the simplified scheme shown in Figure 11. Numbers in brackets refer to the stream tag in the figure.

The CO_2 feed gas pressure is around 30 bar. The necessary refrigeration for plant operation is obtained by evaporating liquid CO_2 at a pressure around 16-17 bar and

5.6 bar and compressing these two low pressure gas streams in the main CO_2 product compressor to the final pipeline delivery pressure of 110 bar.

The dry gas from the TSA unit (102) is fed to the cold box and is cooled with the returning stream evaporating and superheating CO_2 streams and the waste streams in the main exchanger, then it is used as heating medium in the distillation column reboiler (E106). The main heat exchangers are multi-stream plate-fin aluminium blocks.

The stream from the reboiler (105) is further cooled and it partially condenses (106) and is passed to the flash drum. The vapour from the separator (107), containing the separated inerts together with some CO_2, is sent back through the heat exchangers for a first pre-heating. This stream of inerts (108), which is at a pressure of 30 bar, is then heated (heating medium vary from case to case) and is expanded in a power recovery turbo-expander (K103) before being vented (110).

The liquid stream from the separator (111), at 30 bar, is heated in the second main heat exchanger and is then expanded through a valve to 16-17 bar (V103), before entering the distillation column. The vapour stream exiting the distillation column (114), which still contains a large portion of CO_2, is heated through both the main heat exchangers, re-compressed to 30 bar, cooled against cooling water and finally recycled to the dry gas feeding the cold box (117).

The liquid stream exiting the bottom of the distillation column (118) is split into two streams which are both expanded through a valve to two different pressure levels and heated up in the main heat exchangers, providing the necessary refrigeration.

Final compression stage

The CO_2 vapour stream leaving the first main heat exchanger at 5.6 bar (121) is then compressed in an integrally geared compressor to the same pressure as the second CO_2 stream (126) (around 16-17 bar). The two streams are combined and compressed in two intercooled stages (K101) to the required pressure of 110 bar.

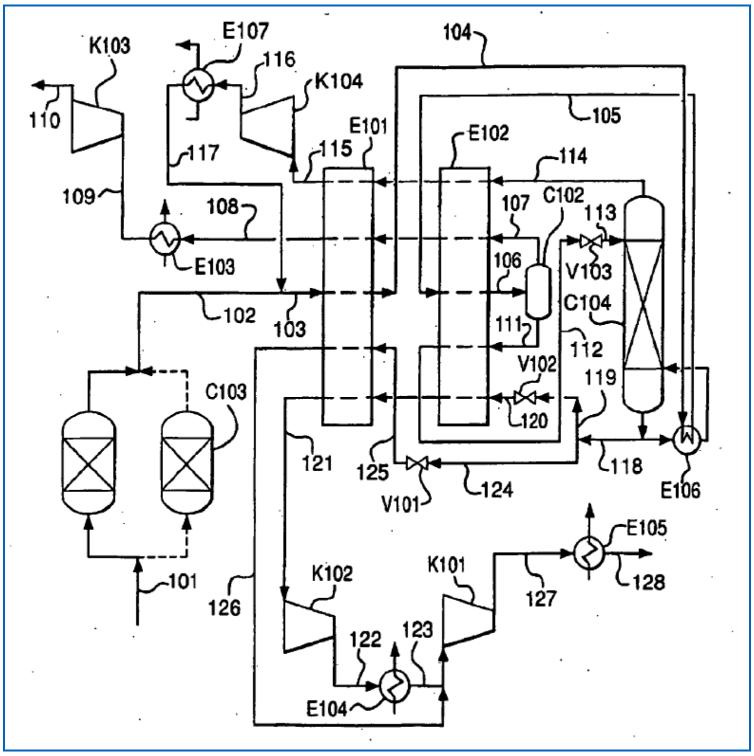


Figure 11. Auto-refrigerated inerts removal process (from EP 1 953 486 B1)

2.3.1. *High-purity oxygen cases*

When high purity oxygen (99.5%mol) is used as oxidant stream in the combustion chamber, the inert content in the CO_2-rich stream sent to the CPU is lower (around 2.5% dry basis) than that of the cases with 97% oxygen purity (around 7% dry basis). In this case, the main source of inert components (i.e. nitrogen) is the natural gas.

As a consequence, the design of the auto-refrigerated inerts removal section is modified by removing the first flashing vessel, while the distillation column is still required to meet the CO_2 purity specification (i.e. oxygen content), as further detailed below.

The dry gas feed from the TSA unit, after being cooled and used as heating medium in the reboiler, when expanded through a valve to 16-17 bar, is directly fed to the distillation column, with no intermediate flash drum.

The vapour from the distillation column, which still contains a large portion of CO_2, is heated via the main heat exchangers, re-compressed to 30 bar, cooled against cooling water and finally partially recycled to the dry gas feeding the cold box.

The remainder of the inerts is then heated (heating medium vary from case to case) and expanded in a power recovery turbo-expander before being vented.

The liquid stream exiting the bottom of the distillation column is split into two streams, which are both expanded through a valve to two different pressure levels and heated up in the main heat exchangers, providing the necessary refrigeration duty.

2.4. Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and is designed to produce oxygen at either 97% mol. or 99.5% mol. purity, depending on the specific case. For this study case a generic ASU has been simulated with no reference to a specific supplier. Different oxygen delivery pressures have been considered to match the specific requirement of each cycle.

The amount of oxygen required for the oxy-combustion gas turbine power plants considered in this study is around 10,900 tons/day (minor difference exists depending on the oxy-fuel cycle). The configuration proposed for this study case is based on two (2 x 50%) cryogenic trains, each sized for 5,500 tons/day. This is within the range of ASU currently being commercially offered. For the main air compressor, 2x50% train configuration is considered to achieve a low and efficient minimum turndown. Due to the high reliability of the air compressor machine, no sparing equipment is foreseen.

The cycle chosen is one in which gaseous oxygen (GOX) is produced by boiling liquid oxygen (LOX) which is ideally suited to this application as no nitrogen production is required.

The ASU configuration typically proposed for oxy-combustion application is based on three-pressure levels distillation columns:

- The conventional double column includes the low pressure column (1.5 bar) with its reboiler integrated with the condenser of the high pressure column (5-6 bar). The column pressures are set to give a temperature driving force in the reboiler/condenser.
- An extra column is added operating at intermediate pressure (around 3 bar). The condenser for this column also integrates with a reboiler in the low pressure column but at a lower temperature, boiling a liquid stream higher up within the low pressure column.

For an oxygen purity specification lower than 99.5%mol, no additional column for O_2/Ar separation is considered. Additional number of plates and power consumptions increases with the purity level required to overcome the lower relative volatility of the streams in the distillation columns.

This arrangement minimises the amount of feed air that must be compressed to the higher pressure of high pressure column condenser, leading to the low power requirement of the whole unit.

With reference to the simplified block flow diagram shown in Figure 12, the plant includes the following main sections.

Compression system

Process air is cleaned from dust and particulate matter through an intake air filter before being fed to the main air compressor (MAC). A cooling water intercooled centrifugal compressor is used to compress the feed air.

Adsorption front end air purification system

Before air is cooled to cryogenic temperatures in the main heat exchanger, water vapour, carbon dioxide and other trace impurities are removed in order to avoid the cryogenic equipment blockage.

The selected configuration includes two purification systems based on dual bed adsorbers: one system after the first air compression stage for feed to the intermediate pressure column and the other after the last compression stage to the high pressure column pressure.

The adsorber operates on a staggered cycle, i.e one vessel adsorbing and the other being reactivated. The adsorbents generally used consist of layers of alumina or silica gel plus layers of zeolite. The adsorber vessels are vertical cylindrical units having annular adsorbent beds.

Cold box

Both the intermediate and high pressure air streams exiting the two adsorbent systems are split in two. These four streams are fed directly to the main heat exchanger, consisting of several parallel aluminium plate-fin heat exchanger blocks manifolded together.

The first intermediate pressure stream is cooled close to its dew point and fed to the bottom of the intermediate pressure column. Downstream of the main heat exchanger the second intermediate pressure stream is expanded in a centrifugal expansion turbine providing the power for the centrifugal compressor, providing the air feed to the high pressure column. The expanded air is fed to the middle of the low pressure column in order to provide refrigeration for the operation of the ASU.

The first high pressure stream is cooled close to its dew point and fed to the bottom of the high pressure column while the second high pressure air stream is cooled and condensed in the main heat exchanger against boiling oxygen. The resulting liquid air from the main exchanger is fed to the middle of both the high pressure and intermediate pressure columns.

In the high and intermediate pressure columns, the gaseous air feed is separated into an overhead nitrogen vapour and an oxygen-enriched bottom liquid. The nitrogen vapour from the high pressure column is condensed against boiling oxygen in the low pressure column sump, providing the liquid reflux for both the high and the low pressure columns. Boiling oxygen in an upper stage of the low pressure column provides the condensing medium also for the nitrogen from the intermediate pressure column is condensed. The resulting liquid nitrogen stream provides the reflux stream for both the low pressure and the intermediate pressure columns.

Liquid oxygen from the bottom of the high and intermediate pressure columns is cooled in the subcooler against waste nitrogen and is flashed to the low pressure column as intermediate feeds. The feeds to the low pressure column are separated into a waste nitrogen overhead vapour and a liquid oxygen bottom product, which reaches the required purity of 97% by volume.

The waste nitrogen is withdrawn from the top of the low pressure column and warmed in the subcooler and the main heat exchanger. A portion of the nitrogen stream from the main exchanger is used for adsorber reactivation. The remaining dry nitrogen is vented through a Chilled Water Tower to produce chilled water by evaporative cooling.

Pure liquid oxygen is withdrawn from the bottom of the low pressure column and returned to the main heat exchanger where it is vaporised and warmed up to ambient conditions against boosted air feed to the columns. The gaseous O_2 is then regulated and supplied to the power plant.

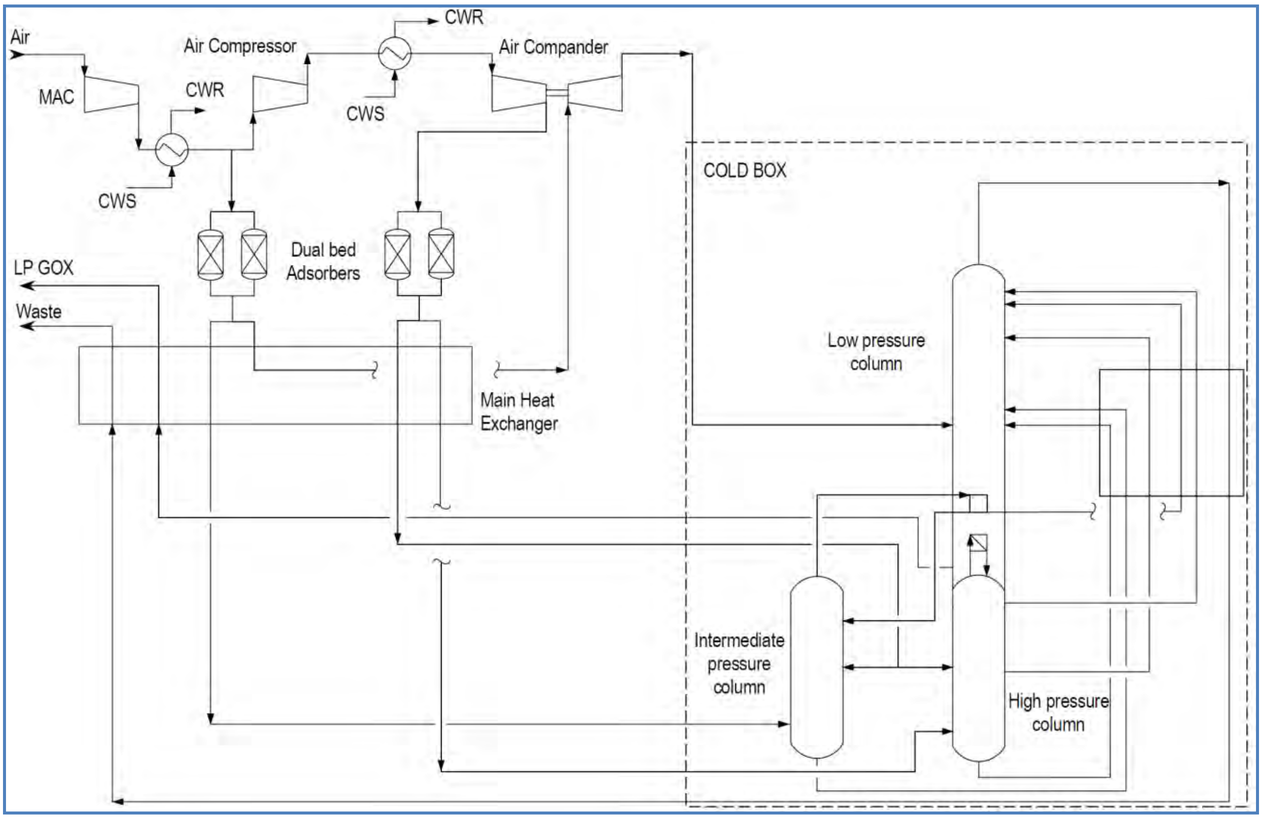


Figure 12. ASU simplified scheme

Oxygen back-up

Oxygen back-up, corresponding to three hours oxygen production from one ASU is considered to enhance the ASU reliability. Backup is in the form of liquid oxygen (LOX) at a pressure of 2.5 bar in a vacuum insulated storage tank, common to all trains.

2.5. Utility and Offsite units

2.5.1. *Cooling water*

The cooling water system consists of raw water in a closed loop, with a natural draft, evaporative cooling tower, equipped with single-stage vertical water pumps.

The maximum allowed cooling water temperature increase is 11°C. The blow-down is used to prevent the concentration of dissolved solids from increasing to the point where they may precipitate and scale-up heat exchangers and the cooling tower fill. The design concentrations cycles (CC) is 4.0.

One concrete tower is considered, with a basin diameter around 110 m and 180 meters high. The tower is equipped with two distribution systems, one primary distribution system supplying water from a concrete duct and one secondary system from PVC pipes equipped with sprayers, connected to the concrete ducts. Tower filling, with vertical channels, increases the cooling and thermal efficiency, allowing pollutants to be easily washed through. Drift eliminators guarantee a low drift rate and low pressure drop. To avoid freezing in winter ambient conditions, the fill pack is divided into zones to allow step by step reduction of cooling capacity while maintaining an excellent water distribution and spray sprinklers are installed to create a warm water screen on the air inlets to preheat the ambient air when freezing ambient conditions occurs.

2.5.2. *Natural gas metering and conditioning station*

Natural gas at 70 bar from network is filtered and metered and let-down (or compressed) to the operating conditions required by the gas turbine.

The fuel will be metered by a fiscal meter (two fiscal meters, one in operation and the other one in stand-by) and the gas pressure will be reduced to match the required values for the gas turbine. If required by the cycle configuration, part of the natural gas is taken upstream the let-down station and sent to a dedicated compressor to meet the turbine requirement. In order to avoid freezing or condensation issues, a preheating section is provided upstream the reduction station.

Filtering and metering section and let-down station will be based on 3 x 50% configuration, two equipment in normal operation and one spare in active stand-by, to ensure reliability and on-line maintenance.

2.5.3. *Raw and Demineralised water*

Raw water is generally used as make-up water for the power plant, in particular as make-up of the cooling tower. Raw water is also used to produce demineralised water. Raw water from an adequate storage tank is pumped to the demineralised water package that supplies make-up water with adequate physical-chemical characteristics to the thermal cycle and to the hydrated lime preparation unit.

The treatment system includes the following:

- Filtering through a multimedia filter to remove solids.
- Removal of dissolved solids: filtered water passes through the Reverse Osmosis (RO) cartridge filter to remove dissolved CO_2 and then to a reverse osmosis system to remove dissolved solids.
- Demineralised water production: an electro de-ionization system is used for final polishing of the water to further remove trace ionic salts of the Reverse Osmosis (RO) permeate.

Adequate demineralised water storage is provided by means of a dedicated demineralised water tank.

The demineralised water make-up supplies the make-up water to the thermal cycle, whilst the demineralised water distribution pump supplies demineralised water to the other plant users or to the plant circuits for first filling.

2.5.4. *Firefighting system*

This system consists of all the facilities able to locate possible fire and all the equipment necessary for its extinction. The fire detection and extinguishing system essentially includes the automatic and manual fire detection facilities, as well as the detection devices with relevant alarm system. An appropriate fire detection and suppression system is considered in each fire hazard area according to the applicable protection requirements. The fire fighting water is supplied by a water pumping station via a looping piping network consisting in a perimetrical circuit fed by water pumped from the cooling tower basin.

2.5.5. *Instrument and plant air system*

The air compression system supplies air to the different process and instrumentation users of the plant.

The system consists mainly of:

- Air compressors, one in operation, one in stand-by.
- Compressed air receiver drum .
- Compressed air dryer for the instrument air.

The ambient air compressed by means of the air compressor is stored in the air receiver in order to guarantee the hold-up required for emergency shutdown.

Plant air is directly taken from the air receiver, while air for instrumentation is sent to the air dryer where air is dried up to reach an adequate dew point, to ensure proper operation of the instrumentation.

2.5.6. *Waste Water Treatment*

All the liquid effluents generated in the plant are treated in the wastewater treatment system in order to be discharged in accordance with the current local regulations.

The following description gives an overview of the waste water treatment configuration, generally adopted in similarly designed power plants; it includes a preliminary identification of the operations necessary to treat the different waste water streams generated in the power plant.

The Waste Water Treatment unit is designed to treat the following main waste water streams:

- Flue gas condensate
- Potentially oil-contaminated rain water
- Potentially dust-contaminated rain water
- Clean rain water
- Sanitary waste water.

Mainly, the above streams are collected and routed to the waste water treatment in different systems according to their quality and final treatment destination.

The WWT system is equipped mainly with the following treatment sections:

- Treatment facilities for the flue gas condensate
- Treatment facilities for the potentially oily contaminated water
- Treatment facilities for the potentially dust contaminated water
- Treatment facilities for not contaminated water
- Treatment facilities for the sanitary wastewater.

Flue gas condensate

Condensate from the flue gas is treated in dedicated section of the Waste Water Treatment to remove the slight acidic component and maximise water recovery. The following alternative treatment can be considered:

- Neutralisation
- Resins
- Reverse osmosis

At this study level, it is not possible to identify the optimum solution for the study case. Main parameters affecting the selection and the severity of the treatment are the recovered water utilisation and destination of the blow-downstream to be discharged (i.e. river or sea).

Potentially Dust Contaminated Water Treatment

Rain water and washing water from areas subject to potential dust contamination is treated in apposite water treatment systems prior to be sent to the "potentially oil contaminated" treatment system.

In particular, they are collected in a dedicated sewer, sent to a lamination tank and then to a chemical/physical treatment to remove the substances that are dissolved and suspended. The system includes also a neutralization system to modify potential acidity and/or alkalinity of washing water used for the air pre-heaters.

Potentially Oil-Contaminated Water Treatment

Potentially oil-contaminated waters are:

- Washing water from areas where there is equipment containing oil.
- Rain water from areas where there is equipment containing oil.

After being mixed with treated water coming from "potentially dust contaminated" system, water is treated in a flotation and filtration system, where emulsified oil and suspended solids are respectively separated.

Treated effluent water will have the characteristics to respect the local regulations so that it can be consequently discharged.

Not Contaminated Water Treatment

Rainwater fallen on clean areas of the plant, such as roads, parking areas, building roofs, areas for warehouse/services/laboratory etc. where there is no risk of contamination, will be collected and disposed directly to the water discharge system.

A coarse solids trap is installed upstream the discharge point in order to retain coarse solids that may be carried together with the discharge water.

Sanitary Water Treatment

The sanitary waste water streams discharged from the different sanitary stations of the plant will be collected in a dedicated sewage and destined to the Sanitary Water Treatment system. This section generally involves the following main water treatment operations:

- Primary sedimentation for coarse solids removal.
- Biological treatment for BOD removal.
- Filtration for residual organic matter and suspended solids separation.
- Disinfection for bacteria inhibition.

ATTACHMENT A.1: S-Graz cycle

Information on the

Graz-Cycle – Oxy-Turbine Package

Project Name: Oxy-Turbine Power Plant

Client: IEAGHG

4.1 Performance

		S-Graz	Modified S-Graz
Fuel			
Pressure	Bar a	41.7	41.7
Temperature	°C	150	150
Quantity	MWth (LHV)	142.305	745.076
Flowrate	t/h	11.017	57.68
Recycle Gas conditions			
Composition			
CO2	Vol-%	10.8458	9.8285
H2O	Vol-%	88.0551	89.1238
N2	Vol-%	0.3517	0.3250
O2	Vol-%	0.2839	0.2976
Ar	Vol-%	0.4635	0.4251
Pressure	Bar a	1.01	1.01
Temperature	°C	96.4	105
Quantity	t/h	165.68	686.97
Oxygen			
Composition			
Pressure	Bar a	41.7	41.7
Temperature	°C	150	150
Quantity	t/h	41.205	216.186
Cooling Steam			
Burner			
Pressure	Bar a	41.7	41.7
Temperature	°C	329.6	287.8
Quantity	t/h	40.35	302.57
Turbine HTT (stage1,2)			
Pressure	Bar a	41.7	41.7
Temperature	°C	329.6	287.8
Quantity	t/h	23.13	98.88
Turbine HTT (stage 3,4)			
Pressure	Bar a	41.7	13.78
Temperature	°C	210.2	194.2
Quantity	t/h	11.654.2	51.385

Exhaust gas			
Composition			
CO2	Vol-%	10.8458	9.8285
H2O	Vol-%	88.0551	89.1238
N2	Vol-%	0.3517	0.3250
O2	Vol-%	0.2839	0.2976
Ar	Vol-%	0.4635	0.4251
Pressure	Bar a	1.053	1.06
Temperature	°C	569.9	575.1
Quantity	t/h	294.45	1421.2
Gross power output			
Difference HTT-Compressor	MW	76.6	405.6
Plant	MW	90.8	471.6
Net Plant Efficiency	%	63.84	63.3

Comment: All Power and mass flow values can be scaled to the intended power output of the plant!

4.2 Technical Information

This information is taken from the ASME Paper GT2006-90032:

DESIGN CONCEPT FOR A VERY LARGE GRAZ CYCLE PLANT OF 400 MW NET OUTPUT

In this work the design concept for a Graz Cycle power plant of 400 MW electrical net output is presented. This power is derived from a 490 MW turbo shaft configuration. The difference is caused by the power demand of the ASU and by the driving power for the oxygen compressor in order to deliver oxygen to the combustor at 42 bar and by the CO_2 compressor which has to deliver the captured CO_2 at a pipeline pressure of over 100 bar.

Gas turbines, compressors and combustors require the best flow development achieved up to now in gas turbine technology. In the course of this project our institute has found novel solutions for blade cooling, steam cooled combustor burner design and optimal rotor construction and rotor dynamics. The innovative cooling burner design helps to achieve the mentioned extreme high thermal efficiency (see [6] for details of the burner design), further improved by the positive change on the lower temperature end of the power cycle flow scheme as described above.

This type of rotor design provides for high blade load carrying capability with acceptable radial stress. The newly developed high chromium ferritic steels will be applied making use of their superior heat conduction and low thermal expansion properties. The relatively high speed selected provides for long blades in the last stages with high flow efficiency and low tip clearance loss.

The one-shaft system as in air-breathing gas turbines is not applicable since in the Graz Cycle system the amount of compressor flow volume is smaller and the number of stages required considerably higher. Therefore a much higher compressor speed as power turbine speed is an effective solution.

The main gas turbine components are arranged on two shafts, the compression shaft and the power shaft (see Fig. 4). The compression shaft consists of the cycle compressors C1 and C2, which are driven by the first part of the high temperature turbine HTT, the compressor turbine HTTC. It runs free on its optimal speed of 8500 rpm. This relatively high speed is selected for reason of obtaining sufficient blade length at outlet of C2 and to reduce the number of stages in both compressors. The second part of the HTT, the power turbine HTTP, delivers the main output to the generator. A further elongation of the shaft is done by coupling the four-flow LPST at the opposite side of the generator. The HPT can be coupled to the far end of the LPST or can drive a separate generator. The two shafts are based on the same spring foundation. The intercooler between C1 and C2 is located on the fixed foundation.

In this design proposal intensive use of steam cooling is made, not only for blades, but for all rotors in the high-speed high-temperature region. In that manner a solid and simple rotor design forged from one piece or welded from separate disks can be used with no internal friction between rotor disks as might be possible in a rotor assembled from separate disks.

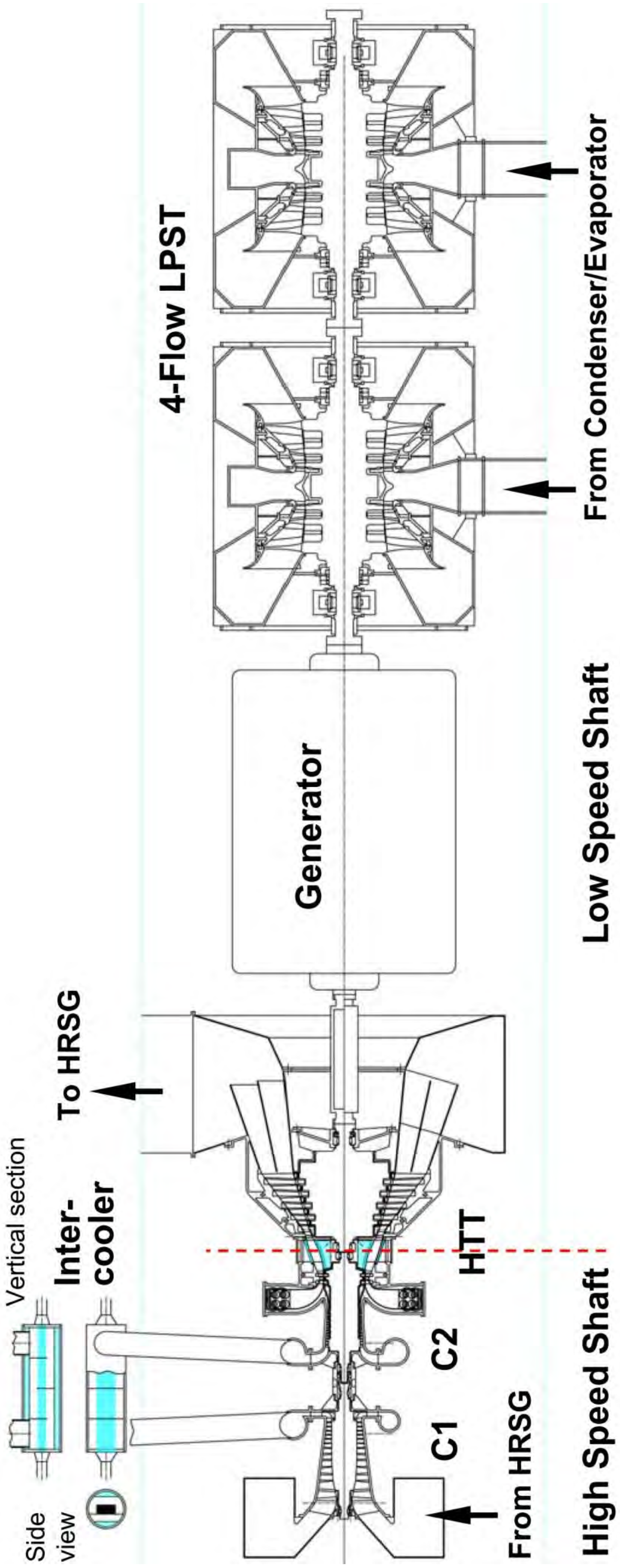


Fig. 4: Arrangement of the main turbomachinery for a 400 MW Graz Cycle plant

C1/C2 compressor design with intercooler:

The working fluid compressor C1 is driven by the HTTC at 8500 rpm. The high speed poses a special problem for the first stage of C1 which has yet been solved by flow research and is now applied in many aircraft jet engines and also stationary compressor designs [14, 15]. The high tip Mach number on the first stage should not exceed the value of 1.4 for reasons of shock formation. With the help of a slight positive inlet swirl an inlet Mach number of 1.3 is designed.

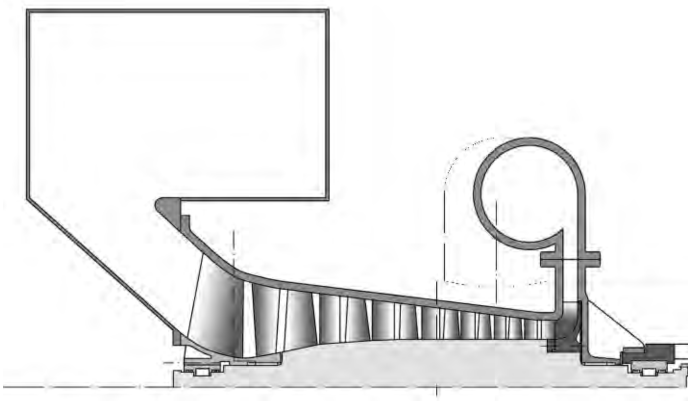


Fig. 5: C1 design with an uncooled drum rotor and an additional radial stage from nickel alloy, with radial diffuser and exit scroll to intercooler

Compression at C1 starts at 106°C and reaches 442°C at the outlet to the intercooler. For reasons of rotor dynamics the shafts of C1 and C2 are separated with intermediate bearings and a solid coupling. This makes the transfer of cooling flow difficult, so that cooling of the drum rotor of C1 will not be applied. This is possible by a combination of rotor materials.

The first part with seven axial stages is a ferritic steel drum, which reaches only 390°C. This material can be highly stressed without creep at temperatures below 400°C. By the application of a final radial wheel which has to be milled separately from nickel alloy and which is mounted to the main drum by elastic centering completes the rotor construction of C1, as shown in Fig. 5. The radial wheel with a wide vaneless diffuser and scroll improves the flow transfer to the intercooler.

The inlet temperature to C2 is somewhat lowered by the intercooler but still reaches 380°C. During course of compression the working fluid reaches an outlet temperature of 580°C, so that from the second stage onwards cooling has to be applied on the rotor surface of the bladed annular flow channel. Seven axial stages with a stepwise decrease of blade length from 90 to 40 mm are supported on a drum rotor with disk extensions of constant diameter. Fig. 6 shows the C2 rotor with the counter flow of cooling steam on the drum surface. It is guided by means of openings under the bladed disk extensions and is prevented by sealing strips from flowing into the main flow. These strips are carried on both sides of the stationary blades. By proper selection of the feed pressure this flow can be optimized at a small penalty in dilution of the main flow.

Excellent flow properties of this compressor can be expected due to its blade mounting on a stiff rotor with very small radial tip clearances and flow losses together with an aspect ratio of outer to inner flow radius of 440 to 400 mm.

The intercooler requires some development work. The fluids on both sides are unconventional insofar as the working fluid on one side is to be cooled by high pressure steam on the other side. Heat transfer from compression work to steam superheat is thus achieved. The authors can point at previous development work at their institute in a similar problem, i.e. the design of an 850°C steam plant in double loop configuration as published by Perz et al. [16], where many boiler heat transfer problems had been treated.

In the case given the intercooler is thermodynamically part of the HRSG superheater and is thus arranged close to the HRSG. Its cooling flow is steam of 196 bar pressure. It is designed as a solid tube which should be supported on solid ground foundation. The heat transfer surface is realized by 180 tubes of 3.1 m length held in support plates which guide the working gas flow from C1 outlet to C2 inlet. The outer shell of the intercooler is internally insulated and is connected by ample flow areas in flexible scroll and tube arrangements to both compressors (see Fig. 4).

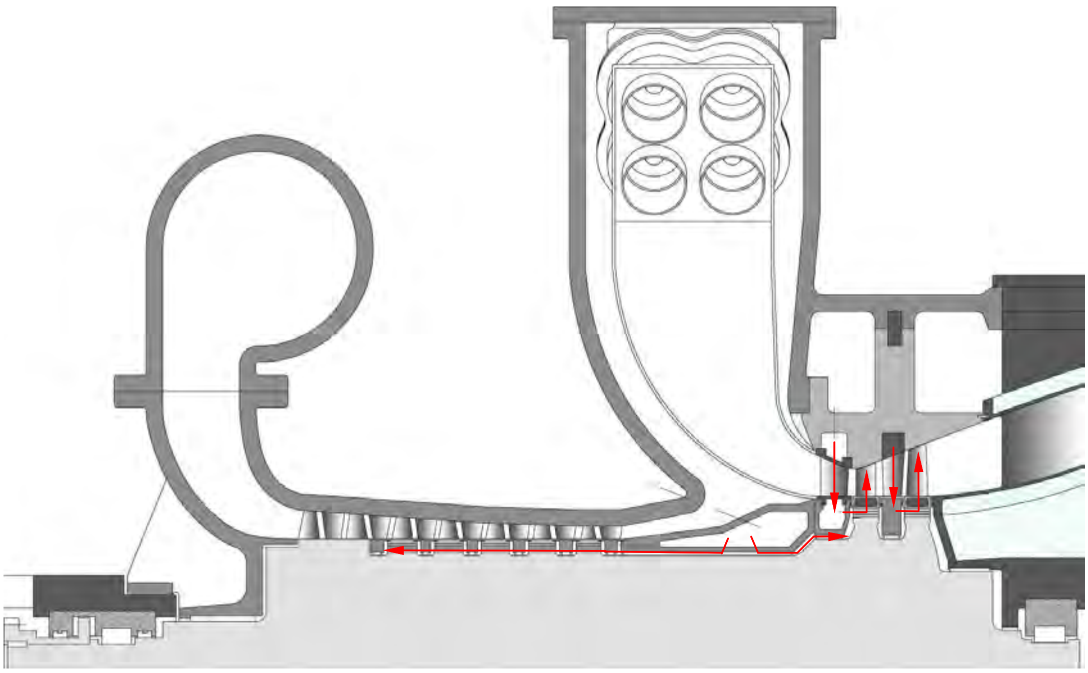


Fig. 6: Design of C2 drum rotor with cooling steam flow arrangement, combuster and HTTC

HTT compressor turbine (HTTC)

The same drum rotor either forged in one piece or welded up from separated disks carries not only the C2 but also the compressor turbine HTTC. The flow design of the HTTC will be a two-stage reaction turbine with 50 % reaction at the mean section of both blade rows. The high rotor speed of 8500 rpm as mentioned before provides for long blade lengths, i.e. a first stage blade of 100 mm and a second stage blade of 164 mm with an inner radius of 533 mm (see Fig. 7). This results in excellent flow properties in subsonic condition and together with the high reaction of the blade on all radii (55 % at mean section and at least 25 % at hub) a high blade flow efficiency is expected. Low tip clearances are applied also contributing to this goal and can be achieved by the excellent rotor dynamics of the stiff drum rotors and very careful blade cooling.

The high speed and power of this turbine is made possible by ample steam cooling. Nozzles and blades are cooled in conventional serpentine passage design with holes as well as the rotor inlet edge as shown in the cooling arrangement of Fig. 7. Rotor cooling steam is supplied along the whole drum surface. It is fed into a labyrinth seal in the inner range of the combustion chamber allowing the steam to flow to both sides. One flow is directed backwards under the dump diffuser into the outer surface of the C2 providing cooling steam as described above. The main amount of cooling steam flows along the rotor drum at the inner radius of the combustor casing towards the first disk of the HTTC.

The first nozzles are hollow with proper cooling passages and are cooled by steam fed from the casing outside in radial inwards direction. The steam is collected in a chamber of the diaphragm just opposite the first blade root. Via nozzles, blowing in direction of rotor speed, the cooling steam is then fed to the lower part of the blade fir-tree root. From there it flows along the serpentine passages under pumping action of the rotor wheel and is delivered to the blade surface via laser drilled holes to form the conventional cooling films at the appropriate locations of the blade. The second guide vanes are supplied with cooling steam which is fed into the outer rim of the diaphragm. There it flows radially inwards also to supply the rotor surface in-between stages and the inlet to the second stage blades which are also built with serpentine passages and the appropriate cooling holes. In principle this design was applied for the well-known gas turbine model GT10, originally designed by F. Zerlauth [17].

In terms of rotor dynamics the drum rotor of C2 and HTTC will be designed for the high stress considering the effect of steam cooling on all surfaces. Stiff bearing shaft extensions and solid double-lobe oil bearings provide for high shaft and high bearing stiffness in order to have all critical speeds sufficiently high above running speed.

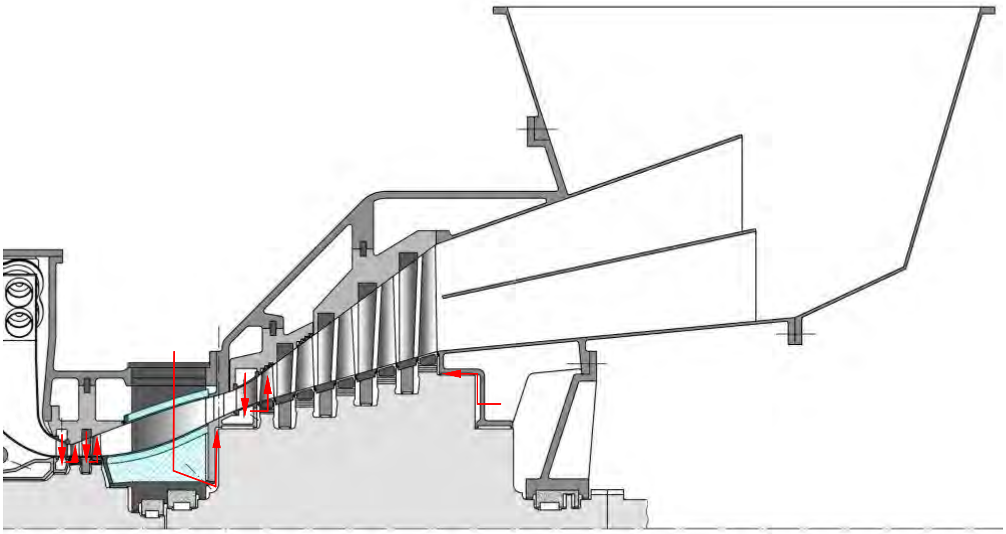


Fig. 7: Design of two-stage HTTC and 50 Hz HTTP

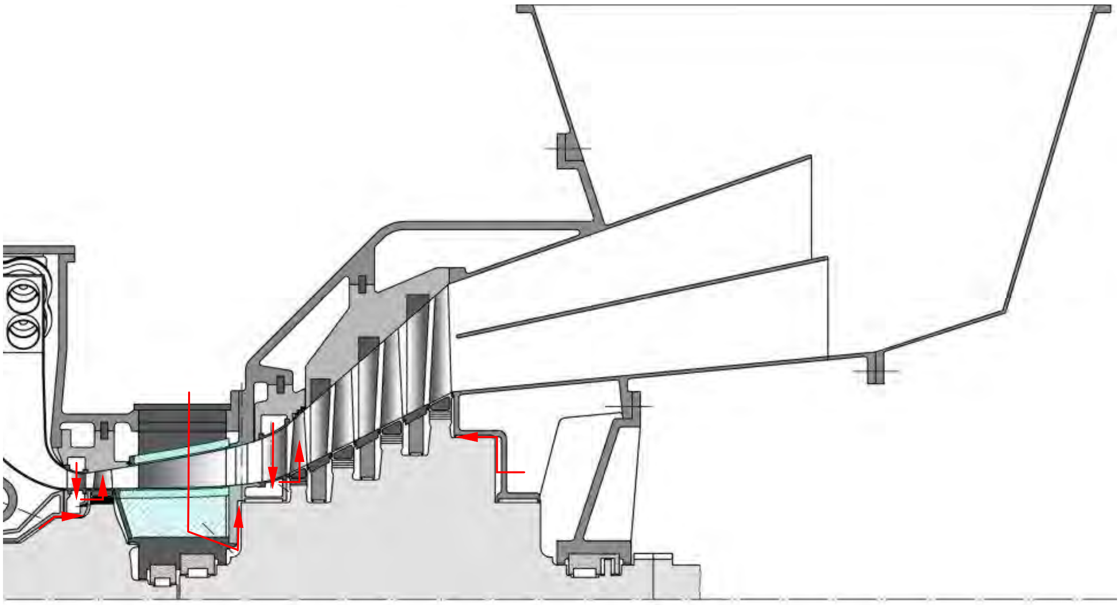


Fig. 8: Design of transonic one-stage HTTC and 60 Hz HTTP

HTTC alternative transonic stage design

The authors' institute has done extensive development work for the design of transonic turbine stages. Not only several computer programs have been developed for investigating three-dimensional transonic flow, but a unique test installation for transonic stages was built where many effects of unsteady viscous transonic flow were investigated (e.g. [18, 19]). The test installation has aided the development of industrial gas turbines and is now in use for several EU projects.

A novel innovative cooling system has also been developed and could be applied here in order to save cooling medium, high temperature material and cost of manufacture at the same time providing most effective blade cooling at the blade leading edge in transonic flow [20, 21]. The design could follow the development path of General Electric in providing thermal barrier coating on the rotating blades since these are free of the multitude of cooling holes and are supplied only by low number of slots creating cooling steam films covering the whole surface.

Therefore, alternatively the HTTC expansion could also be done with one transonic stage as shown in Fig. 8. This can be achieved by a higher radius and stage loading at a somewhat reduced degree of reaction. Such a stage would sit on the same rotor as described before and it would have a mean radius of 750 mm at a blade length of 120 mm. A further advantage of a transonic stage would be the much smaller radius difference from compressor turbine outlet to power turbine inlet, depending also on the speed of the power turbine for which design proposals for 50 and 60 Hz are presented here.

HTT power turbine (HTTP)

A gas turbine system with two shafts at highly different speed as it has to be built here, requires an intermediate bearing to be arranged right between the stages of compressor turbine outlet and power turbine inlet. The flow of gas transmitted is at very high velocity, at temperatures of 1075°C and at a pressure of 14 bar. A conventional design would provide an outlet diffuser, an outlet casing, a transition duct and an inlet casing in between this gas turbine parts. Frictional loss, heat loss, even with internal and external insulation, would be unavoidable. In previous design solutions for industrial size turbines with almost the same cycle conditions the authors have proposed a single overhung disk with a transonic stage for the compressor turbine and one or two overhung disks for the power turbine directly opposite to take over the gas flow in a common casing [10]. This solution requires a high speed power turbine which is only possible to build relying on gears of high speed and high power. The power output of 92 MW in that case made it possible since gas turbines transferring around 100 MW from gas turbine speeds at 5400 rpm to 3000 rpm are in use in industry.

The large power output in the case presented here forbids the use of gears for such high-speed power transfer. Therefore the possible electrical frequencies of 50 Hz in Europe and 60 Hz in USA and western hemisphere were investigated. The power turbine is proposed with a strong change of inner radius on a solid shaft. Five stages are necessary for the 50 Hz design of Fig. 7 and four stages for the 60 Hz of Fig. 8. So the axial outlet speed should be kept at medium value in order to reduce the exhaust loss, to reduce axial diffuser exit length and to facilitate the flow transfer to the HRSG inlet.

The design proposed provides last blade lengths of 750 mm at 50 Hz and of 600 mm at 60 Hz, both at 1300 mm inner radius. At the inlet the inner radius at 50 Hz is somewhat larger than the HTTC outlet, but at 60 Hz, together with a transonic HTTC, it provides a flow path at almost the same radius as shown in Fig. 8.

The intermediate bearing casing in its hot environment has to be insulated on its outer surface in a mode of insulation withstanding the friction of the hot outer flow. The same holds true for the three supporting ribs, which have to provide ample inner space for transfer of oil, cooling air and steam leakage outlet from labyrinths on both shaft sides as well as for monitoring equipment. At the same time the bearing should be as short as possible and the ribs should provide only a minimum of flow resistance. Certainly this is an object that deserves intensive flow, stress and heat transfer deliberations.

The thrust equalization of both types of power turbines cannot be made in the conventional manner of steam turbines. A balance piston requires a diameter of about the mean blade mean diameter to give proper balance of axial forces. In this case such a design would require an unacceptable flow turn and deviation of the hot gas flow. (In a one-shaft gas turbine the problem does not exist, since compressor thrust and turbine thrust balance each other.) On the other hand, to carry the axial thrust of a large power turbine especially in the conical form is impossible for oil thrust bearings. Size and oil friction power loss would be too high. Therefore a stepped labyrinth on the exhaust side of the rotor drum is proposed as shown in Figs. 7 and 8, which is supplied with internal steam pressure to provide for the necessary thrust equalization. The steam supply feeds also the cooling flow which is led along the rotor drum surface under the root sealing plates for the last and the penultimate stage, whereas cooling flow to the first and second stage is supplied via the hollow nozzle blades to an inner diaphragm cavity from which the inflow to the hollow rotor blades is affected. Power turbine thrust bearing is arranged outboard of casing in vicinity of steam operated balance piston (see Figs. 7 and 8).

Low Pressure Steam Turbine (LPST)

The LPST is fed with steam of 0.75 bar and 175°C. Expanding the steam to a condensation pressure of 0.021 bar leads to a high volume flow. At 50 Hz a four-flow design with three stages, as shown in Fig. 4, is able to handle the high steam flow with excellent efficiency. The last stage is transonic with a blade length of 970 mm. In the shaft arrangement this steam turbine is coupled to the far side of the main generator.

HPT

The HPT is a standard high-speed back-pressure steam turbine of 50 MW power output for which many designs are in the market. A geared type seems to be a superior solution since better flow efficiency and operability due to nozzle boxes and low number of stages with long blades and low leakage loss can be achieved. It can be coupled to the far end of the LPST or can be used to drive a separate smaller electric generator.

Compressors C3 and C4

The delivery compressors C3 and C4, which increase the pressure of the working fluid prior to condensation in order to obtain better evaporation conditions for the bottoming steam cycle, are also needed to vent the internal volume before start up. They are driven by two separate speed-controlled motors.

Combustion

The combustion chamber and burner design proposed has been thoroughly tested in science of combustion. Research partners have run CH_4/oxygen burners in a steam environment successfully [22, 23]. The authors' proposal [7] of setting a separate oxygen and fuel inlet in the center of a strong steam vortex in a large number of separate burners within the combustion chamber provides for easy control of amount and ratio of oxygen and fuel together with ignition and flame observation. The steam vortex keeps together both reactants. The independent supply of both reactants together with the high flame speed caused by pure oxygen lets expect improvements compared to the otherwise acoustic vibration prone conventional low-NO_x combustion chamber flows.

4.3 Economic Information

This information is taken from the ASME Paper GT2006-90032:

Economic Evaluation

Despite the high efficiency and the positive impact on the environment by a Graz Cycle power plant, a future application of this technology and an erection of a power plant mainly depends on the economical balance. The main indicator characterizing the economic performance of a power plant for CO2 capture are the mitigation costs. They represent the increased capital and operational costs incurred by new and additional equipment and lower cycle efficiencies in relation to the CO2 mass flow avoided. The CO2 captured has an economic value of about 10 $/ton, if it can be used for enhanced oil recovery (EOR) or of about 30 $/ton in the future CO2 emission trading scenario. These prices show the current threshold for the economic operation of zero emission power plants.

In order to estimate the mitigation costs for a Graz Cycle plant, an economic comparison with a state-of-the-art combined cycle power plant of 58% efficiency is performed. The economic balance is based on following assumptions: 1) the yearly operating hours is assumed at 8500 hrs/yr; 2) the capital charge rate is 12%/yr, which corresponds to an interest rate of 8 % over a depreciation period of 15 years; 3) methane fuel costs are 1.3 ¢/kWhth; 4) the investment costs per kW are the same for the reference plant of about 400 MW net power output and the Graz Cycle plant (see below); 5) additional investment costs are assumed for the air separation unit (ASU), for additional equipment and CO2 compression to 100 bar (see Table 2 [24]); 6) the costs of CO2 transport and storage are not considered because they depend largely on the site of a power plant.

Table 2: Estimated investment costs

Component	**Scale parameter**		**Specific costs**
Reference Plant			
Investment costs	Electric power	$\$/kW_{el}$	414
Graz Cycle Plant			
Plant investment costs	Electric power	$\$/kW_{el}$	414
Air separation unit [24]	O_2 mass flow	$\$/(kg\ O_2/s)$	1 500 000
Other costs (Piping, CO_2-Recirc.) [24]	CO_2 mass flow	$\$/(kg\ CO_2/s)$	100 000
CO_2-Compression system [24]	CO_2 mass flow	$\$/(kg\ CO_2/s)$	450 000

The assumption of similar investment costs for a conventional and a Graz Cycle power plant is based on a comparison with typical turbomachinery sizes for a 400 MW combined cycle plant as given in Table 3. It shows that the turbine power and the HRSG is of similar size, whereas the compressor power is remarkably smaller. On the other hand the Graz Cycle needs a larger generator due to the additional power consumption for ASU and CO2 compression. Development efforts needed especially for HTT and combustor are not considered in the investment costs.

Table 3: Comparison of equipment size for a 400 MW plant in terms of power

	Conventional CC plant	**Graz Cycle plant**
turbine of "gas turbine"/ HTT	667 MW	618 MW
compressor of "gas turbine"/ C1+C2+C3+C4	400 MW	232 MW
steam turbine/ HPT+LSPT	133 MW	120 MW
HRSG	380 MW	360 MW
Generator	400 MW	490 MW

Table 4: Economic data for a 400 MW Graz Cycle plant

	Reference plant	**S-GC base version**
Reference Plant		
Plant capital costs [$/kW$_{el}$]	414	414
Addit. capital costs [$/kW$_{el}$]		288
CO_2 emitted [kg/kWh$_{el}$]	0.342	0.0
Net plant efficiency [%]	58.0	53.1
COE for plant amort. [¢/kWh$_{el}$]	0.58	0.99
COE due to fuel [¢/kWh$_{el}$]	2.24	2.45
COE due to O&M [¢/kWh$_{el}$]	0.7	0.8
Total COE [¢/kWh$_{el}$]	**3.52**	**4.24**
Comparison		
Differential COE [¢/kWh$_{el}$]		**0.72**
Mitigation costs [$/ton CO_2 avoided]		**21.0**

Table 4 shows the result of the economic evaluation. Compared to the reference plant, the capital costs are about 70 % higher only by considering the additional components for O2 generation and CO2 compression. So they contribute mostly to the difference in COE. The fuel costs have the major influence on the COE, especially for syngas firing, but they do not differ largely between reference and Graz Cycle plant. The O&M costs are assumed 15 % higher for a Graz Cycle plant due to the operation of additional equipment.

Based on these assumptions, the COE of a methane fired Graz Cycle plant of 53.1 % net efficiency is 0.72 ¢/kWhel higher than for the reference plant. The mitigation costs are 21.0 $/ton of CO2 avoided, if CO2 liquefaction is considered. This value is clearly below the threshold value of 30 $/ton showing the economic potential of the Graz Cycle.

The results of the economic study depend mainly on the assumptions about investment costs, fuel costs and capital charge rate. A cost sensitivity analysis performed in [11] showed that a variation of the capital costs has the main influence on the economics, since they contribute most to the mitigation costs. Unfortunately, there is a large uncertainty of these costs. A survey of the ASU costs vary in the range of 230 to 400 $/kWel (the same price as for a complete power plant). These costs are for a cryogenic ASU as used e.g. in steel industry for half a century. There is certainly a potential for effectivity increase. Oxygen from membranes which are under intensive development now are not yet available for plants of the output in discussion. The ASU appears to be a decisive cost factor. Only considering its cost variation, the mitigation costs vary between 21.0 and 27.9 $/ton CO2 for the methane fired plant (see Fig. 9).

This high sensitivity to the capital costs shows the dilemma in performing an exact economic evaluation, since their estimation for a Graz Cycle power plant is very difficult because of new turbomachinery components. But the authors claim that their design of high-speed transonic stages with innovative steam cooling allows a cost-effective manufacture. In these considerations about the height of additional investment costs, a further advantage of the Graz Cycle, the almost NOx-free combustion was not evaluated. According to [25] exhaust flow NOx and CO catalytic reduction to achieve single-digit emissions (in strict attainment areas) can increase gas turbine genset plant costs by 40 to 50 percent.

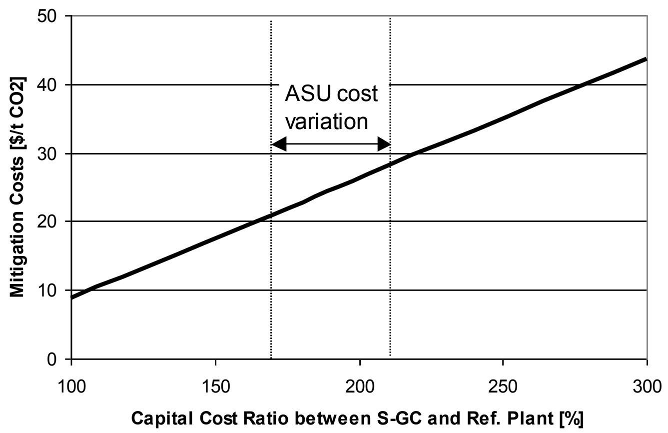


Fig. 9: Influence of capital costs on the mitigation costs (CO2 provided at 100 bar)

4.4. Further Information

All our papers can be found at our website

graz-cycle.tugraz.at

Graz, July 11th, 2014

Wolfgang Sanz

Institute for Thermal Turbomachinery and Machine Dynamics

Graz University of Technology

Graz, Austria

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : CASE 1: SCOC-CC
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 1 of the study, which is a semi-closed oxy-combustion combined cycle (SCOC-CC) plant, with cryogenic purification and separation of the carbon dioxide. The plant is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected SCOC-CC plant configuration is based on two parallel trains, each composed of one F-class equivalent oxy-fired gas turbine and one heat recovery steam generator (HRSG), generating steam at three levels of pressure, including a LP integrated deaerator. The generated steam feeds one steam turbine, water-cooled and condensing type, common to the two parallel trains.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 1 – Unit arrangement

Unit	Description	Trains
3000	Power Island	
	Gas Turbine	2 x 50%
	HRSG	2 x 50%
	Steam Turbine	1 x 100%
4000	CO_2 purification and compression	
	Raw gas compression	2 x 50%
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power Island

The unit is mainly composed of:

- Two F-class equivalent oxy-fired gas turbines.
- Two heat recovery steam generators (HRSG), generating steam at two levels of pressure, plus a LP integrated deaerator.
- Two recycle gas indirect contact cooling systems.
- One steam turbine, water-cooled and condensing type, common to the two parallel trains.

Technical information relevant to this unit is reported in chapter D, section 2.1.1, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the stream at gas turbine boundary are summarised in the following Figure 1.

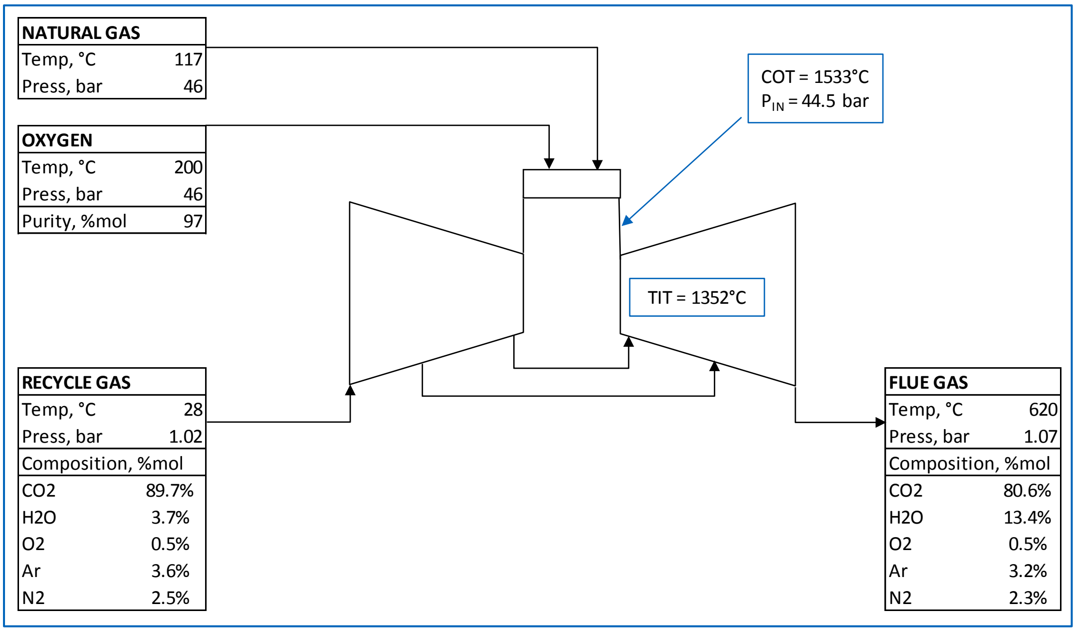


Figure 1. SCOC-CC gas turbine

The natural gas from the let-down and metering station is heated using heat available from the CO_2 compression in the CPU before entering the burners of the gas turbine at 117°C.

Oxygen is delivered from the ASU at the required pressure level and heated using heat available from the raw gas compressor in the CPU before entering the burners of the gas turbine at 200°C.

The pressure ratio of the SCOC-CC is higher than the one of an equivalent standard air blown commercial plant. Turbine inlet pressure of 44.5 bar has been selected to bring about the same temperature increase (385°C) across the compressor of the reference air-fired gas turbine. A pressure slightly higher than the ambient pressure (to avoid leakages into the CO_2 loop) has been selected so as to keep the design of the turbomachines closer to the current standards.

The gas turbine expander has 5 stages, so to have acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine, at a rotational speed of 3000 RPM.

The gas turbine recycle flowrate is fixed by two design requirements to:

- control the combustion outlet temperature at 1533°C;
- provide the required cooling flow to the gas turbine blade in order to control the blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st stator	860°C
1st rotor	830°C
from 2nd stator	830°C
from 2nd rotor	800°C

The exhaust gases from the gas turbine enter the HRSG at 620°C. The HRSG recovers heat available from the exhaust gas producing steam at the following pressure levels for the steam turbine, including the steam generator with integral deaerator.

- HP steam: 160 bar, (SH: 590°C)
- MP steam: 37 bar, (RH: 590°C)
- LP steam: 5.5 bar.

The final exhaust gases are cooled down in a conventional contact cooler to the minimum temperature allowed by the cooling medium available. Most of the flue gas at 28°C from the top of the contact column (around 93%) is recycled back to the gas turbine compressors, while the remaining stream is sent to the downstream CO_2 purification and compression unit.

The combined cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular, heat available at high temperature level from the oxy-turbine cycle is used to provide heat required in the CPU mainly for TSA regenerator heating and inert gas heating before expansion, while heat available at low temperature level in the CO_2 compressor intercoolers is used for oxygen and natural gas heating in order to enhance gas turbine efficiency.

The following interfaces have been considered:

- Natural gas is heated using as heating medium compressed CO_2 from the final compression before being sent to plant B.L.
- Oxygen is heated using as heating medium raw flue gas from the first compression stage (1-15 bar) in the CPU.
- Saturated HP water is used as heating medium in the TSA regenerator heater of the CPU. Cold water from the exchanger is sent to the LP degassing section of each HRSG.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 50 bar. Oxygen pressure is set by the requirement of the gas turbine combustor.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- Raw flue gas compression (1 - 34 bar);
- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

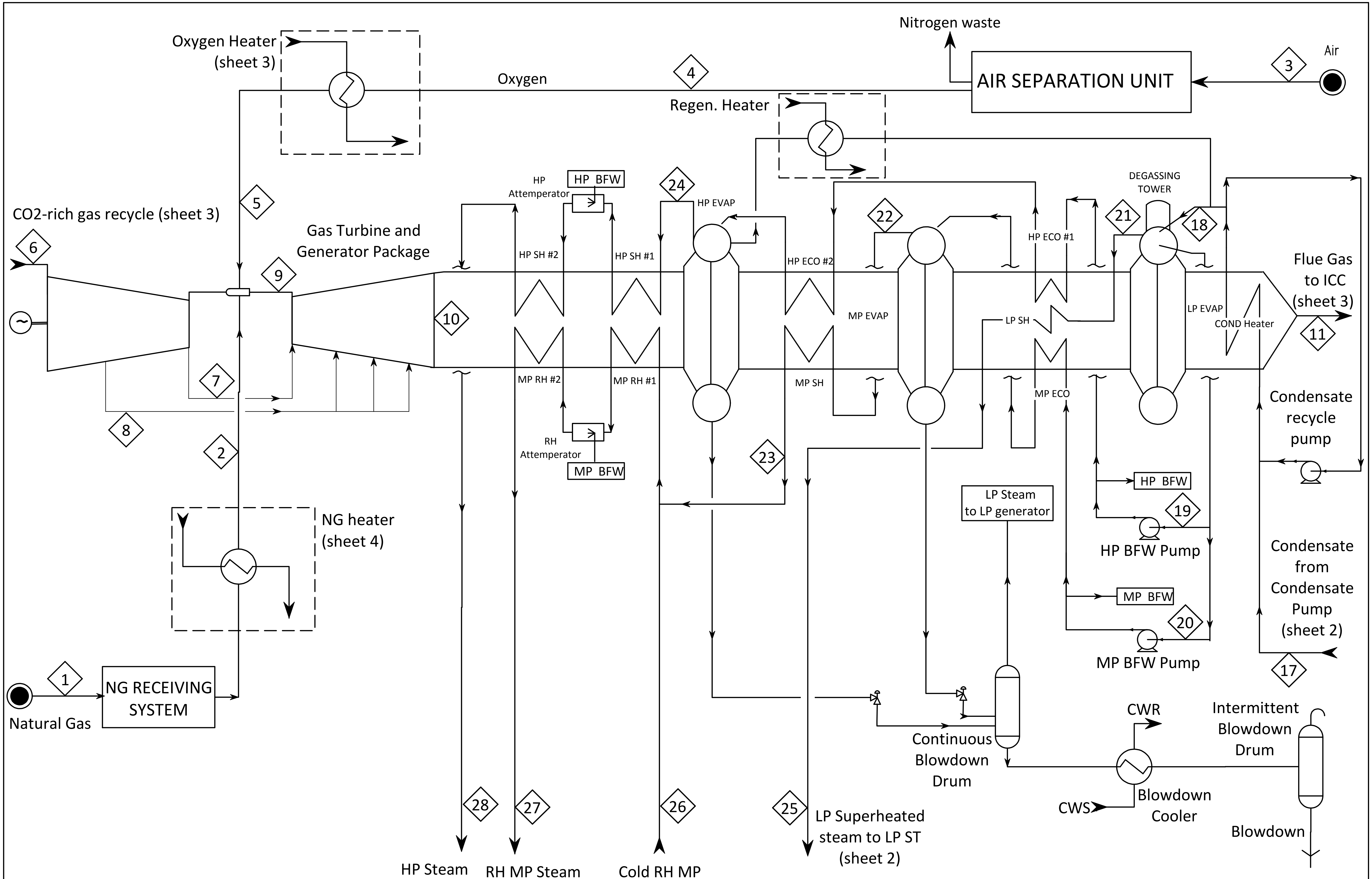
Basin diameter	120 m
Cooling tower height	210 m
Water inlet height	17 m

- Natural gas receiving station;
- Raw water system;
- Demineralised water plant;
- Waste Water Treatment
- Fire fighting system;
- Instrument and Plant air.

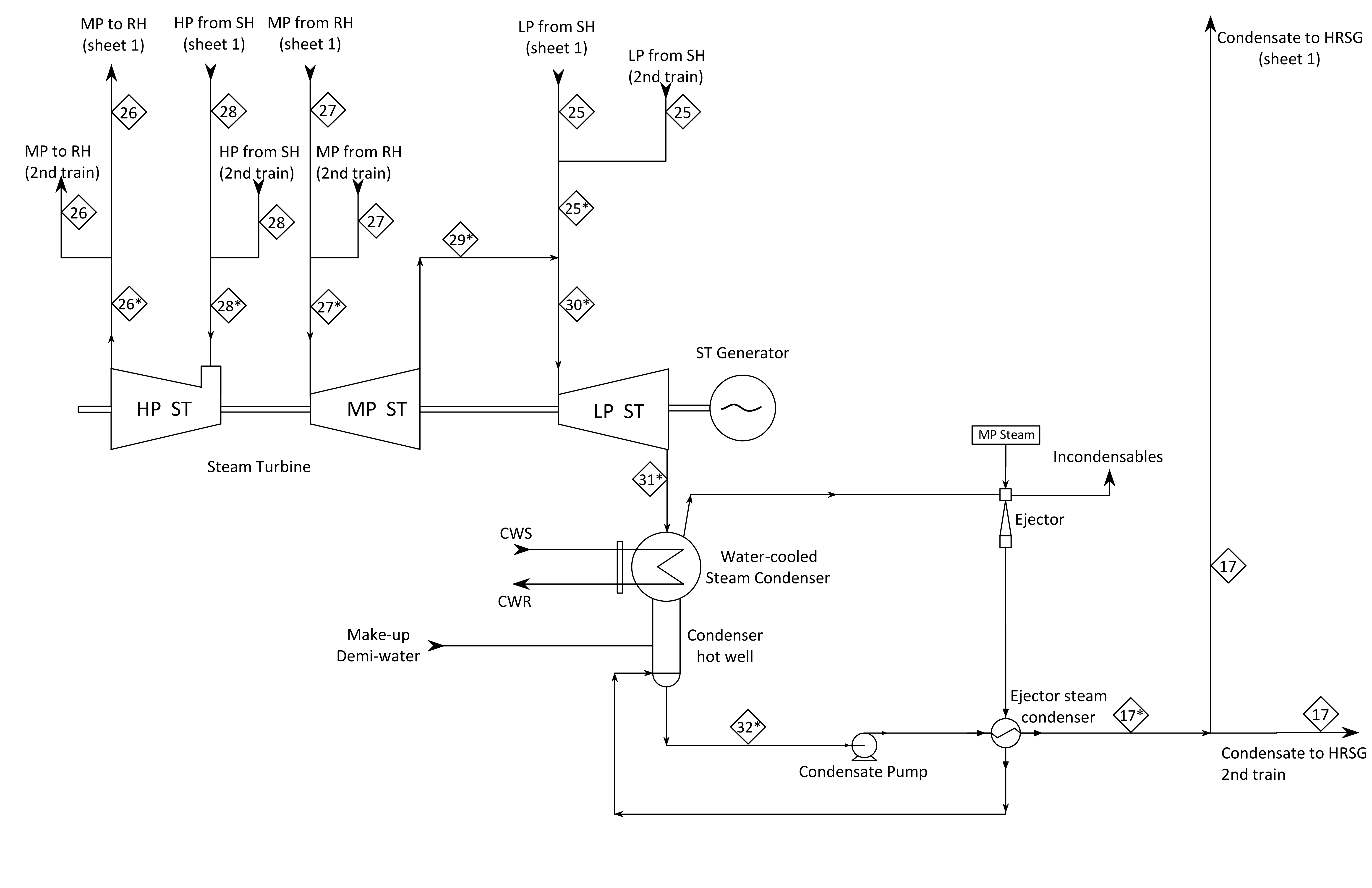
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

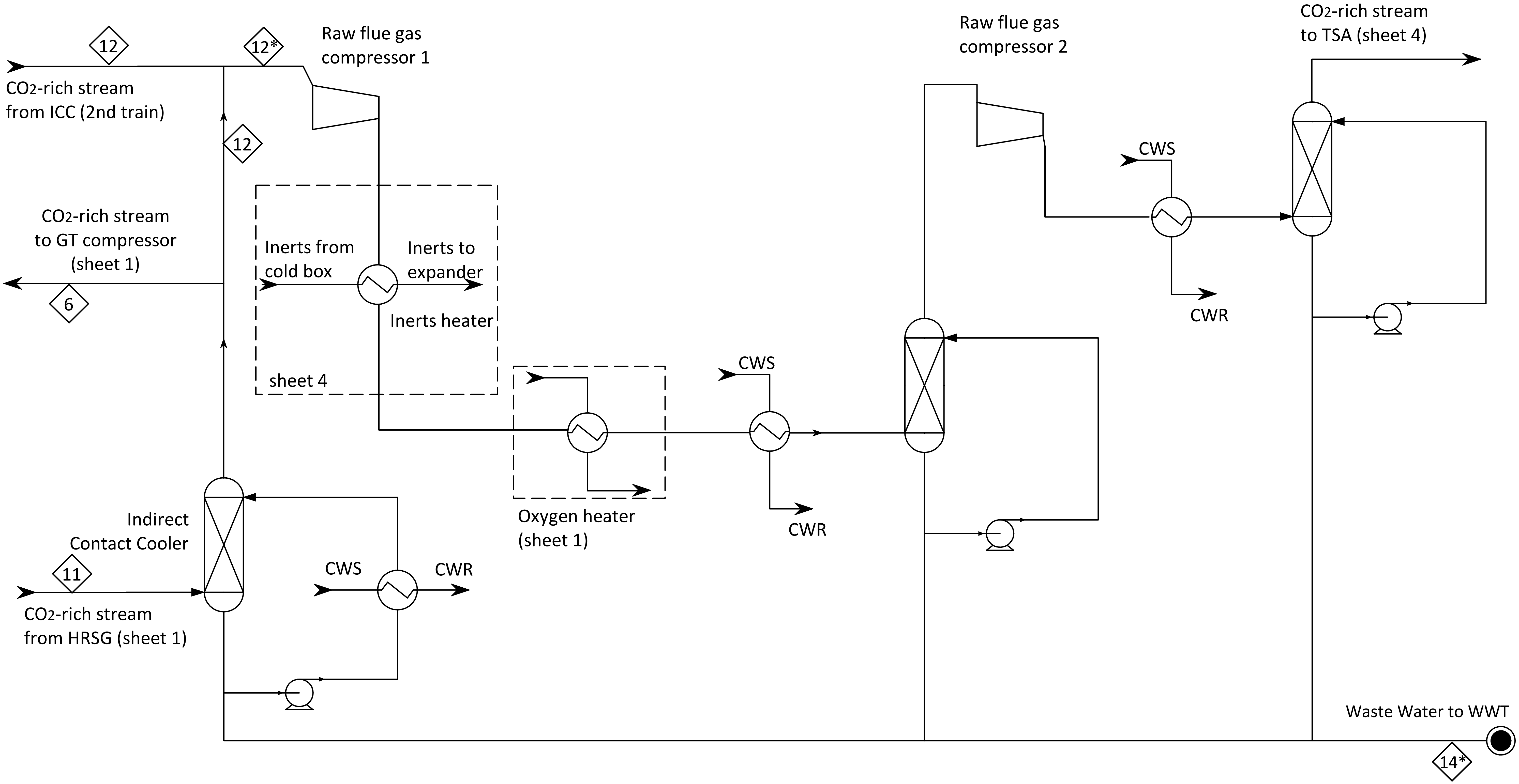
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



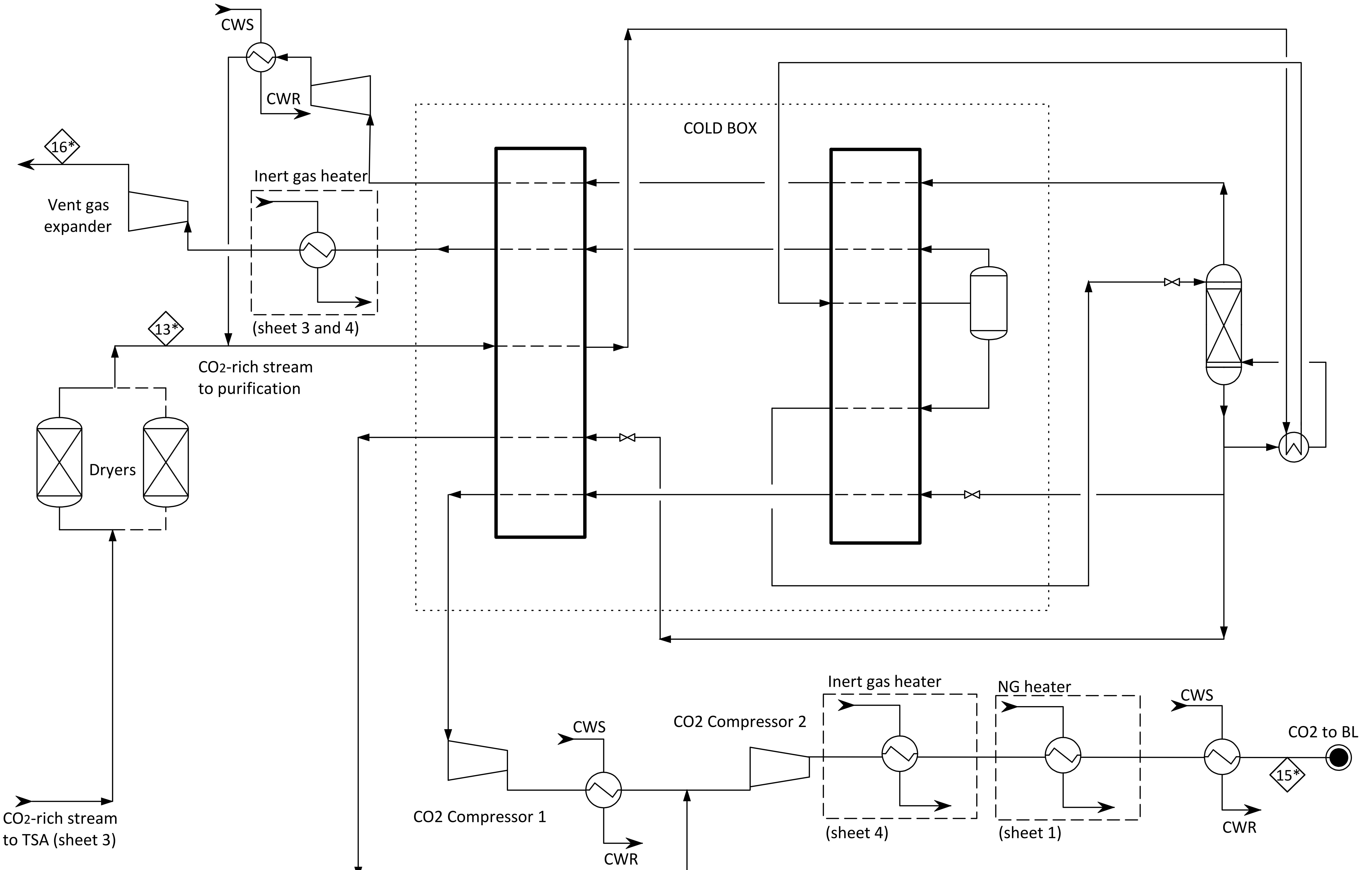
0	Jun 2014	Draft report issue	FF	LM	**UNIT:** GT and HRSG	
Rev.	Date	Comment	By	Appr	**CASE:** 1 - SCOC CC	Sheet 01 of 04



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Steam turbine and condenser	
Rev.	Date	Comment	By	Appr	**CASE:** 1 - SCOC CC	Sheet 02 of 04



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** CO2 purification - compression section	
Rev.	Date	Comment	By	Appr	**CASE:** 1 - SCOC CC	Sheet 03 of 04



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** 1 - SCOC CC	Sheet 04 of 04

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

FOSTER WHEELER	**Case 1 - SCOC-CC - HEAT AND MATERIAL BALANCE**						**REVISION**	0		
	CLIENT :		IEAGHG				**PREP.**	FF		
	PROJECT NAME:		Oxy-turbine power plants				**CHECKED**	NF		
	PROJECT NO:		1-BD-0764 A				**APPROVED**	LM		
	LOCATION:		The Netherlands				**DATE**	September 2014		

HEAT AND MATERIAL BALANCE
UNIT 3000 - GAS TURBINE

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	Heated NG to combustor	Air to ASU	Oxygen from ASU	Heated Oxygen to combustor	CO2-rich gas recycle	1st coolant stream	2nd coolant stream	Flue gas from combustor	Flue gas to HRSG
Temperature (°C)	15	117	9	15	200	28	413	variable	1533	619
Pressure (bar)	70	45.8	amb	46.7	45.8	1.02	45.8	variable	44.5	1.07
TOTAL FLOW										
Mass flow (kg/h)	59,470	59,470	995,960	228,170	228,170	2,298,960	391,500	424,300	1,761,840	2,586,600
Molar flow (kmol/h)	3,300	3,300	34,510	7,105	7,105	54,165	9,225	10,000	45,290	64,720
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	59,470	59,470	995,960	228,170	228,170	2,298,960	391,500	424,296	1,761,840	2,586,600
Molar flow (kmol/h)	3,300	3,300	34,510	7,105	7,105	54,165	9,224	9,996	45,290	64,720
Molecular Weight (kg/kmol)	18.0	18.0	28.9	32.1	32.1	42.4	42.4	42.4	38.9	40.0
Composition (%mol)	As assigned	As assigned								
Ar			0.92%	2.00%	2.00%	3.56%	3.56%	3.56%	3.05%	3.20%
CO_2			0.04%	0.00%	0.00%	89.73%	89.73%	89.73%	76.71%	80.62%
H_2O			0.97%	0.00%	0.00%	3.67%	3.67%	3.67%	17.64%	13.45%
N_2			77.32%	1.00%	1.00%	2.52%	2.52%	2.52%	2.15%	2.26%
O_2			20.75%	97.00%	97.00%	0.52%	0.52%	0.52%	0.44%	0.47%
Total			100%	100%	100%	100%	100%	100%	100%	100%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

FOSTER WHEELER	Case 1 - SCOC-CC - HEAT AND MATERIAL BALANCE		REVISION	0		
	CLIENT :	IEAGHG	PREP.	FF		
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	NF		
	PROJECT NO:	1-BD-0764 A	APPROVED	LM		
	LOCATION:	The Netherlands	DATE	September 2014		

HEAT AND MATERIAL BALANCE
UNIT 4000 - CPU

STREAM	11	12*	13*	14*	15*	16*				
	Flue gas from HRSG	CO2 rich stream to compression unit	CO2 rich stream to purification unit	Waste water to WWT	Purified CO2	Vent gas from CPU				
Temperature (°C)	67	28	26.1	26.2	30	70				
Pressure (bar)	1.05	1.02	34.90	20.00	110.00	1.10				
TOTAL FLOW										
Mass flow (kg/h)	2,586,600	338,470	333,100	120,990	286,600	46,500				
Molar flow (kmol/h)	64,720	7,975	7,680	6,715	6,515	1,165				
LIQUID PHASE										
Mass flow (kg/h)				120,990						
GASEOUS PHASE										
Mass flow (kg/h)	2,586,600	338,470	333,100		286,600	46,500				
Molar flow (kmol/h)	64,720	7,975	7,680		6,515	1,165				
Molecular Weight (kg/kmol)	40.0	42.4	43.4		44.0	39.9				
Composition (%mol)										
Ar	3.20%	3.56%	3.70%	0.0%	0.44%	21.99%				
CO_2	80.62%	89.73%	93.15%	traces	99.55%	57.25%				
H_2O	13.45%	3.67%	0.00%	100.0%	0.00%	0.00%				
N_2	2.26%	2.52%	2.61%	0.0%	0.00%	17.26%				
O_2	0.47%	0.52%	0.54%	0.0%	0.01%	3.50%				
Total	100%	100%	100%	100%	100%	100%				

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

FOSTER WHEELER

	Case 1 - SCOC-CC - HEAT AND MATERIAL BALANCE		Revision	0	
	CLIENT:	IEAGHG	**Prepared**	FF	
	PROJECT NAME:	Oxy-turbine power plants	**Checked**	NF	
	PROJECT NO:	1-BD-0764 A	**Approved**	LM	
	LOCATION:	The Netherlands	**Date**	September 2014	

HEAT AND MATERIAL BALANCE
UNIT 3000 - STEAM CYCLE

Stream	Description	Flowrate t/h	Temperature °C	Pressure bar a	Entalphy kJ/kg
17	Condensate from condensate pump	412.4	30	8.0	124.4
18	Preheated condensate to LP steam drum	415.1	154	6.0	649.5
19	BFW to HP BFW Pump	329.5	159	6.0	670.4
20	BFW to MP BFW Pump	43.6	159	6.0	670.4
21	Saturated LP Steam from LP drum	42.0	159	6.0	2756
22	Saturated MP Steam from MP drum	43.6	250	40.0	2801
23	MP Steam from Superheater	43.6	334	39.0	3056
24	Saturated HP Steam from HP steam drum	326.8	350	165.9	2562
25*	LP Superheated steam to LP Steam Turbine	84.0	300	5.5	3063
26*	Cold MP steam from Steam Turbine to reheater	647.2	382	40.0	3170
27*	Reheated MP steam to Steam Turbine	734.4	593	36.8	3662
28*	HP Superheated steam to Steam Turbine	653.7	590	152.7	3554
29*	Exhaust from MP Steam Turbine	734.4	312	5.5	3087
30*	LP steam to LP Steam Turbine	818.4	310	5.5	3085
31*	Exhaust from LP steam turbine	818.4	29	0.04	2306
32*	Condensate from condenser	824.9	29	0.04	121.5

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 1 – Water consumption summary

CLIENT: IEAGHG		**REVISION**	0
PROJECT NAME: Oxy-turbine power plant		**DATE**	Apr-14
PROJECT No. : 1-BD-0764A		**MADE BY**	NF
LOCATION : The Netherlands		**APPROVED BY**	LM

Case 1 - SCOC-CC

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water [DT = 11°C] [t/h]
3000	**OXY-TURBINE CYCLE**			
	Condenser			**39,830**
	Turbine and generator Auxiliaries		**10**	**4,000**
	Indirect contact cooler			**18,020**
5000	**AIR SEPARATION UNIT**			
	MAC intercoolers			**9,300**
	BAC intercoolers			**1,310**
4000	**CO_2 PURIFICATION UNIT**			
	CO2 purification unit			**1,750**
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	**1,335**		
	Demineralized water unit	**15**	**-10**	
	Waste Water Treatment and Condensate Recovery	**-230**		
	Balance of plant			
	BALANCE	**1,120**	**0**	**74,210**

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 1 – Electrical consumption summary

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Apr-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 1 - SCOC-CC

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**OXY-TURBINE CYCLE**	
	BFW and condensate pumps	**5,120**
	Turbine Auxiliaries + generator losses	**5,190**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**128,000**
	Booster air compressor and miscellanea	**19,100**
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	**29,070**
	Autorefrigerated inerts removal unit compression consumption	**13,940**
	Autorefrigerated inerts removal unit expander production	**-2,260**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**9,290**
	Balance of plant	**1,460**
	BALANCE	**208,910**

6. Overall performance

The following table shows the overall performance of Case 1, including CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	Apr-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 1 - SCOC-CC

OVERALL PERFORMANCES		
Natural Gas flow rate	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
Gas turbine power output (@ gen terminals)	MWe	611.8
Steam turbine power output (@ gen terminals)	MWe	356.2
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**968.0**
Oxy-turbine cycle	MWe	10.3
Air separation unit	MWe	147.1
CO2 purification and compression unit	MWe	40.8
Utility & Offsite Units	MWe	10.8
ELECTRIC POWER CONSUMPTION	**MWe**	**208.9**
NET ELECTRIC POWER OUTPUT	**MWe**	**759.1**
(Step Up transformer efficiency = 0.997%) (B)	MWe	756.8
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**63.0%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.3%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**56.9%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**44.6%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6487
CO_2 removal efficiency	**%**	**90.6**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**
CO_2 emission per net power production	**kg/MWh**	**38.7**

6.1. Performance sensitivity to significant design parameters

In addition to the base case, the plant performance has been evaluated after modifying some of the most significant design parameters. The following Table 4 summarises the sensitivity cases selected for the SCOC-CC, highlighting the main design features and performance figures affected by each modified parameter.

Table 4. SCOC-CC – Sensitivity cases

Case 1 – Sensitivity to ambient temperature		
Base case figure	Sensitivity	Impact
T_{ref} = 9°C	T = 25°C	• Cooling water temperature: 30-41°C • Condenser operating conditions: 44°C, 9.2 kPa • Higher ASU and CPU compressor consumptions • Lower power generation from the steam turbine
Case 1 – Sensitivity to COT		
Base case figure	Sensitivity	Impact
T_{ref} = 1533°C	T = 1453°C	• Same maximum allowable metal temperature • Higher power generation from the gas turbines • Lower power generation from the steam turbine
T_{ref} = 1533°C	T = 1613°C	• Increased maximum allowable metal temperature • Higher power generation from the gas turbines • Lower power generation from the steam turbine • Combustor pressure increases in order to maintain the exhaust gas temperature within the suitable range for the downstream HRSG • Higher ASU consumptions (higher oxygen delivery pressure)
Case 1 – Sensitivity to discharge pressure (same compression ratio)		
Base case figure	Sensitivity	Impact
P_{ref} = 1.07 bar	P = 2 bar	• Combustor pressure increases in order to keep the same compression/expansion ratio. This allows to maintain the exhaust gas temperature within the suitable range for the downstream HRSG • Higher ASU consumptions (higher oxygen delivery pressure) • NG compressors required • Lower CPU electrical consumption but lower heat made available to the combined cycle

Case 1 – Sensitivity to compression ratio (different inlet pressure)		
Base case figure	Sensitivity	Impact
β_{ref} = 44.5	β = 39.5	• Lower power generation from the gas turbines • Higher exhaust gas temperature and steam turbine power production
β_{ref} = 44.5	β = 34.5	• Lower power generation from the gas turbines • Higher exhaust gas temperature and steam turbine power production

6.1.1. *Sensitivity to ambient temperature*

Table 5 shows the impacts on key design parameters and plant performances when the ambient temperature varies from reference (9°C) to a higher value (25°C).

Table 5. SCOC-CC – Sensitivity to ambient temperature

			BASE CASE	SENSITIVITY CASE
SENSITIVITY				
Ambient temperature		**°C**	**9**	**25**
DESIGN FEATURES				
Cooling water temperature	Supply	°C	15	30
	Return	°C	26	41
Condensing temperature		°C	29	44
Indirect contac cooler temperature		°C	28	43
Condensing pressure		kPa	4.0	9.1
PERFORMANCES COMPARISON				
Natural Gas flow rate (A.R.)		t/h	118.9	118.9
Natural Gas LHV		kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)		**MWth**	**1536**	**1536**
Gas turbine power output (@ gen terminals)		MWe	611.8	602.8
Steam turbine power output (@ gen terminals)		MWe	356.2	334.6
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)		**MWe**	**968.0**	**937.4**
Oxy-turbine cycle		MWe	10.3	10.3
Air separation unit		MWe	147.1	154.3
CO2 purification and compression unit		MWe	40.8	44.5
Utility & Offsite Units		MWe	10.8	10.9
ELECTRIC POWER CONSUMPTION		**MWe**	**208.9**	**220.0**
NET ELECTRIC POWER OUTPUT		**MWe**	**759.1**	**717.4**
(Step Up transformer efficiency = 0.997%) (B)		MWe	756.8	715.2
Gross electrical efficiency (C/A x 100) (based on LHV)		**%**	**63.0%**	**61.0%**
Net electrical efficiency (B/A x 100) (based on LHV)		**%**	**49.3%**	**46.6%**
Fuel Consumption per net power production		**MWth/MWe**	**2.03**	**2.15**
CO_2 emission per net power production		**kg/MWh**	**38.7**	**40.9**

Based on the above data, the following considerations can be drawn:

- Gross power production is reduced by around two (2) percentage points due to the lower power production of both the steam turbine (higher condensing pressure) and the gas turbine (performance degradation due to the higher recycle gas temperature from the indirect contact cooler).

- Net power production is also affected by the higher compressor power demand, both in the ASU and in the CPU, due to the higher cooling level and consequently the higher compressor inlet temperature downstream the intercoolers. Resulting penalty is nearly three (3) percentage points on the net electrical efficiency.

6.1.2. Sensitivity to combustion outlet temperature

As specified in the section 2.2, the combustion outlet temperature (COT) of the base case is set at 1533°C, controlled by the recycled gas flowrate.

Table 6 and Table 7 show the impacts on key design parameters and plant performances when the COT varies from reference to respectively higher and lower values (1613°C and 1453°C).

Higher COT case

The following design changes have been considered as a consequence of the increased COT:

- The maximum allowable blade metal temperature is increased as detailed below, thus affecting the required cooling flow to the gas turbine blade and hence the recycle flowrate:

Turbine section	Maximum allowable temperature	
	High COT case	Base case
1st stator	950°C	860°C
1st rotor	890°C	830°C
from 2nd stator	890°C	830°C
from 2nd rotor	860°C	800°C

  The temperature increased has been selected keeping constant the cooling flows in an equivalent air blown combined cycle, having the same temperature profile of the gas turbine included in the SCOC-CC.

- The compressor outlet pressure (and consequently the combustor and the turbine inlet pressure) is increased in order to maintain an exhaust gas temperature in a suitable range for an efficient heat recovery in the downstream HRSG as for the base case, while increasing the turbine inlet temperature. In fact, as the maximum steam temperature is around 600°C (both for superheated and reheated steam) the optimum range for the exhaust temperature is around 620-630°C.

Table 6. Case 1 – Sensitivity to COT – Higher COT case

		BASE CASE	SENSITIVITY CASE (Higher COT)
SENSITIVITY			
Combustion outlet temperature	**°C**	**1533**	**1613**
DESIGN FEATURES			
Compressor outlet pressure	bar	45.8	56.8
Compressor ratio	-	44.5	55.1
Oxygen delivery pressure from ASU	bar	47	58
Turbine inlet temperature	°C	1352	1412
Max turbine metal temperature		see table above	
Flue gas temperature (HRSG inlet)	°C	620	625
Flue gas temperature (HRSG outlet)	°C	67	80
PERFORMANCES COMPARISON			
Natural Gas flow rate	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
Gas turbine power output (@ gen terminals)	MWe	611.8	636.6
Steam turbine power output (@ gen terminals)	MWe	356.2	342.3
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**968.0**	**978.9**
Oxy-turbine cycle	MWe	10.3	10.1
Air separation unit	MWe	147.1	149.8
CO2 purification and compression unit	MWe	40.8	40.8
Utility & Offsite Units	MWe	10.8	10.6
ELECTRIC POWER CONSUMPTION	**MWe**	**208.9**	**211.2**
NET ELECTRIC POWER OUTPUT	**MWe**	**759.1**	**767.7**
(Step Up transformer efficiency = 0.997%) (B)	MWe	756.8	765.4
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**63.0%**	**63.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.3%**	**49.8%**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**	**2.01**
CO_2 emission per net power production	**kg/MWh**	**38.7**	**38.3**

Based on the above data, the following considerations can be drawn:

- The higher inlet temperature, combined with the higher expansion ratio, leads to the higher gas turbines power output, which is partially offset by the lower power generation from the steam turbine.
- The gross electrical efficiency gain is around 0.7 percentage points, while the net electrical efficiency is only 0.5 percentage points higher due to the additional consumptions of the ASU, as the oxygen shall be delivered at higher pressure.

- The reduction of the electrical consumptions relevant boiler feed water pumps and cooling water pumps related to the lower steam generation in the HRSGs has a negligible impact on the overall power plant efficiency.

Lower COT case

The case with lower COT is developed by considering the same design bases as reference case for the maximum allowable metal temperature and the compressor ratio.

In fact, as a lower cooling stream flowrate is required to control the maximum metal temperature due to the lower combustion outlet temperature, the exhaust gas temperature remains in the proper range for downstream heat recovery, even with no change of the compressor ratio.

Table 7. Case 1 – Sensitivity to COT– Lower COT case

		BASE CASE	SENSITIVITY CASE (Lower COT)
SENSITIVITY			
Combustion outlet temperature	**°C**	**1533**	**1453**
DESIGN FEATURES			
Compressor outlet pressure	bar	45.8	45.8
Compressor ratio	-	44.5	44.5
Oxygen delivery pressure from ASU	bar	47	47
Turbine inlet temperature	°C	1352	1318
Flue gas temperature (HRSG inlet)	°C	620	610
Flue gas temperature (HRSG outlet)	°C	67	82
PERFORMANCES COMPARISON			
Natural Gas flow rate	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
Gas turbine power output (@ gen terminals)	MWe	611.8	613.6
Steam turbine power output (@ gen terminals)	MWe	356.2	345.1
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**968.0**	**958.7**
Oxy-turbine cycle	MWe	10.3	10.1
Air separation unit	MWe	147.1	147.1
CO2 purification and compression unit	MWe	40.8	40.8
Utility & Offsite Units	MWe	10.8	10.8
ELECTRIC POWER CONSUMPTION	**MWe**	**208.9**	**208.8**
NET ELECTRIC POWER OUTPUT	**MWe**	**759.1**	**749.9**
(Step Up transformer efficiency = 0.997%) (B)	MWe	756.8	747.7
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**63.0%**	**62.4%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.3%**	**48.7%**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**	**2.05**
CO_2 emission per net power production	**kg/MWh**	**38.7**	**39.2**

Based on the above data, the following considerations can be drawn:

- The power production of the gas turbine is only marginally affected by the lower combustion outlet temperature. In fact, with respect to an ideal cycle for which the higher the combustion temperature the higher the cycle efficiency, the requirements to cool down the metal temperature of the gas turbine blades partially mitigates the benefit of higher COT. The main impact is the lower heat recovery in the downstream HRSGs and consequently the lower steam production from the gas turbine.
- As differences in the auxiliary consumptions are negligible, both gross power efficiency and net power efficiency reduction is around 0.6 percentage points.

6.1.3. *Sensitivity to discharge pressure*

Table 8 shows the impacts on key design parameters and plant performances when the expander discharge pressure varies from reference (1.07 bar) to a higher value (2 bar).

Table 8. SCOC-CC – Sensitivity to discharge pressure

		BASE CASE	SENSITIVITY CASE (High Pressure)
SENSITIVITY			
Expander outlet pressure	**bar**	**1.07**	**2**
DESIGN FEATURES			
Compressor ratio	**-**	**44.5**	**44.5**
Compressor outlet pressure	bar	45.8	89.0
Gas turbine rotational speed	RPM	3000	4234
Oxygen delivery pressure from ASU	bar	47	92
Natural gas compressor	-	-	2 x 1120 kWe
PERFORMANCES COMPARISON			
Natural Gas flow rate	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
Gas turbine power output (@ gen terminals)	MWe	611.8	600.4
Steam turbine power output (@ gen terminals)	MWe	356.2	347.3
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**968.0**	**947.7**
Oxy-turbine cycle	MWe	10.3	12.1
Air separation unit	MWe	147.1	154.2
CO2 purification and compression unit	MWe	40.8	32.9
Utility & Offsite Units	MWe	10.8	10.7
ELECTRIC POWER CONSUMPTION	**MWe**	**208.9**	**209.9**
NET ELECTRIC POWER OUTPUT	**MWe**	**759.1**	**737.8**
(Step Up transformer efficiency = 0.997%) (B)	MWe	756.8	735.6
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**63.0%**	**61.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.3%**	**47.9%**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**	**2.09**
CO_2 emission per net power production	**kg/MWh**	**38.7**	**39.8**

As summarised in the above Table 8, the sensitivity case is developed by considering the same compression/expansion ratio of the base case, in order to keep the exhaust gas temperature within the suitable range for the downstream HRSG. As a consequence, the gas turbine compressor discharge pressure and the combustor pressure increase to around 90 bar against the 45 bar of the base case.

Due to the lower volumetric flow of the flue gas, the gas turbine rotational speed is increased at 4234 RPM against the 3000 RPM of the base case.

The above changes in the gas turbine design imply the following modifications for the process units of the power plant:

- The ASU delivers the oxygen at higher pressure, suitable for the new combustor operating conditions (92 bar).
- As the natural gas is available at power plant battery limits at 70 bar, a natural gas compressor per train shall be considered in the design. The let-down station is not required in this case.
- The pressurised HRSG (operating pressure around 2 bar) and downstream contact cooler is foreseen in the power island design
- The first compressor stage of the CPU shall be designed for a lower compressor ratio.
- A gearbox is required between the gas turbine and its generator. The gearbox will be very large (< 300 MWe) compared to the commercially available equipment (around 140 MW), but their feasibility is assessed at feasibility study level.

Based on the above data, the following considerations can be drawn:

- The plant gross electrical efficiency is reduced by more than one (1) percentage point with respect to the base case, due to the lower power generation from both the gas and steam turbines.
- The auxiliary consumptions are almost the same as the base case because the additional ASU and NG compressor consumptions are offset by the lower power demand of the CPU. No significant impact on utility capacity and related consumptions is foreseen.
- As for the above the electrical efficiency is reduced of around 1.4 percentage point.
- It has to be noted that the main advantage of the configuration at high discharge pressure is the lower space requirement. In fact, the equipment should be more compact as higher pressure leads to lower volume.

6.1.4. *Sensitivity to compressor ratio*

Table 9 shows the impacts on key design parameters and plant performance when the compressor/expander pressure ratio varies from reference (44.5) to lower values (39.5 and 34.5).

Table 9. SCOC-CC – Sensitivity to pressure ratio

		BASE CASE	SENSITIVITY CASE (lower ratio)	SENSITIVITY CASE (lower ratio)
SENSITIVITY				
Compressor ratio	**-**	**44.5**	**39.5**	**34.5**
DESIGN FEATURES				
Compressor outlet pressure	bar	45.8	40.7	35.5
Gas turbine rotational speed	RPM	3000	3000	3000
Oxygen delivery pressure from ASU	bar	47	42	36
Exhaust gas temperature (HRSG inlet)	°C	620	641	666
Exhaust gas temperature (HRSG outlet)	°C	81	77	73
Heat duty available in the HRSG	MWth	958	970	984
PERFORMANCES COMPARISON				
Natural Gas flow rate	t/h	118.9	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**	**1536**
Gas turbine power output (@ gen terminals)	MWe	611.8	599.2	585.6
Steam turbine power output (@ gen terminals)	MWe	356.2	362.0	374.3
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**968.0**	**961.2**	**959.9**
Oxy-turbine cycle	MWe	10.3	10.2	10.5
Air separation unit	MWe	147.1	142.4	139.9
CO2 purification and compression unit	MWe	40.8	40.8	40.8
Utility & Offsite Units	MWe	10.8	10.8	10.8
ELECTRIC POWER CONSUMPTION	**MWe**	**208.9**	**204.1**	**202.0**
NET ELECTRIC POWER OUTPUT	**MWe**	**759.1**	**757.1**	**758.0**
(Step Up transformer efficiency = 0.997%) (B)	MWe	756.8	754.8	755.7
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**63.0%**	**62.6%**	**62.5%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.3%**	**49.13%**	**49.18%**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**	**2.04**	**2.03**
CO_2 emission per net power production	**kg/MWh**	**38.7**	**38.8**	**38.7**

As summarised in the above Table 9, the sensitivity case is developed by considering the expander discharge pressure of the base case, while modifying the combustor pressure, and consequently the delivery pressure of the natural gas and the oxygen from the ASU. As for that the main modification of the process units of the power plant is in the ASU, because oxygen shall be delivered at lower pressure.

Based on the above data, the following considerations can be drawn:

- Reducing the compressor/expansion ratio leads to a lower gas turbine power output, while the steam turbine power production increases due to higher heat recovery in the HRSG (higher flue gas discharge temperature from the gas turbine combined with lower flue gas discharge temperature from the HRSG).

- The auxiliary consumptions decrease with the compressor ratio due to the lower power consumption of the ASU (lower oxygen delivery pressure).
- The best net electrical efficiency corresponds to the base case, achieving the optimised balance between gas turbine efficiency (and hence power production) and the heat recovery in the downstream HRSG and consequently the steam turbine power production.
- Lower efficiencies are achieved in the two cases at lower pressure ratio. The non-monotone behaviour of the efficiency is mainly due to the parameter settings in the calculation model for the gas turbine, as some design parameters are set discretely.

7. Environmental impact

The oxy-combustion gas turbine plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 10 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power island and in the CO_2 purification unit.

Table 10. Case 1 – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	46,330
Flow, Nm^3/h	26,000
Composition	(%mol)
Ar	21.99
N_2	17.26
O_2	3.50
CO_2	57.25
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from flue gas final contact cooler and CO_2 purification unit) are treated to recover water, so the main liquid effluent is cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 320 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	May-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 1 - SCOC-CC	APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	GAS TURBINE							
PK- 3101-1/2	**Gas turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
	Gas turbine		310 MWe					*One per train, two in total*
	Gas turbine generator							*One per train, two in total*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 1 - SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	May-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	HEAT RECOVERY STEAM GENERATOR							
PK- 3201-1/2	**Heat recovery steam generator**	Horizontal, Natural Circulated, 3 Pressure Levels, Simple Recovery, Reheated						***2 x 50% HRSG package***
	Each including:							
D- 3201	HP steam drum							
D- 3201	MP steam drum							
D- 3201	LP steam drum with degassing section							
E- 3201	HP Superheater 2nd section							
E- 3202	MP Reheater 2nd section							
E- 3203	HP Superheater 1st section							
E- 3204	MP Reheater 1st section							
E- 3205	HP Evaporator							
E- 3206	MP Superheater							
E- 3207	HP Economizer 2nd section							
E- 3208	LP Superheater							
E- 3209	MP Evaporator							
E- 3210	HP Economizer 1st section							
E- 3211	MP Economizer							
E- 3212	LP Evaporator							
E- 3213	Condensate heater							
X- 3201	HP steam desuperheater							
X- 3202	MP steam desuperheater							
X- 3203	Start-up Stack							*Including silencer and CEMS*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 1 - SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	May-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	HEAT RECOVERY STEAM GENERATOR							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	HP BFW pumps	Centrifugal	380 m3/h x 2180 m	2750 kW				*One operating one spare, per each train*
P- 3202 A/B	MP BFW pumps	Centrifugal	50 m3/h x 510 m	90 kW				*One operating one spare, per each train*
	HEAT EXCHANGER							
	Blowdown cooler							
	DRUM							
	Continuous Blowdown drum							
	Intermittent Blowdown drum							
	PACKAGES (Common to both train)							
PK- 3202	**Fluid Sampling Package**							
PK- 3203	**Phosphate Injection Package**							
	Phosphate storage tank							
	Phosphate dosage pumps							*One operating one spare*
PK- 3204	**Oxygen scavenger Injection Package**							
	Oxygen scavenger storage tank							
	Oxygen scavenger dosage pumps							*One operating one spare*
PK- 3204	**Amine Injection Package**							
	Amine storage tank							
	Amine dosage pumps							*One operating one spare*
PK- 3101-1/2	**Indirect contect cooler**							

		REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
FOSTER WHEELER	CLIENT: IEAGHG	DATE	May-14	Feb-15		
	LOCATION: The Netherlands	ISSUED BY	NF	NF		
	PROJ. NAME: Oxy-turbine power plants	CHECKED BY	LM	LM		
	CONTRACT N.: 1-BD-0764 A	APPROVED BY	LM	LM		
	CASE.: 1 - SCOC-CC					

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	STEAM TURBINE							
PK- 3301	**Steam turbine and Generator Package**							***1 x 100% package***
	Including:							
ST- 3301	Steam turbine		360 MWe					*Including:* *Lube oil system* *Cooling* *Idraulic control system* *Seals system* *Drainage system*
G- 3301	Steam turbine generator		450 MVA					*Including relevant auxiliaries*
	Inter/after condenser							
	Gland Condenser							
PK- 3302	**Steam Condenser Package**							***1 x 100% condenser package***
	Each including:							
	Steam condenser		510 MWth					*Including:* *Condenser hotwell* *Ejector* *Start-up Ejector*
PK- 3303	**Steam Turbine by-pass system**							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3301 A/B	Condensate pumps	Centrifugal	1075 m3/h x 180 m	670 kW				*One operating one spare*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 1 - SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	May-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 4000 - CO2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	CO2 COMPRESSION AND PURIFICATION							
PK - 4001	**CO2-rich gas compression**							
	Including:							
	Raw flue gas compressors							
	- *Raw flue gas compressor #1*	axial	Flowrate: 2 x 90,000 Nm3/h Pin: 1.02 bar; Pout: 15 bar Compression ratio: 14.7	*2 x 12.5 MWe*				*2x50%*
	- *Raw flue gas compressor #2*	axial	Flowrate: 2 x 96,000 Nm3/h Pin: 14.4 bar; Pout: 35 bar Compression ratio: 2.44	*2 x 3.5 MWe*				*2x50%*
	Condesate separators							
	Intercoolers							
	Inert gas heater							
	Oxygen heater							
	Cooling water intercoolers							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO2 liquid separator							
	- CO2 distillation column							
	- CO2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MWe*				*2x50%*
	- CO2 compressor 2	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6.5 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		2400 kW					*1x100%*
	- Overhead recycle compressors (optional)							
	- Intercoolers							
	Inert gas heater							
	NG heater							
	Cooling water intercoolers							

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 1 - SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	May-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 5000 - Air Separation Unit

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5470 t/d Oxygen purity: 97%mol Oxygen pressure: 50 bar					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 35.5 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		9.5 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 1 - SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	May-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower** including: Cooling water basin	Natural draft	1 x 950 MWth Diameter: 115 m, Height: 180 m, Water inlet: 17 m				concrete	
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/…/I	**Cooling Water Pumps**	Centrifugal	15,000 m3/h x 40 m	2000 kW				*Eight in operation, one spare*
P- 6002 A/B	**Cooling tower make up pumps**	Centrifugal	2,150 m3/h x 30 m	250 kW				*One operating, one spare*
	PACKAGES							
	Cooling Water Filtration Package Cooling Water Sidestream Filters		11600 m3/h					
	Sodium Hypochlorite Dosing Package Sodium Hypochlorite storage tank Sodium Hypochlorite dosage pumps							
	Antiscalant Package Dispersant storage tank Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
PK- 6002	Let down station							
	RAW WATER SYSTEM							
PK- 6003	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Potable water pumps	centrifugal						*One operating, one spare*

FOSTER WHEELER	CLIENT: IEAGHG				REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands				DATE	May-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants				ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A				CHECKED BY	LM	LM		
	CASE.: 1 - SCOC-CC				APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	DEMINERALIZED WATER SYSTEM							
PK- 6004	Demin Water Package, including:							
	- Multimedia filter							
	- Reverse Osmosis (RO) Cartidge filter							
	- Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	CASE 2: NET POWER
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	N. FERRARI
CHECKED BY	:	L. MANCUSO
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 2 of the study, which is the NET Power gas-fired power plant, with cryogenic purification and separation of the carbon dioxide.

To complete the assessment for this case, Foster Wheeler has received assistance from NET Power with respect to the definition of the NET Power process configuration. The configuration shown in this study, however, does not necessarily represent the final design that NET Power is currently developing and intends to commercialize; it is a rational design originating from the simplified information that is in the public domain (e.g. literature data, patents, conference proceedings) and engineering modelling that can be done with the commercially available software. Furthermore, Foster Wheeler was informed by NET Power that there are several design features and trade secret techniques that are part of NET Power's intellectual properties and, as such, were not be able to be disclosed in this public feasibility study. Consequently, it was not possible for Foster Wheeler to replicate the performance of the NET Power system, as claimed in recent literature papers or conference proceedings. Further explanations on these differences are given in a specific section of this chapter (section 6.1), purposely prepared by NET Power for this specific study.

The plant of this study case is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected NET Power plant configuration is based on two parallel trains, each composed of one oxy-combustion direct-fired sCO_2 turbine (same thermal input of the other study cases), downstream main heat exchanger and recompression loop.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 2 – Unit arrangement

Unit	Description	Trains
3000	Power and CO_2 cycle	
	NG compressor	2 x 50%
	Direct-fired sCO_2 Turbine	2 x 50%
	Recycled gas compression and pumping loop	2 x 50%

Unit	Description	Trains
	Main heat exchanger (MHE)	2 x 50%
4000	CO_2 purification and compression	
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power and CO_2 cycle

The power island is based on the configuration as developed by Foster Wheeler with the support of NET Power, based on a direct-fired sCO_2 turbine at high inlet pressure and discharge pressure in the range of 30-50 bar.

The unit is mainly composed by two trains, each including:

- One oxy-combustion direct-fired sCO_2 turbine.
- One main heat exchanger (MHE), for recycled gas pre-heating, compact multi-channel plate-fin type.
- One recycled gas compression loop.

General information relevant to the NET Power cycle is reported in chapter D, section 2.1.2, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.2.1. Direct-fired sCO_2 Turbine expander design features

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the stream at the direct-fired sCO_2 turbine boundary are summarised in the following Figure 1.

The natural gas from the metering station is compressed to 310 bar and injected in the combustion chamber at 145°C resulting from the compression of the natural gas from the grid pressure up to the pressure level of the combustion chamber.

Oxygen is delivered from the ASU at around 100-120 bar pressure level and then combined with a fraction of recycle stream. The resulting oxidant stream is compressed up to 305 bar and heated up to 720°C in the MHE before being fed to the combustor.

The high pressure recycled CO_2-rich stream is heated up at a temperature of 720°C against the exhaust gas in MHE. The combined recycled gas and oxidant stream flowrate to the combustor is set in order to have a combustion outlet temperature of 1150°C.

The temperature of the recycled stream is fixed by the minimum approach of 5°C considered for the design of the MHE.

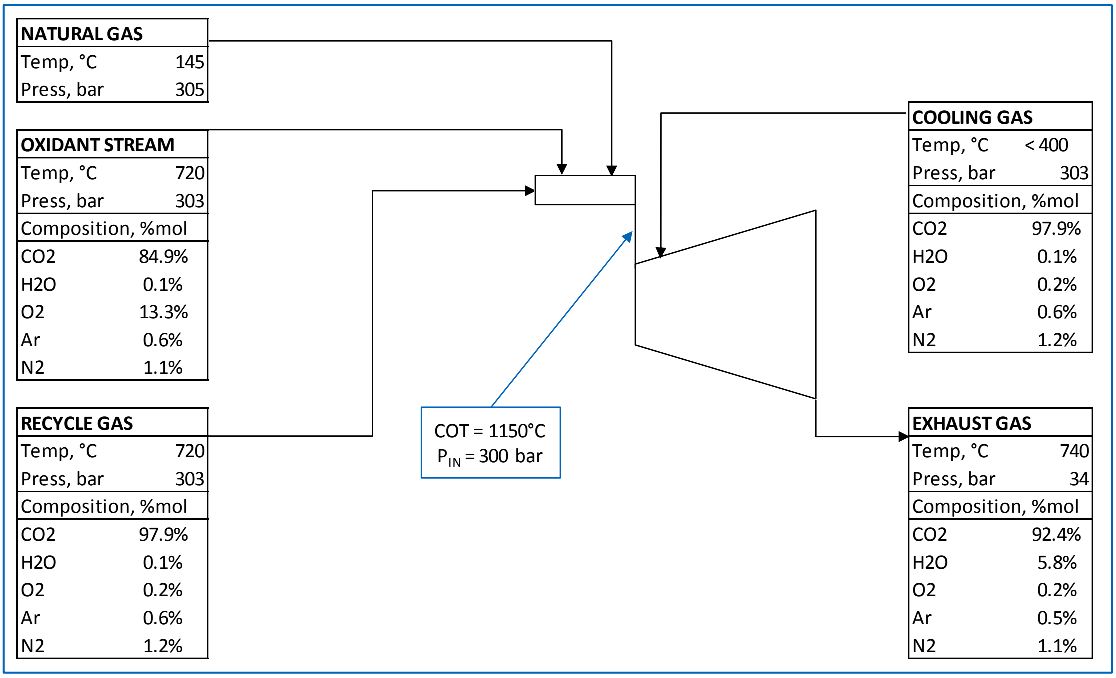


Figure 1. Direct-fired sCO_2 turbine

The combustion products enter the expander at 300 bar and 1150°C. The pressure ratio of the NET Power cycle considered in this study is around 9, in order to have the final CO_2 stream at 34 bar as required by the downstream CO_2 purification unit.

In addition to the recycled flowrate to the combustor chamber, required to control the combustion outlet temperature at 1150°C, an additional cooling stream is required to provide the required cooling flow to the turbine blade in order to maintain the blade metal temperature lower than 860°C. This stream is diverted from the recycle stream and taken through the MHE as it has to be heated-up to lower temperature (lower than 400°C).

The exhaust gases are discharged from the high pressure direct-fired sCO_2 turbine at 740°C.

2.2.2. *Main Heat Exchanger (MHE)*

The exhaust gases from the turbine enter the MHE at 740°C. The heat available from the exhaust gas is recovered heating up the three recycle streams, i.e. the CO_2-rich stream recycled to the combustor, the oxidant stream recycled to the combustor and the CO_2-rich stream to be used for turbine blade cooling.

Heat is also recovered from the heated compressed air from the main air compressor of the air separation unit.

The final exhaust gases from the regenerative heat section are cooled down and water is condensed against cooling water. Most of the exhaust gas from the condenser is recycled back through the recycle gas compression loop, while the net product stream is sent to the downstream CO_2 purification and compression unit. Condensed water is sent to the waste water treatment for treatment and recovery.

2.2.3. Recycle gas compression loop

Recycle gas re-compression is performed through four stage intercooled compressor and two intercooled pumping stages. Inter cooling is performed with cooling water.

Considering the exhaust gas recycle composition and the temperature achieved with cooling water, exhaust gas has to be compressed up to around 80 bar before being pumped.

The first pumping stage increases the recycle stream pressure up to 100-120 bar. The stream is then diverted into three streams.

- Around 45-50% of the flowrate is the CO_2-rich stream recycle to the combustor, after being pumped in the final pumping stage to around 305 bar and preheated in the MHE;
- Around 10-12% of the flowrate is used as cooling stream of the turbine blade after being pumped in the final pumping stage;
- The remainder is combined with high purity oxygen, resulting in the oxidant stream to be recycled back to the combustor after final compression up to 305 bar and pre-heating in the MHE.

2.2.4. Heat integration

The power cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular, heat available at high temperature level from the turbine exhaust is used to provide heat required in the CPU mainly for TSA regenerator heating and inert gas heating before expansion.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 99.5%mol. O_2 purity and around 100-120 bar.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following main systems:

- TSA unit;
- Auto-refrigerated inerts removal (high-purity oxygen configuration), including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Due to the design feature of the NET Power cycle and in particular the selection of the turbine discharge pressure in line with the feeding pressure to the auto refrigerated inerts removal section, the CPU for this particular case does not include the raw gas compression section.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

It has to be highlighted that NET Power is developing a system also including the purification of the CO_2 product, but no specifics of this process are disclosed publicly. Therefore, for this study a generic auto-thermal process has been considered, while the potential for optimisation and integration in process fully developed by NET Power are addressed in section 6.1.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

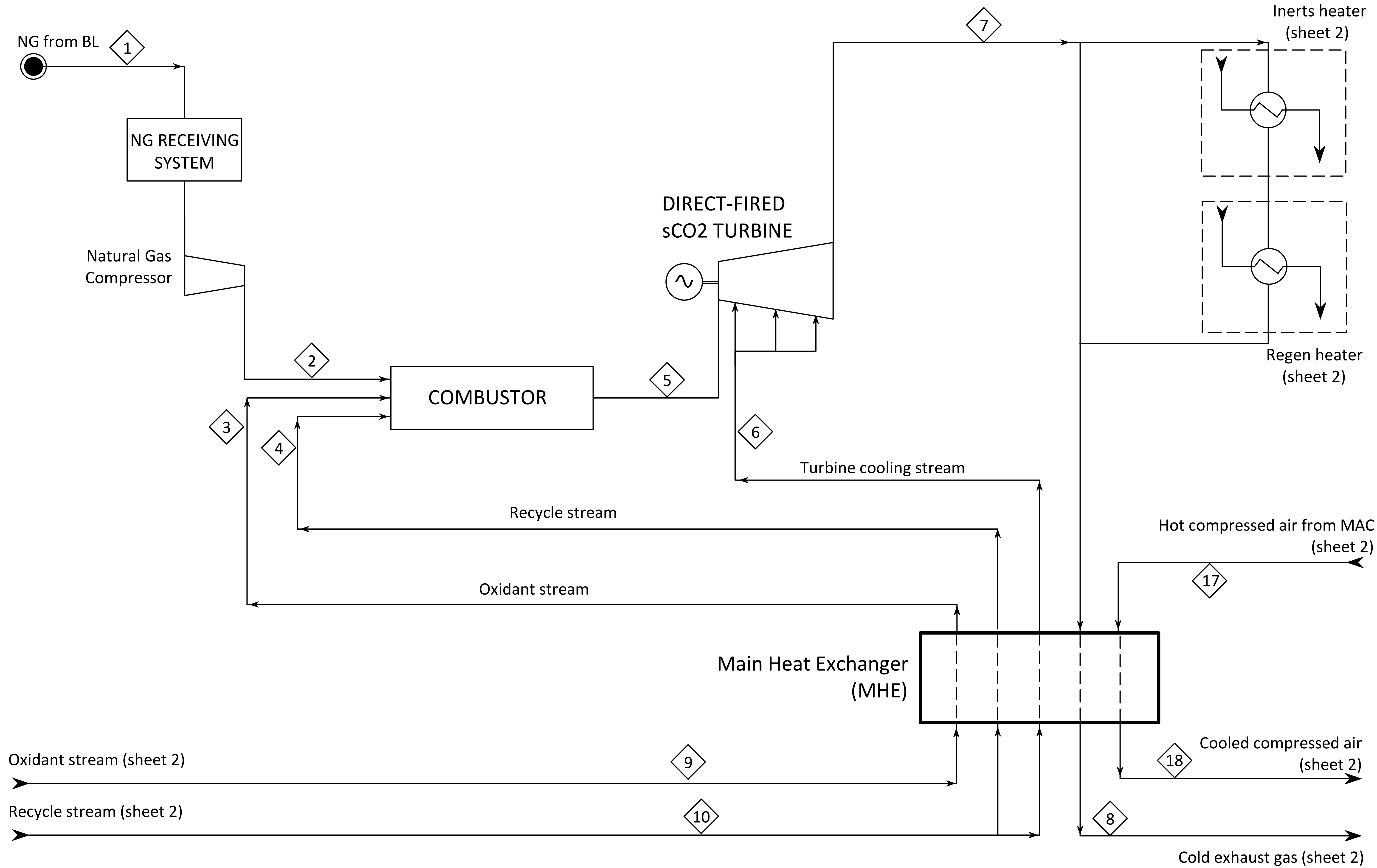
Basin diameter	110 m
Cooling tower height	180 m
Water inlet height	17 m

- Natural gas metering station;
- Raw water system;
- Waste Water Treatment
- Firefighting system;
- Instrument and Plant air.

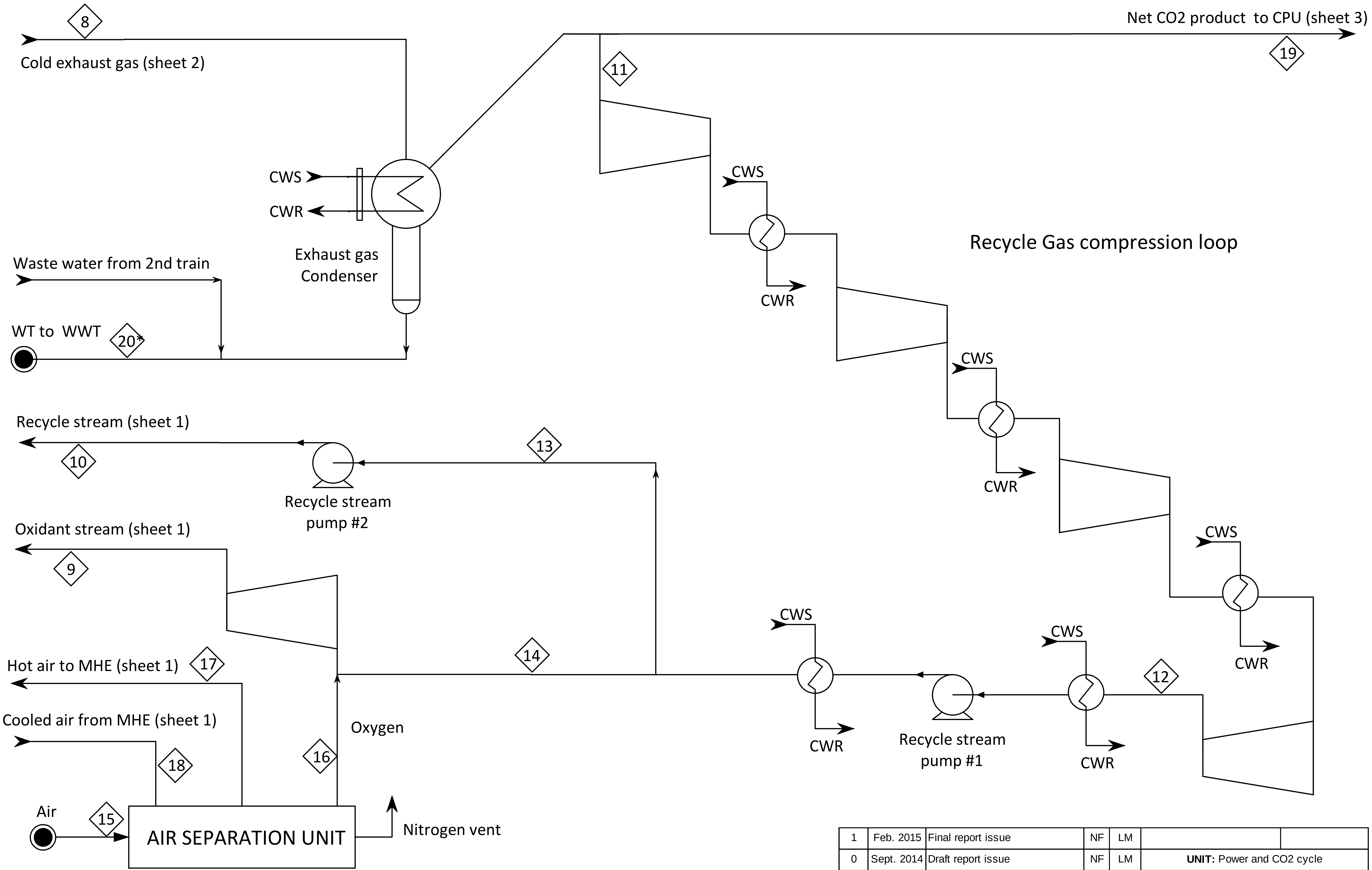
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

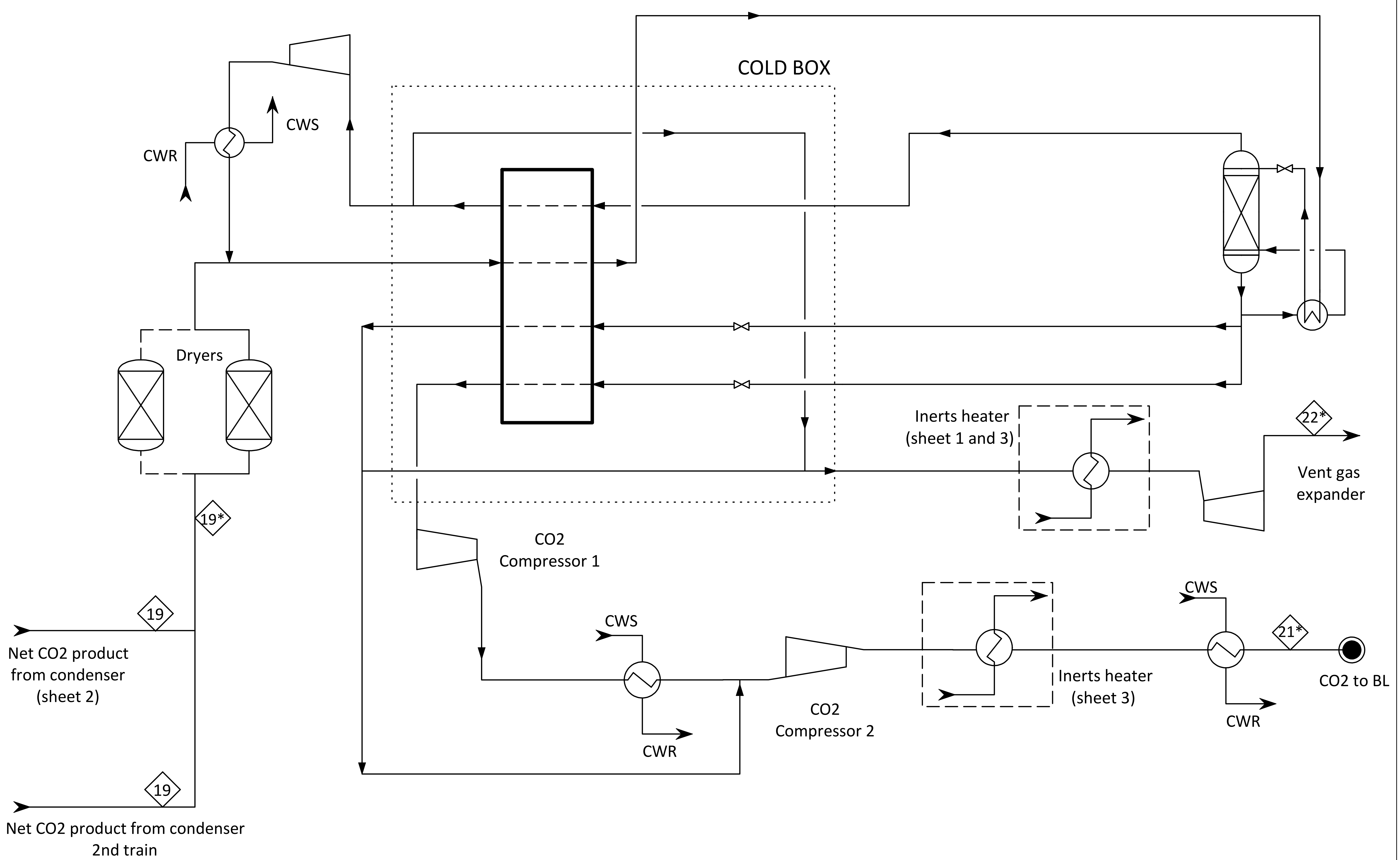
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



1	Feb. 2015	Final report issue	NF	LM		
0	Sept. 2014	Draft report issue	NF	LM	**UNIT:** Power and CO2 cycle	
Rev.	Date	Comment	By	Appr	**CASE:** 2 - NET POWER	Sheet 01 of 03



1	Feb. 2015	Final report issue	NF	LM		
0	Sept. 2014	Draft report issue	NF	LM	**UNIT:** Power and CO2 cycle	
Rev.	Date	Comment	By	Appr	**CASE:** 2 - NET POWER	Sheet 01 of 03



1	Feb. 2015	Final report issue	NF	LM		
0	Sept. 2014	Draft report issue	NF	LM	**UNIT:** CO2 Purification and compression	
Rev.	Date	Comment	By	Appr	**CASE:** 2 - NET POWER	Sheet 03 of 03

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

	Case 2 - NET Power - HEAT AND MATERIAL BALANCE		REVISION	0	1	
	CLIENT :	IEAGHG	PREP.	NF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	September 2014	November 2014	

HEAT AND MATERIAL BALANCE
NET POWER PROCESS

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	HP Natural Gas to combustor	Oxidant stream to combustor	Recycle stream to combustor	Exhaust gas from combustor	Direct-fired sCO2 turbine cooling stream	Exhaust gas to MHE	Cold exhaust gas from MHE	Oxidant stream to MHE	Recycle stream to MHE
Temperature (°C)	15	145	720	720	1150	< 400	740	55	45	50
Pressure (bar)	70	305	303	303	300	303	34	33	305	305
TOTAL FLOW										
Mass flow (kg/h)	59,470	59,470	2,206,170	2,287,180	4,552,820	521,750	5,074,570	5,074,570	2,206,170	2,808,930
Molar flow (kmol/h)	3,300	3,300	52,300	52,300	108,055	11,930	119,985	119,985	119,985	64,230
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	59,470	59,470	2,206,170	2,287,180	4,552,820	521,750	5,074,570	5,074,570	2,206,170	2,808,930
Molar flow (kmol/h)	3,300	3,300	52,300	52,300	108,055	11,930	119,985	119,985	119,985	64,230
Molecular Weight (kg/kmol)	18.0	18.0	42.2	43.7	42.1	43.7	42.3	42.3	18.4	43.7
Composition (%mol)	as assigned	as assigned								
Ar			0.53%	0.57%	0.53%	0.57%	0.54%	0.54%	0.53%	0.57%
CO_2			84.94%	97.88%	91.80%	97.88%	92.41%	92.41%	84.94%	97.88%
H_2O			0.13%	0.15%	6.36%	0.15%	5.74%	5.74%	0.13%	0.15%
N_2			1.05%	1.18%	1.11%	1.18%	1.12%	1.12%	1.05%	1.18%
O_2			13.34%	0.21%	0.20%	0.21%	0.20%	0.20%	13.34%	0.21%
Total			100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 2 - NET Power - HEAT AND MATERIAL BALANCE		REVISION	0	1	
	CLIENT :	IEAGHG	PREP.	NF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	September 2014	November 2014	

HEAT AND MATERIAL BALANCE
NET POWER PROCESS

STREAM	11	12	13	14	15	16	17	18	19*	20*
	Total recycle stream to compression	Total recycle stream from compression	Recycle stream to final pumping stage	Stream to oxygen mixer	Air to ASU	Oxygen from ASU	Hot compressed air to MHE	Cold air from MHE	Net CO2 product to CPU	Water to WWT
Temperature (°C)	29	43	26	26	9	15	275	55	29	25
Pressure (bar)	33	80	100-120	100-120	amb	100-120	7.5	7.3	33	2.5
TOTAL FLOW										
Mass flow (kg/h)	4,793,710	4,793,710	2,808,930	1,984,780	994,520	221,390	994,520	994,520	319,860	241,200
Molar flow (kmol/h)	109,615	109,615	64,230	45,385	34,460	6,915	34,460	34,460	7,330	13,390
LIQUID PHASE										
Mass flow (kg/h)										241,200
GASEOUS PHASE										
Mass flow (kg/h)	4,793,710	4,793,710	2,808,930	1,984,780	994,520	221,390	994,520	994,520	319,860	
Molar flow (kmol/h)	109,615	109,615	64,230	45,385	34,460	6,915	34,460	34,460	7,330	
Molecular Weight (kg/kmol)	43.7	43.7	43.7	43.7	28.9	32.0	28.9	28.9	43.6	
Composition (%mol)										
Ar	0.57%	0.57%	0.57%	0.57%	0.92%	0.30%	0.92%	0.92%	0.57%	0.00%
CO_2	97.88%	97.88%	97.88%	97.88%	0.04%	0.00%	0.04%	0.04%	97.88%	0.00%
H_2O	0.15%	0.15%	0.15%	0.15%	0.97%	0.00%	0.97%	0.97%	0.15%	100.00%
N_2	1.18%	1.18%	1.18%	1.18%	77.32%	0.20%	77.32%	77.32%	1.18%	0.00%
O_2	0.21%	0.21%	0.21%	0.21%	20.75%	99.50%	20.75%	20.75%	0.21%	0.00%
Total	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 2 - NET Power - HEAT AND MATERIAL BALANCE		REVISION	0	1	
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	NF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	September 2014	November 2014	

HEAT AND MATERIAL BALANCE
NET POWER PROCESS

STREAM	21*	22*								
	CO2 to BL	Inert stream								
Temperature (°C)	30	80								
Pressure (bar)	110	1.1								
TOTAL FLOW										
Mass flow (kg/h)	284,020	35,840								
Molar flow (kmol/h)	6,455	870								
LIQUID PHASE										
Mass flow (kg/h)	284,020									
GASEOUS PHASE										
Mass flow (kg/h)		35,840								
Molar flow (kmol/h)		870								
Molecular Weight (kg/kmol)		41.2								
Composition (%mol)										
Ar	0.00%	3.67%								
CO_2	99.83%	81.38%								
H_2O	-	-								
N_2	0.16%	9.96%								
O_2	0.01%	4.99%								
Total	100.00%	100.00%								

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 2 – Water consumption summary

CLIENT: IEAGHG		REVISION	0
PROJECT NAME: Oxy-turbine power plant		DATE	Sep-14
PROJECT No. : 1-BD-0764A		MADE BY	NF
LOCATION : The Netherlands		APPROVED BY	LM

Case 2 - NET POWER cycle

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water [DT = 11°C] [t/h]
3000	POWER AND CO2 CYCLE			
	Recycle gas compressor intercooler			52,210
	Condenser			7,910
	Turbine and generator Auxiliaries			4,360
5000	AIR SEPARATION UNIT			
	MAC aftercoolers			2,800
	BAC aftercoolers			1,440
4000	CO_2 PURIFICATION UNIT			
	CO2 purification unit			2,380
6000	UTILITY and OFFSITE UNITS			
	Cooling Water System	1,280		
	Demineralized water unit			
	Waste Water Treatment and Condensate Recovery	-230		
	Balance of plant			
	BALANCE	1,050	0	71,100

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 2 – Electrical consumption summary

FOSTER WHEELER

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Sep-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 2 - NET POWER cycle

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**POWER AND CO2 CYCLE**	
	Natural gas compressor	**8,600**
	Turbine Auxiliaries + generator losses	**5,260**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**150,000**
	Booster air compressor and miscellanea	**20,900**
4000	**CO_2 PURIFICATION UNIT**	
	Raw gas compression section	**0**
	Autorefrigerated inerts removal unit compression consumption	**14,230**
	Autorefrigerated inerts removal unit expander production	**-1,830**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**8,900**
	Balance of plant	**1,460**
	BALANCE	**207,520**

6. Overall performance

The following table shows the overall performance of Case 2, including CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Sep-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 2 - NET POWER cycle

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
Direct-fired sCO2 turbine power output	MWe	1263.9
Recycle gas compression train	MWe	-207.9
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1056.0**
Power cycle (including NG compressor)	MWe	13.9
Air separation unit	MWe	170.9
CO2 purification and compression unit	MWe	12.4
Utility & Offsite Units	MWe	10.4
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**62.1%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**49.9%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6444
CO_2 removal efficiency	**%**	**90.0**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**
CO_2 emission per net power production	**kg/MWh**	**36.8**

6.1. Comparison with cycle developer results

NET Power has provided a detailed commentary on the performance and configuration of the NET Power based power plant as modelled in this study. This is included as ATTACHMENT A.1.

6.2. Performance sensitivity to significant design parameters

In addition to the base case, the plant performance has been evaluated while modifying some key process design parameters. The following Table 4 summarises the sensitivity cases assessed for the NET Power cycle, highlighting the main design features and performance figures affected by each modified parameter.

Table 4. NET Power – Sensitivity cases

Case 2 – Sensitivity to COT		
Base case figure	Sensitivity	Impact
T_{ref} = 1150°C	T = 1100°C	• Increased exhaust gas recycle (the increased flowrate required for combustion temperature control more than offsets the lower requirements for turbine blade cooling) • Lower gross power generation from the turbines (the lower specific production at lower COT more than offsets the increased exhaust gas flow) • Higher recycle gas compressor consumption (increased recycle flow), affecting the turbine net power generation (equal to the plant gross power output)
T_{ref} = 1150°C	T = 1200°C	• Decreased exhaust gas recycle (the decreased flowrate required for combustion temperature control more than offsets the higher requirements for turbine blade cooling) • Lower gross power generation from the turbines (the higher specific production at higher COT does not offset the decreased exhaust gas flow) • Higher net power generation from the turbine due to the lower recycle gas compressor consumption (decreased recycle flow)
T_{ref} = 1150°C Metal temperature = 860°C	T = 1200°C Metal temperature = 950°C	• Increased maximum allowable metal temperature • Decreased exhaust gas recycle due the combined effect of lower recycle flowrate required for COT control and lower requirements for turbine blade cooling) • Higher gross power generation from the turbines (the higher specific production at higher COT

		more than offsets the decreased exhaust gas flow as no negative effect due to the increasing of blade cooling is present, as for the above case) • Higher plant gross power output because, in addition to the increased gross power generation, recycle gas compressor consumptions are also lower
Case 2a – Sensitivity to heat exchanger approach		
Base case figure	Sensitivity	Impact
$\Delta T_{ref} = 5°C$	$\Delta T = 10°C$	• Lower temperature of the exhaust gas recycle to the combustor • Higher temperature of the exhaust gas exiting the MHE; more cooling water required by the condenser
Case 2 – Sensitivity to oxygen purity		
Base case figure	Sensitivity	Impact
O_2 purity ref: 99.5%mol	O_2 purity 97.0%mol	• Lower ASU power demand • Higher inerts content in the exhaust gas flowrate to the auto-refrigerated removal section, leading to higher expander production • Higher power demand for the recycle gas re-compression
Case 2 – Near-zero emission plant: sensitivity to CO_2 purity		
Base case figure	Sensitivity	Impact
CO_2 capture: 90% CO_2 purity: >99.8%	CO_2 capture: ~99.8% CO_2 purity: ~98%	• No CPU included in the plant design • Increased recycle compressor size up to 80 bar as processing also the net CO_2 product stream. • Final CO_2 pump to be included
CO_2 capture: 90% CO_2 purity: >99.8%	CO_2 capture: 98% CO_2 purity: >99.8%	• Dedicated membrane included in the CPU design to recover most of the CO_2 in the inert gas • Permeate stream to be re-compressed recycled back in the exhaust gas stream • Increased cold box and CO_2 compressor size and smaller inert gas expander (higher CPU consumptions)

Case 2 – Sensitivity to cooling tower approach		
Base case figure	Sensitivity	Impact
Cooling tower approach: 7°C	Cooling tower approach: 4°C	• Lower cooling water temperature: 12-23°C (vs. 15-26°C) • Mechanical draft cooling tower selected for this case as an approach as aggressive as 4°C is normally not recommended (even if possible) or natural draft cooling tower due to the cost increase. • Lower compressor power consumptions (mainly recycle gas compressor, but also ASU and CPU compressors) due to the lower gas inlet temperature after intercooling • Additional cooling tower fan power consumption

6.2.1. *Sensitivity to combustor outlet temperature*

As specified in the above section, the combustor outlet temperature (COT) of the study base case is set at 1150°C, controlled by the recycled gas flowrate.

The following two different assessments have been considered, based on the COT variation:

- Analysis of the impacts of COT variation on key design features and plant performances, considering the same turbine design and materials of the reference case. In this scenario, two alternatives with respectively higher and lower COT figures (1200°C and 1100°C) have been evaluated in order to define the optimum operating parameter.

  The results of this analysis are shown in Table 5.

- This analysis has been based on the assumption that allowing higher COT means that the technology development and the material selection for the turbine allow the operation of the oxy-fuel cycle at higher temperature. As for that, also the maximum allowable blade metal temperature is increased from 860°C assumed for the base case to 950°C for the high-temperature case, thus affecting the required cooling flow to the turbine blade and hence the recycle flowrate.

  For this analysis, a case considering both higher COT and higher maximum allowed blade metal temperature is compared with the reference case.

  The results of this analysis are shown in Table 6.

Sensitivity to COT (same turbine material)

Table 5 shows the impacts on key design parameters and plant performances when the COT varies from reference to respectively higher and lower values (1200°C and 1100°C).

Table 5. Case 2 – Sensitivity to COT – Same turbine material

		BASE CASE	SENSITIVITY CASE (Higher COT)	SENSITIVITY CASE (Lower COT)
SENSITIVITY				
Combustion outlet temperature	**°C**	**1150**	**1200**	**1100**
DESIGN FEATURES				
Exhaust gas temperature	°C	740	761	716
Recycle gas temperature	°C	720	754	684
MHE hot side approach	°C	20	7	32
Exhaust gas to condeser temperature	°C	55	56	54
Exhaust gas to turbine	t/h	4,553	4,360	4,754
Recycle gas flowrate (recycle + oxidant)	t/h	4,493	4,301	4,695
Cooling stream	t/h	522	657	389
Cooling stream (percentage of flue gas)	-	11.5%	15.1%	8.2%
Total recycle flowrate (recycle + oxidant + cooling stream)	t/h	5,015	4,958	5,084
PERFORMANCES COMPARISON				
Natural Gas flow rate	t/h	118.9	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**	**1536**
Direct-fired sCO2 turbine power output	MWe	1263.9	1262.8	1255.2
Recycle gas compression train	MWe	-207.9	-205.7	-210.7
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1056.0**	**1057.1**	**1044.5**
Power cycle (including NG compressor)	MWe	13.9	14.0	13.8
Air separation unit	MWe	170.9	170.9	170.9
CO2 purification and compression unit	MWe	12.4	12.4	12.4
Utility & Offsite Units	MWe	10.4	10.3	10.4
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**	**207.6**	**207.6**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**	**849.5**	**836.9**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9	847.0	834.4
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**	**68.8%**	**68.0%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**	**55.1%**	**54.3%**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**	**1.81**	**1.84**
CO_2 emission per net power production	**kg/MWh**	**36.8**	**36.8**	**37.3**

Based on the above data, the following considerations can be drawn:

- As a consequence of the increased combustor outlet temperature (‘Higher COT case’), a lower recycle flowrate is required to control the combustion temperature, while the cooling flowrate increases in order to control the turbine blade metal temperature to the fixed design data. As a consequence of these two opposite effects, the total recycle flowrate slightly decreases at higher COT. The opposite trend is experienced when reducing the combustion outlet temperature (‘Lower COT case’).
- Despite the small changes in the total recycle flowrate, as the cooling flowrate needs to be heated up to a lower figure with respect to the recycle streams to the combustor, the heat exchanged in the MHE varies with the COT.

At lower COT, the heat available from the exhaust gas (and from the ASU) is not enough to heat up the recycle gas up to the maximum temperature level (assumed 5°C lower than the turbine exhaust).

On the other hand, the design with higher COT fully exploits the potentiality of the MHE, heating the lower recycle flowrate required for combustion control up to 7°C lower than the exhaust gas temperature. Further increasing the COT would lead to the opposite effect: as the cooling flowrate increases and the recycle exhaust gas to combustor decreases, the heat available would become greater than required, resulting in a higher cold exhaust gas discharge temperature but not to higher recycle temperature and consequently turbine efficiency.

- Among the above listed cases, the highest turbine gross generation is achieved at the reference COT value of 1150°C.

  In fact, the gross turbine power output depends on two opposite effects: the specific production per tons of hot sCO_2 expanded increases with the COT, but the amount of hot sCO_2 from the combustor decreases as a lower recycle flowrate is required to control the combustor temperature. Even if a higher flowrate is fed to the turbine for blade cooling purpose, the additional flowrate does not offset the reduction of the recycle streams flow to the combustor, also considering that the cooling flow is injected at low temperature, reducing the turbine efficiency.

  At higher COT, the combined effect of the reduced recycle flowrate and the increased cooling stream offsets the higher specific power production.

  At lower COT, the higher mass flowrate expanded does not offset the penalty related to the lower inlet temperature.

  The non-linear behaviour of the turbine gross power production can be explained considering that the recycle streams temperature (and consequently the flowrate) does not vary linearly with COT, but also depends on the temperature-heat profile in the MHE.

  For example, in the lower COT case, the final temperature of the recycle streams is far from the maximum possible level (more than 30°C approach), and consequently the recycle flowrate is lower than the amount ideally achieved if the heat exchanged in the MHE had been enough to achieve lower approach. As for that, the higher mass flowrate expanded in the turbine does not offset the negative effect of the lower specific production, which is not enough to compensate the effect of the lower COT.

  On the other hand, the fact that the cooling stream temperature is kept constant while varying the COT and recycle temperatures adds another source of non-linearity to the turbine behaviour, and it probably explains the reduction of production at higher COT (dominant negative effect of the increased cooling).

- The gross plant power output (i.e. the turbine net power production at generator terminal) increases with the COT, as the power demand of the recycle compressors is lower. In fact, a higher COT corresponds at lower recycle flowrate with a consequently lower compressor power demand.
- The difference between the reference case and the higher COT is limited (lower than 0.1 percentage points efficiency gain), while significant penalty is associated to the lower COT figure (0.7 percentage points efficiency losses with respect to the reference case).
- As the COT variation has a negligible impact on the plant auxiliary consumptions, the net electrical efficiency behaviour is the same as the gross electrical efficiency.
- The main result of this sensitivity is that the best efficiency in a regenerative cycle is achieved optimising not only the turbine design but also fully exploiting the potentially of the regenerative section, as graphically shown in the below Figure 2. The graph also includes an additional point at higher COT (i.e. 1250°C), showing that increasing excessively the COT does not represent a benefit (assuming a constant metal temperature) as the increased cooling requirements offsets the benefit of the increased COT and the heat exchanger approach can not be reduced below the design figure of 5°C. Based on this sensitivity, the estimated optimum figure for the COT is in the range between 1150°C and 1200°C.

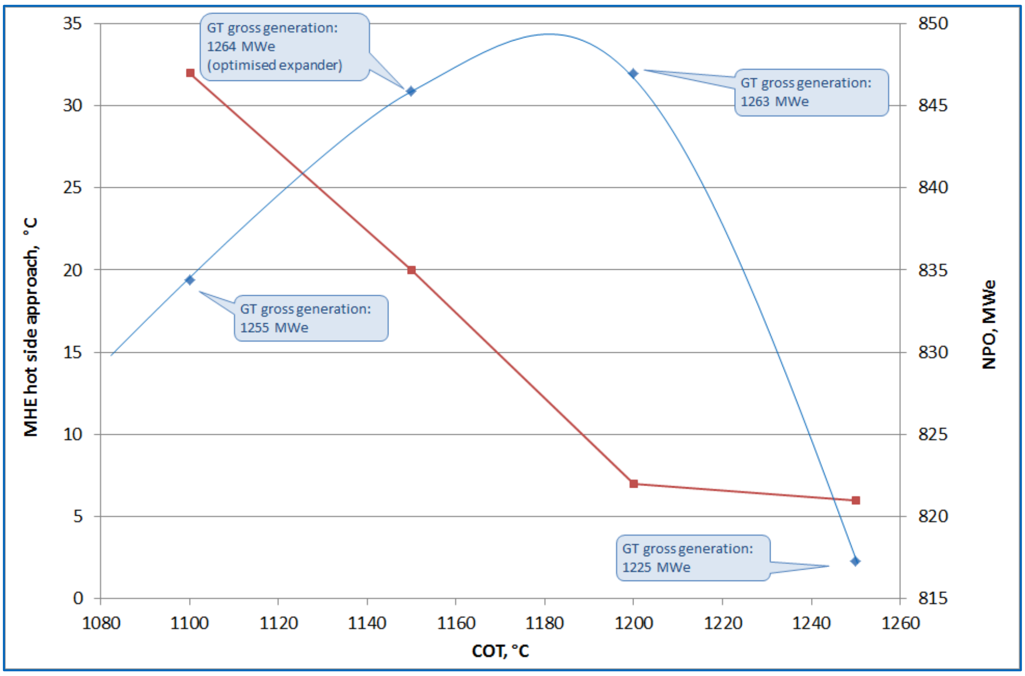


Figure 2. NET Power – COT sensitivity

Increasing COT: high temperature material applied

Table 6 shows the impacts on key design parameters and plant performances when considering high temperature materials for the turbine blades. Key parameter to quantify this effect is the increased maximum allowed turbine metal temperature from the reference figure of 860°C to 950°C, allowing to increase the COT and to reduce the cooling flow rate. As for the COT increment, only a 50°C increment was considered (from 1150°C to 1200°C) so as to keep the turbine outlet temperature within a range reasonably suitable for the MHE.

Table 6. Case 2 – Increasing COT: high temperature material applied

		BASE CASE	SENSITIVITY CASE (high T materials)
SENSITIVITY			
Combustion outlet temperature	**°C**	**1150**	**1200**
Max turbine metal temperature	**°C**	**860**	**950**
DESIGN FEATURES			
Exhaust gas temperature	°C	740	793
Recycle gas temperature	°C	720	767
MHE hot side approach	°C	20	26
Exhaust gas to condeser temperature	°C	55	58
Exhaust gas to turbine	t/h	4,553	4,492
Recycle gas flowrate (recycle + oxidant)	t/h	4,493	4,432
Cooling stream	t/h	522	336
Cooling stream (percentage of flue gas)	-	11.5%	7.5%
Total recycle flowrate (recycle + oxidant + cooling stream)	t/h	5,015	4,768
PERFORMANCES COMPARISON			
Natural Gas flow rate	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
Direct-fired sCO2 turbine power output	MWe	1263.9	1278.6
Recycle gas compression train	MWe	-207.9	-197.5
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1056.0**	**1081.2**
Power cycle (including NG compressor)	MWe	13.9	13.9
Air separation unit	MWe	170.9	170.9
CO2 purification and compression unit	MWe	12.4	12.4
Utility & Offsite Units	MWe	10.4	10.1
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**	**207.4**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**	**873.8**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9	871.2
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**	**70.4%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**	**56.7%**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**	**1.76**
CO_2 emission per net power production	**kg/MWh**	**36.8**	**35.8**

Based on the above data, the following considerations can be drawn:

- The direct-fired sCO_2 turbine gross power generation increases by around 1% if high temperature materials are considered due to the combined effect of the higher COT and the reduced cooling flowrate required for turbine blade cooling (half the flowrate required for the case with same COT but lower metal temperature allowed).
- In addition, as both cooling stream and recycle stream to the combustor decrease, the recycle gas compressor consumptions significantly decrease (around 5% lower).
- As a consequence of the above, the gross electrical efficiency gain is around 1.6 percentage point, when high temperature materials are applied.
- As COT variation does not influence the plant auxiliary consumptions, the same efficiency increase is experienced for the net electrical efficiency.

6.2.2. *Sensitivity to heat exchanger approach*

As specified in the above section, the heat recovery section of the study base case is designed considering an approach temperature of 5°C in the final heat exchanger, thus affecting the recycle gas feeding temperature to the combustor chamber and consequently the flowrate required to control the combustion outlet temperature.

Table 7 shows the impact on plant performance and the key design features when the heat exchanger approach increases from reference (5°C) to a higher figure (10°C).

Table 7. Case 2 – Sensitivity to heat exchanger approach

		BASE CASE	SENSITIVITY CASE
SENSITIVITY			
Heat exchanger approach	**°C**	**5**	**10**
DESIGN FEATURES			
Recycle gas flowrate (recycle + oxidant)	t/h	4,493	4,399
Recycle gas final temperature	°C	720	709
Exhaust gas to gas turbine	t/h	4,553	4,459
Exhaust gas to final cooler temperature	°C	55	64
PERFORMANCES COMPARISON			
Natural Gas flow rate (A.R.)	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
Direct-fired sCO2 turbine power output	MWe	1263.9	1237.6
Recycle gas compression train	MWe	-207.9	-203.5
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1056.0**	**1034.1**
Power cycle (including NG compressor)	MWe	13.9	13.8
Air separation unit + Oxygen compressor	MWe	170.9	170.9
CO2 purification and compression unit	MWe	12.4	12.4
Utility & Offsite Units	MWe	10.4	10.6
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**	**207.6**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**	**826.5**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9	824.0
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**	**67.3%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**	**53.6%**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**	**1.86**
CO_2 emission per net power production	**kg/MWh**	**37.0**	**37.8**

Based on the data listed above the following consideration can be drawn:

- Increasing the heat exchanger approach by around 5°C leads to an efficiency loss of 1.5 percentage points.
- The higher the heat exchanger approach, the lower the recycle final temperature and flowrate and hence the power generation from the turbine.
- On the other hand, the power demand of the recycle gas compressor decreases.
- Exhaust gas exits the MHE at higher temperature, slightly increasing cooling water flowrate to the exhaust gas condenser. Impact on utility unit power demand is negligible.

6.2.3. *Sensitivity to oxygen purity*

In a power plant based on the NET Power cycle, the main driver for the selection of the optimum oxygen purity is to minimise the combined consumption of ASU, CPU and recycle gas re-compression. The higher the oxygen purity, the higher the CPU and ASU power demand and the lower the recycle gas compressor power consumption. In fact:

- As purity is increased above 97%mol, the ASU power demand increases significantly as the distillation changes from a nitrogen/oxygen separation into an oxygen/argon separation, which is characterised by a lower relative volatility.
- In the NET Power configuration, the CPU net power demand increases with the oxygen purity as the only consequence of the reduced inert content in the oxygen stream is the lower power production from the expander.

  It has to be noted that typically in an oxy-combustion power plant, the CPU power consumption increases with the inert content in the oxygen stream as exhaust gas flowrate to the CPU and hence raw gas compressor consumptions increase. In the NET Power configuration, raw gas compression is not included in the CPU as the exhaust gas is made available at the CPU battery limits already compressed at 33 bar.
- The minimum pressure at which the recycle stream has to be compressed before being pumped depends on the CO_2 purity in the recycle stream, and consequently on the inerts content in the oxygen. The higher the CO_2 content in the recycle stream, the lower is the minimum pressure to be achieved with compression and consequently the lower the power demand of the re-compression section.

  More specifically, lowering the oxygen purity from 99.5%mol down to 97%mol, the minimum pressure at which the main recycle stream has to be compressed before being pumped increases from the 80 bar to 120 bar.

Table 8 shows the impacts on key design features and plant performance when the purity of the oxygen produced in the ASU varies from reference (99.5%mol) to lower values (97%mol).

Table 8. Case 2 – Sensitivity to oxygen purity

		BASE CASE	SENSITIVITY CASE (lower purity)
SENSITIVITY			
Oxygen purity	**%mol**	**99.5%**	**97.0%**
DESIGN FEATURES			
Exhaust gas compression configuration			
Final compression pressure	bar	80	120
CO2 content	%mol	97%	93.5%
# compression stages		4	5
# pumping stages		2	1
Raw exhaust gas to CPU	kmol/h	7300	7660
Inert gas	kmol/h	870	1170
PERFORMANCES COMPARISON			
Natural Gas flow rate (A.R.)	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
Direct-fired sCO2 turbine power output	MWe	1263.9	1267.4
Recycle gas compression train	MWe	-207.9	-228.4
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1056.0**	**1039.1**
Power cycle (including NG compressor)	MWe	13.9	13.8
Air separation unit	MWe	170.9	162.5
CO2 purification and compression unit — Compressor	MWe	14.2	14.2
Expander	MWe	-1.8	-2.4
Utility & Offsite Units	MWe	10.4	10.4
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**	**198.5**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**	**840.6**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9	838.1
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**	**67.6%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**	**54.5%**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**	**1.83**
CO_2 emission per net power production	**kg/MWh**	**36.8**	**36.2**

Based on the above data the following considerations can be drawn:

- The higher efficiency is achieved considering a high purity oxygen stream (99.5%mol), as the power saving in the recycle stream re-compression section more than offsets the higher ASU power consumptions.
- Considering plant design features, the configuration with higher oxygen purity implies additional separation stage in the ASU distillation column, but also a simplified recycle gas compressor section (lower number of compression stages) and a smaller and less complicated CPU.

6.2.4. *Near-zero emission plant: sensitivity to CO_2 purity*

Currently there is not either a well-defined legislation or market demand that regulates the CO_2 emissions in term of both carbon tax and CO_2 purity for future CO_2 networks. This section analyses the impact on the plant performances and costs of a NET Power-based plant designed for near-zero emission, with two different CO_2 purity targets, as detailed below.

- *Low CO_2 purity case*

  If a lower CO_2 purity were allowed by regulation, the whole amount of CO_2 generated in the combustion could be captured and stored, achieving almost 100% capture rate. In fact, the net CO_2 product can be diverted from the recycle stream at the required network pressure and sent to B.L. without further purification of components like water and inert gases.

  This plant configuration is particularly suited for the oxy-combustion gas plant, due to the lower inert content in the CO_2, this latter being a direct consequence of the lower oxygen excess required for combustion in the gas turbine (around 3%), with respect to the coal-fired oxy-combustion boiler (around 20-30%).

  It has to be noted also that the higher the purity of the oxygen from the ASU, the higher the quality of the CO_2. As for that, this plant configuration, even if applicable to all the oxy-turbine cycles, is particularly suited for the NET Power cycle, as it achieves the best performance with high purity oxygen (99.5%mol) from the ASU.

- *High capture rate and CO_2 purity case with additional membrane*

  Including dedicated membranes in the CPU to recover the CO_2 in the inert gas and recycling it back to the exhaust gas stream allows achieving high capture rate (around 98%), still maintaining the high CO_2 purity of the base case (<99.8%mol).

The main plant design changes of the near-zero emission plants with respect to the base case are described below.

Low CO_2 purity case

The main design basis for this case is that the net CO_2 product diverted from the gas turbine exhaust stream is delivered at plant BL, and then to the CO_2 distribution network, without additional purification; consequently, the CPU for water and inert gas removal is not required.

The following additional design changes to the power and CO_2 cycle are the direct consequence of the above.

- The capacity of the recycle compressors loop from the exhaust gas condenser up to 80 bar is increased by the amount of CO_2 product. In fact, while in the base case the net CO_2 product is diverted from the exhaust gas stream from the condenser at the pressure required by the cold box in the purification section, in this case it is compressed up to 80 bar in the recycle compressor package. As the net CO_2 product is a small amount with respect to the overall recycle stream (around 3-3.5%), it is not convenient to include a separate compressor dedicated to the CO_2 stream export.

  However, in order to decouple the selection of the pressure of the downstream pumping stage, and consequently of the oxygen delivery pressure from the network pressure, a dedicated pump is foreseen to pump the CO_2 stream to be exported from the compressor discharge up to the network pressure (Figure 3).

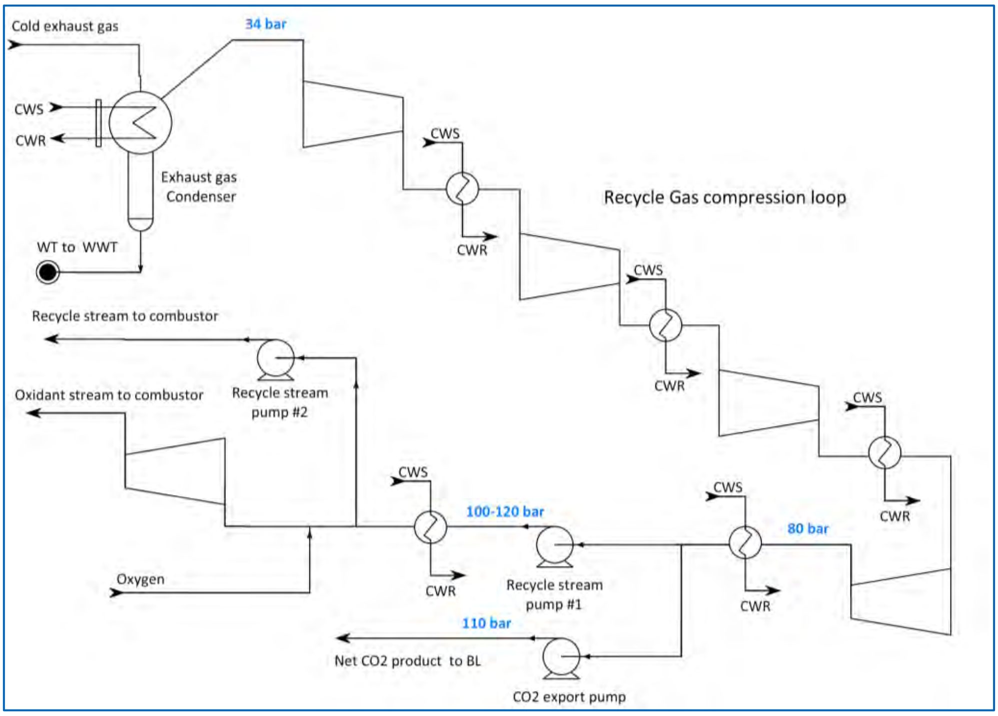


Figure 3. NET Power – near zero emission: low purity case – Recycle compressor

  It has to be noted also that, while in the main case the gas turbine discharge pressure has been set to 34 bar, as required by the cold box in the CPU, in this plant configuration it can be selected within the whole range of feasibility as claimed by NET Power (30-50 bar), thus increasing plant flexibility.

- Due to the removal of the dessicant system, and consequently of both the regenerator and the inert gas heaters, the exhaust gas from the gas turbine is directly fed to the MHE for heat recovery against the recycle gas. The small amount of additional duty available to the MHE leads to a slightly higher recycle temperature and consequently flowrate required to control the combustion outlet temperature at 1150°C, leading to an increased power production from the gas turbine.

High capture rate and CO_2 purity case with additional membrane

The CPU differs from the base case process for the following features:

- The inerts gas from the auto-refrigerated section is processed through dedicated membranes, in order to recover enough CO_2 to meet around 98% carbon capture rate.
- The membrane permeate at low pressure, rich of CO_2, is compressed, recirculated back to the MHE enhancing heat recovery and then mixed with the exhaust gas upstream the condenser.
- As for the above, the design capacity of the CPU, as well as the compression consumption and the heat recovery, is increased with respect to the base case. Vice versa, the inert gas expander is smaller, reducing the internal electrical production of the unit.

The different design of the CPU affects the design of the power and CO_2 cycle, as follows:

- The main impact is the additional hot membrane permeated stream entering the MHE for heat recovery against the recycle streams to the combustor.
- Due to the lower amount of inert gas, the heat required from the exhaust gas from the gas turbine upstream the MHE for inert gas pre-heating before expansion is reduced.
- As for the above, a small amount of additional duty is available to the MHE, leading to a slightly higher recycle temperature and consequently to a higher flowrate for the control of the combustion outlet temperature at 1150°C, and finally to an increased power production from the gas turbine.

The above described design changes are reflected and highlighted in blue in the simplified scheme shown in Figure 4.

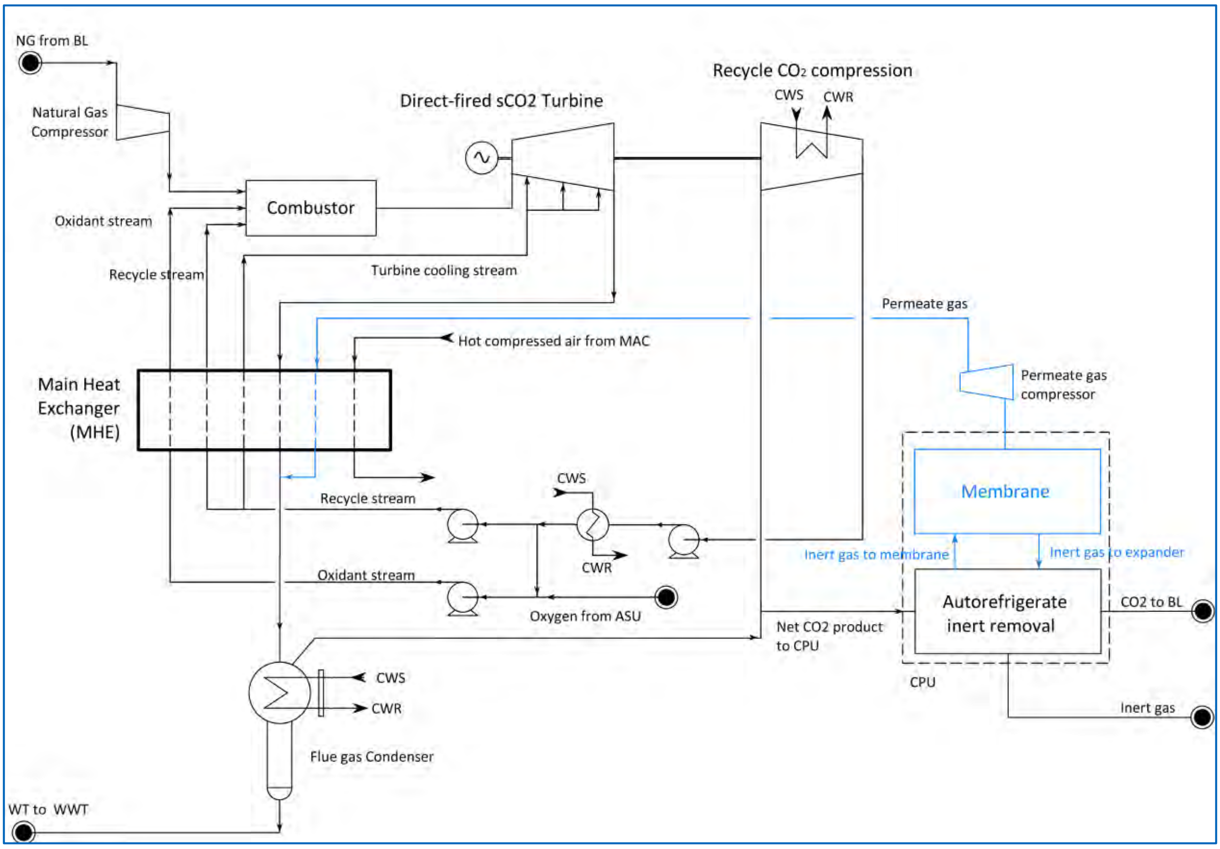


Figure 4. NET Power – near zero emission with additional membrane

It has to be noted that the plant configuration proposed is one of the possible schemes that could be considered for this case; possible alternatives are the following:

- Membrane configuration as for the selected alternative, with rich-CO_2 permeate stream at low pressure, recycled back downstream and mixed with the exhaust gas downstream the MHE. In this case, the permeate compressor is intercooled and its power demand reduce, while there is no impact on the MHE design. On the other hand, heat from the compression is not recovered, the gas turbine power augmentation is lower and additional cooling water demand shall be considered.
- Membrane configuration based on high-pressure rich-CO_2 permeate stream and inert gas at low pressure. In this case, the permeate recycle compressor is smaller and the stream is recycled back downstream the MHE, as the heat available from the compressor is negligible. In addition, the inert gas at low pressure is directly released to the atmosphere, without need for the expander.

Impact on performance and cost of the different alternatives is negligible, also considering that the amount or permeated gas is negligible with respect to the whole amount of exhaust and recycled gas flowrates, corresponding to a smaller compressor power demand and heat recovery.

The configuration with the permeate stream at high pressure and the inert stream directly discharged to atmosphere is less complicated, as no expander is required and there is no impact on the MHE. However, the membrane technology currently developed by the CPU supplier (and for which performance is proven) is now based on the configuration with high pressure inert gas and low pressure permeate stream. As for that, the case included in this report is based on this configuration.

Table 9 compares the CPU material balance and the final CO_2 product characteristics for the above described near-zero emission plants in comparison with the base case.

Table 9. Case 2 – Near-zero emission cases – CPU balance

			BASE CASE 90% CO2 capture CO2 Purity > 99.8%	SENSITIVITY CASE No CPU (100% CO2 capture) ~ 98% CO2 purity	SENSITIVITY CASE CPU with membrane (98% CO2 capture) CO2 Purity > 99.8%
SENSITIVITY					
CO2 purification scheme		**-**	**with distillation column**	**NA**	**with distillation column and membrane**
CO2 capture rate		**-**	**90%**	**~ 100%**	**98%**
CO2 purity		**%mol**	**> 99.8%**	**~ 98%**	**> 99.8%**
CO2 to BL					
Composition					
CO2		%mol	99.83%	97.88%	99.83%
N2 / Ar		%mol	0.16%	1.76%	0.16%
O2		%mol	0.01%	0.21%	0.01%
Water		%mol	-	0.15%	-
Flowrate		kmol/h	6,455	7,296	7,025
CPU balance					
Exhaust gas to CPU	Flowrate	kmol/h	7,330	N/A	7,988
	CO2 content	%mol	97.88%		97.55%
inert gas	Flowrate	kmol/h	870		254
	CO2 content	%mol	81.38%		50.60%
Membrane permeate to MHE (1)	Flowrate	kmol/h	N/A		694
	CO2 content	%mol			93.80%
CO2 product			see above		

(1) Recompressed and sent to the MHE for heat recovery

Table 10 shows the impacts on plant performance for the above described near-zero emission plant, compared with the base case.

Table 10. Case 2 – Near-zero emission cases – plant performance

		BASE CASE 90% CO2 capture CO2 Purity > 99.8%	SENSITIVITY CASE No CPU (100% CO2 capture) ~ 98% CO2 purity	SENSITIVITY CASE CPU with membrane (98% CO2 capture) CO2 Purity > 99.8%
SENSITIVITY				
CO2 purification scheme	**-**	**with distillation column**	**NA**	**with distillation column and membrane**
CO2 capture rate	**-**	**90%**	**~ 100%**	**98%**
CO2 purity	**%mol**	**> 99.8%**	**~ 98%**	**> 99.8%**
PERFORMANCES COMPARISON				
Natural Gas flow rate	t/h	118.9	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**	**1536**
Direct-fired sCO2 turbine power output	MWe	1263.9	1266.3	1266.3
Recycle gas compression train	MWe	-207.9	-214.4 (1)	-211.5
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1056.0**	**1051.9**	**1054.8**
Power cycle (including NG compressor)	MWe	13.9	13.9	13.9
Air separation unit	MWe	170.9	170.9	170.8
CO2 purification and compression unit	MWe	12.4	4.1 (1)	16.2 (2)
Utility & Offsite Units	MWe	10.4	10.3	10.5
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**	**199.1**	**211.4**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**	**852.8**	**843.4**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9	850.2	840.9
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**	**68.5%**	**68.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**	**55.3%**	**54.7%**
Equivalent CO_2 flow in fuel	kmol/h	7159	7159	7159
Captured CO_2	kmol/h	6444	7142	7013
CO_2 removal efficiency	**%**	**90.0**	**99.8**	**98.0**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**	**1.81**	**1.83**
CO_2 emission per net power production	**kg/MWh**	**36.8**	**0.9**	**6.7**

(1) Total recycle compressor power demand: 218.5 MW, splitted in the table with the only purpose to highlight the comsumption relevant to the net CO2 to BL.
(2) Including the permeate compressor (Motor rating: 1.6 MWe)

Based on the above data, the following considerations can be drawn:

- The power generation of the gas turbine slightly increases for the two near-zero emission plants, mainly due to the enhanced heat recovery in the MHE. On the other hand, the gross power production including the recycle gas compressor consumption is reduced, due to the increased power demand of the compressor.
- As no purification is foreseen, the low CO_2 purity case shows the lower auxiliary power consumptions and consequently the best efficiency (+0.2 percentage points).

- The increased size and consequently consumptions of the CPU compressors and the additional permeate compressor are the main reason of the higher auxiliary power consumptions and consequently of the lowest efficiency (-0.4 percentage points) of the membrane case.
- The impact on utility and ASU power demand is negligible. It is interesting to point out that, as the excess oxygen required for the combustion is low (around 3%), the saving in the oxygen demand (and consequently ASU consumptions and size) is negligible, compared to the coal-fired oxy-combustion boiler, where the higher oxygen excess (around 20-30%) leads to high oxygen content in the permeate stream and finally to an oxygen demand reduction of around 3-4%.

6.2.5. *Sensitivity to the cooling tower approach*

As defined in chapter B of the report, the cooling water system included in the base cases of the study is based on a natural draft cooling tower, designed with 7°C approach temperature and a resulting cooling water supply temperature of 15°C.

This sensitivity case analyses the impact on plant performance of selecting the extremely aggressive approach for the cooling tower of 4°C. As a consequence, in order to limit the cost impact of this design change, the cooling system is modified from the natural draft to the mechanical draft type. Amec Foster Wheeler would like to acknowledge SPIG for the support in the development of this sensitivity case.

Cooling water system design

Cooling water approach to wet bulb temperature:	4°C
Supply temperature	
- normal:	12 °C
- maximum:	33 °C
Maximum temperature difference at users:	11°C
Mechanical design temperature:	50°C
Operating pressure at user:	3.0 bar
Mechanical design pressure:	6.0 bar
Maximum allowable ΔP for users:	1.0 bar

Mechanical draft cooling tower design features

The cooling tower system will be based on three towers, each including 12 cells (18m x 18m) with relevant fan (250 kW motor rating). A single basis is foreseen for each tower.

The cooling tower structure will be made in FRP. The tower frame work shall consist of structural shapes of fibreglass composite, stiffened with diagonal braces to transfer wind, earthquake and other live loads to the basin.

Each cell has a fiberglass reinforced polyester fan stack with bell-shaped inlet and conical, diverging exit for reduced air pressure drop losses and pressure recovery for high fan efficiency. Fan stack sections are sized to permit the optimal fan blade tip clearance, maximizing fan performance. Each stack has a removable access panel opening for maintenance on the fans and gears. Distribution system, inside the cooling tower, is of the gravity type. It will be made of a main channel with connection to secondary pipes. Pipes are fitted with spray nozzles spaced so to guarantee uniform water feed to all cross section of the cell.

Fan is multi-blade type and has the main function of ensuring the Cooling Tower design air volume necessary for cooling. The fan is placed on the high section of the cell and it is directly coupled with the gear box.

The design concentrations cycles (CC) is 4.0, as for the base case, thus implying similar blowdown requirements. As evaporation losses depend on the ambient conditions, the difference is negligible. Therefore, the resulting cooling tower make up is similar to the base case.

Table 11 shows the impact on plant performance and the key design features when moving from a cooling water system based on natural draft with 7°C approach to a mechanical draft system with a lower approach (4°C).

Table 11. Case 2 – Sensitivity to cooling tower approach

			BASE CASE	SENSITIVITY CASE
SENSITIVITY				
Cooling tower approach		**°C**	**7**	**4**
Cooling tower type		**-**	**natural draft**	**mechanical draft**
DESIGN FEATURES				
Cooling water temperature	Supply	°C	15	12
	Return	°C	26	23
Cooling tower features			see section 2.5	see above descrip.
Cooling water pumps differential pressure (to take into account the different water inlet height in cooling tower height)		bar	4	3.5
PERFORMANCES COMPARISON				
Natural Gas flow rate (A.R.)		t/h	118.9	118.9
Natural Gas LHV		kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)		**MWth**	**1536**	**1536**
Direct-fired sCO2 turbine power output		MWe	1263.9	1265.9
Recycle gas compression train		MWe	-207.9	-197.9
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)		**MWe**	**1056.0**	**1068.0**
Power cycle (including NG compressor)		MWe	13.9	13.9
Air separation unit + Oxygen compressor		MWe	170.9	170.7
CO2 purification and compression unit		MWe	12.4	12.2
Utility & Offsite Units		MWe	10.4	16.8 (1)
ELECTRIC POWER CONSUMPTION		**MWe**	**207.5**	**213.6**
NET ELECTRIC POWER OUTPUT		**MWe**	**848.4**	**854.4**
(Step Up transformer efficiency = 0.997%) (B)		MWe	845.9	851.9
Gross electrical efficiency (C/A x 100) (based on LHV)		**%**	**68.7%**	**69.5%**
Net electrical efficiency (B/A x 100) (based on LHV)		**%**	**55.1%**	**55.4%**
Fuel Consumption per net power production		**MWth/MWe**	**1.82**	**1.80**
CO_2 emission per net power production		**kg/MWh**	**37.0**	**36.6**

(1) Relevant to cooling tower fans: 7.5 MW

Based on the above data, the following considerations can be drawn:

- The selection of an aggressive approach for the cooling tower design leads to an increased gross power output (+0.8 percentage points), mainly due to the lower recycle compressor power demand resulting from the colder conditions downstream the intercoolers, and a small increase of the gas turbine production resulting from the optimisation of the MHE with these conditions.
- Only minor power savings are estimated in the ASU and CPU compressors, as in the NET Power configuration the MAC is not intercooled while the CPU only includes the final CO_2 compressors.
- For the mechanical draft cooling towers, the total auxiliary power consumption increases because the additional power demand of the cooling tower fan more than offsets the reduction of the power demand of the CPU, the ASU and the cooling water pumps, due to their lower discharge pressure.
- Based on the above, the selection of a mechanical draft cooling tower design with 4°c approach leads to a net electrical efficiency increase of 0.3 percentage point.

7. Environmental impact

The oxy-combustion direct-fired sCO_2 turbine NET Power plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 12 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power and CO_2 cycle and in the CO_2 purification unit.

Table 12. Case 2 – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	36,290
Flow, Nm^3/h	19,500
Composition	(%mol)
Ar	3.67
N_2	9.96
O_2	4.99
CO_2	81.38
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from the exhaust gas condenser/ compressor intercoolers and CO_2 purification unit) are treated to recover water, so the main liquid effluent is cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 305 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 2 - NET POWER cycle	

REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
DATE	Sep-14	Nov-14	Feb-15	
ISSUED BY	NF	NF	NF	
CHECKED BY	LM	LM	LM	
APPROVED BY	LM	LM	LM	

EQUIPMENT LIST

Unit 3000 - Power and CO_2 Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	TURBINE PACKAGE							
PK- 3101-1/2	**Turbine and Generator Package**							***2 x 50% turbine package***
	Each including:							
T- 3101	Direct-fired sCO2 Turbine		635 MWe Pin: 300 bar; Pout 34 bar					*Including:* *Lube oil system* *Cooling system* *Hydraulic control system* *Seals system* *Drainage system*
G- 3101	Turbine generator		*650 MVA*					*Including relevant auxiliaries*
F- 3101	Combustor		*770 MWt*					*One per train, two in total*
K- 3101	**NG compressor**		Flowrate: 74,000 Nm3/h Pin: 70 bar; Pout: 315 bar Compression ratio: 4.5	*4,750 kWe*				*One per train, two in total*
	RECYCLE GAS COMPRESSION PACKAGE							
PK- 3102-1/2	**Recycle gas compression package**							
	- Recycle gas compressor #1		17 MWe					
	- Recycle gas compressor #2		15 MWe					
	- Recycle gas compressor #3		14 MWe					
	- Recycle gas compressor #4		10 MWe					
	- Pumping stage #1		9 MWe					
	- Pumping stage #2		22 MWe					
	- Oxidant recycle compressor		17 MWe					
	- CW intercoolers							
PK- 3103-1/2	**Main Heat Exchanger**							*Compact multi-channel plate-fin type*

CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
LOCATION: The Netherlands	DATE	Sep-14	Nov-14		
PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
CASE.: 2 - NET POWER cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 4000 - CO_2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	PACKAGES							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO_2 liquid separator							
	- CO_2 distillation column							
	- CO_2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MW*				
	- CO_2 compressor 2	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		2000 MWe					*1x100%*
	- Overhead recycle compressors							
	- Intercoolers							
	Oxygen pre-heater							
	Cooling water intercoolers							

	CLIENT:	IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
FOSTER WHEELER	LOCATION:	The Netherlands	DATE	Sep-14	Nov-14	Feb-15	
	PROJ. NAME:	Oxy-turbine power plants	ISSUED BY	NF	NF	NF	
	CONTRACT N.:	1-BD-0764 A	CHECKED BY	LM	LM	LM	
	CASE.:	2 - NET POWER cycle	APPROVED BY	LM	LM	LM	

EQUIPMENT LIST
Unit 5000 - Air Separation Unit (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5400 t/d Oxygen purity: 99.5%mol Oxygen pressure: 100-120 bar					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 39 MWe				*Not intercooled*
	- Booster air compressor	Centrifugal		10.5 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 2 - NET POWER cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Sep-14	Nov-14	Feb-15	
		ISSUED BY	NF	NF	NF	
		CHECKED BY	LM	LM	LM	
		APPROVED BY	LM	LM	LM	

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower** including: Cooling water basin	Natural draft	910 MWth Diameter: 110 m, Height: 180 m, Water inlet: 17 m				concrete	
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/…/F	**Cooling Water Pumps**	Centrifugal	12500 m3/h x 40 m	1650 kW				*Six in operation, one spare*
P- 6003 A/B	**Cooling tower make up pumps**	Centrifugal	1350 m3/h x 30 m	160 kW				
	PACKAGES							
	Cooling Water Filtration Package Cooling Water Sidestream Filters		7110 m3/h					
	Sodium Hypochlorite Dosing Package Sodium Hypochlorite storage tank Sodium Hypochlorite dosage pumps							
	Antiscalant Package Dispersant storage tank Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
	RAW WATER SYSTEM							
PK- 6002	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Potable water pumps	centrifugal						*One operating, one spare*

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Sep-14	Nov-14	Feb-15	
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF	NF	
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM	LM	
	CASE.: 2 - NET POWER cycle	APPROVED BY	LM	LM	LM	

EQUIPMENT LIST

Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	FIRE FIGHTING SYSTEM							
T- 6003	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							

ATTACHMENT A.1: NET Power commentary

1. Introduction to NET Power commentary

The authors of the report (IEAGHG and Amec Foster Wheeler Italiana) have asked to the various oxy-combustion turbine technology developers if they wished to prepare a summary of their state-of-the-art techno-economic assessments. However, NET Power has preferred to provide a commentary to those sections of the report, done by the authors and relevant to their technology.

This commentary is included in the overleaf pages, as received from NET Power. Though IEAGHG and Amec Foster Wheeler Italiana have agreed to include this commentary in the report, a brief introduction to it has been deemed anyhow necessary, because many of the assumptions and results mentioned in the commentary deviate from the main study design bases, while some other considerations are an incorrect interpretation of the study results.

The main subject of the NET Power's economic commentary is the overestimated cost of their plant, as prepared by Amec Foster Wheeler, mainly because this is not representative of a NOAK plant, but rather of an "early commercial" plant, as the potential future technology improvements have not been taken into account.

However, it is Amec Foster Wheeler Italiana's opinion that the estimates shown in the report are NOAK estimates, based on current state of the technology. In fact, even if past experience shows that costs of technologies tend to increase as they move from initial concepts towards pilot plants, commercial plant designs and then actual completed FOAK plants, in order not to penalize the oxy-fuel technology it has been decided of not considering any additional cost related to the development status of the technology.

It is also Amec Foster Wheeler Italiana's opinion that potential future technology improvements are likely to come for all the oxy-fuel technologies and not only for the NET Power cycle. This is also valid for the NGCC, especially because the combined cycle of the study does not even exploit some of the latest technology updates (e.g. H class turbines, ceramic-based high temperature components), though their development status is well ahead that of the oxy-fuel technology. If the study had to be based on potentialities of the technologies in the coming years, then the benefit would have to be considered for all the assessed cycles and not only for the NET Power technology, as mentioned in their commentary.

Therefore, Amec Foster Wheeler Italiana thinks that the additional saving claimed by NET Power only for their technology (while also claiming an increased cost of the NGCC) would be in contrast with the general methodological approach of the study.

Furthermore, other observations included in the NET Power's commentary are based on a misinterpretation of the study results, probably mainly because their review was relevant to a limited portion of the report.

As an example, cost saving for multiple trains is not included in the estimate of each technology (not only for the 120 bar ASU of NET Power scheme), regardless the status of its development. In fact, this cost saving could be taken into account only if supported by a vendor quote tailored for the case-specific design bases (e.g. pressure, capacity), which is not applicable to the one mentioned by NET Power. Furthermore, it is noted that the ASU cost estimate in the report includes some significant cost benefits, originating from the selection of the largest commercially available train capacity, although the ASU configuration in the NET Power cycle requires some further development with respect to a more standardized configuration of the other cycles, because there is a high level of thermal integration with the other units.

In summary, it is the author's opinion that the study report is based on a good balancing between current and new technology, avoiding any penalties relevant to the status of development and demonstration of the new technologies, therefore based on the technology that are currently available. In general, considerations on future market scenarios and possible developments are avoided, as these would have involved too many speculations and would have been affected by biased considerations and interests of the technology developer. Also in this respect, the authors deem to have based the report on current technologies, rather than on expected data, with the main objective of providing fair and impartial techno-economic assessments of the various oxy-turbine technologies.

8 Rivers Capital, LLC
NET Power, LLC
406 Blackwell Street
4th Floor, Crowe Building
Durham NC 27701

Commentary on Chapter D.2 relating to the NET Power System in the IEAGHG Report on Oxy-Turbine Power Plants

1. Introduction

NET Power and its technology provider, 8 Rivers Capital, are pleased to have been given the opportunity by the IEAGHG and Amec Foster Wheeler Italiana (AmecFWI) to provide this commentary on the evaluation of the NET Power system contained in the draft IEAGHG Report on Oxy-Turbine Power Plants. This commentary has been prepared after reviewing a draft of the Report's Chapter D.2 relating to the NET Power system.

The NET Power system will achieve high-efficiency, cost-effective power generation from natural gas with near-zero emissions, including near-100% capture of all CO_2 produced by combustion. The NET Power system generates electricity at a cost comparable to natural gas combined cycles, and the captured CO_2 is delivered at high quality and pressure. The system is the product of significant research and validation by 8 Rivers, together with NET Power and its investors and partners. A 50MWth small-scale plant to be commissioned starting 2016 will demonstrate the NET Power system and its ability to produce electricity while inherently capturing CO_2. NET Power and 8 Rivers have, together with EPC firm CB&I, completed detailed design, engineering and costing for this demonstration plant (to be located in Texas) and a pre-FEED for a commercial-scale plant based on a 500MWth supercritical CO_2 (sCO_2) turbine (operating in ISO ambient conditions). The results of this work have given NET Power and 8 Rivers a detailed understanding of the efficiency and costs of the system at the commercial scale and certainty of the expected benefits over incumbent and other proposed technologies. NET Power is currently in discussions with customers about the planning and development of the first generation of NET Power plants that will come online between 2019 and 2025.

AmecFWI has undertaken its evaluation for the Report based on information that is publicly available, and thus it is not reflective of the performance that can be expected from actual commercial NET Power plants. The work undertaken by 8 Rivers over the last five years has generated significant proprietary and confidential information and trade secret learnings (in terms of system design, operation and optimisation) that underpin the cost and performance estimates prepared for NET Power by 8 Rivers and others, and would be incorporated in commercial NET Power plants. This confidential information was not considered by AmecFWI in its evaluation. The following commentary explains in general terms the nature of this know-how and how it enables the NET Power system to achieve the performance that has been disclosed publically.

2. Plant Performance

As noted above, AmecFWI's modelling is not representative of the NET Power system's actual commercial plant performance. Nevertheless, AmecFWI's modelling demonstrates a proficient level of understanding of the core principles of the NET Power process.

The net overall LHV efficiency of 55.4% modelled by AmecFWI (in the mechanical draft cooling case) used only the information that is publicly available regarding the NET Power process. This efficiency is in general agreement with the efficiency modelled by 8 Rivers when using the same publicly available information.

By incorporating proprietary information into the modelling and analysis, the NET Power system achieves a net overall LHV efficiency of 58.8% based on the Report's conditions and assumptions, such as natural gas composition and purified CO_2 specification, (except to the extent noted in the sections below). This efficiency indicates the performance that can be expected from the first and early commercial NET Power plants subject to the constraints of the Report's conditions and assumptions. This system incorporating confidential information would also capture and purify a larger percentage of the produced CO_2 than predicted by AmecFWI. The performance breakdown for 8 Rivers' model of this system, and a comparison with the AmecFWI model results, is set out in Table 1 below.

Table 1. Comparison of Overall Performances

		8 Rivers Model (confidential information)	**AmecFWI Model Mechanical Draft (public information)** (see Note 1 below)
Natural Gas flow rate (A.R.)	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46493	46502
Natural Gas HHV	kJ/kg	51464	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1700**	**1701**
sCO_2 turbine power output	MWe	Confidential vendor data (see Note 2 below)	1265.9
CO_2 compression and pumping (mechanical)	MWe		-197.9
GROSS ELECTRIC POWER OUTPUT (C) (@ gen terminals)	**MWe**	**1298.3**	**1068.0**
CO_2 compression and pumping (electric)	MWe	201.8	-
Natural gas compression	MWe	13.7	13.9
Air separation unit	MWe	162.2	170.7
CO_2 purification and compression unit	MWe	2.4 (see Note 3 below)	12.2
Utility & Offsite Units	MWe	13.1	16.8
ELECTRIC POWER CONSUMPTION	**MWe**	**393.2**	**213.6**
NET ELECTRIC POWER OUTPUT	**MWe**	**905.1**	**854.4**
(Step Up transformer efficiency = 0.997%) (B)	MWe	902.4	851.9
Gross electrical efficiency (C/Ax100) (LHV)	**%**	**84.6%**	**69.5%**
Net electrical efficiency (B/Ax100) (LHV)	**%**	**58.8%**	**55.4%**
Gross electrical efficiency (C/A'x100) (HHV)	**%**	**76.4%**	**62.8%**
Net electrical efficiency (B/A'x100) (HHV)	**%**	**53.1%**	**50.1%**
Equivalent CO_2 flow in fuel	kmol/h	7158	7159
Captured and purified CO_2	kmol/h	6730	6448
CO_2 purification efficiency	**%**	**94.0%**	**90.1%**
Fuel Consumption per net power production	**MWth/MWe**	**1.70**	**1.80**
CO_2 emission per net power production	**kg/MWh**	**20.9**	**36.6**

Note 1. AmecFWI's mechanical draft cooling sensitivity case is used for all comparisons because mechanical draft cooling towers are specified for the NET Power system. Compared with natural draft cooling, mechanical draft cooling increases the plant's overall net electrical output and efficiency and has a lower capital cost. Further discussion of the advantages of mechanical draft cooling is in section 2.2 below.

Note 2. The precise gross turbine efficiency of Toshiba's turbine is not publicly available information. In NET Power's model, only the main CO_2 compressor is assumed to be mechanically coupled to the turbine, so the CO_2 compression and pumping parasitic load is therefore split between the "CO_2 compression and pumping (mechanical)" and the "CO_2 compression and pumping (electric)" line items. As the main CO_2 compressor is not an integrated part of the sCO_2 turbine, there would be the ability to change the shaft load, including by mechanically coupling additional or alternative equipment.

Note 3. 8 Rivers has modelled its own integrated CO_2 purification system, which is described further in section 2.3 below, and some of its parasitic load is included in the "CO_2 compression and pumping (electric)" line item.

NET Power, 8 Rivers and their partners have gained an extensive understanding of the NET Power process, which has led to the development of proprietary methods to take advantage of the unique thermodynamics of the cycle and the performance of commercially available equipment suitable for CO_2 service. These methods generally relate to:

- Heat integration and recovery.
- Cooling.
- Optimisation of main process streams.
- Equipment specification, selection and operation.

Further explanation of the effect of these methods on the performance of the system is contained in the following sections 2.1 to 2.4 below. 8 Rivers' modelling techniques and results have been confirmed both by independent third parties and its partners who have had access to these confidential practices, methods and designs.

2.1. Heat Integration and Recovery

Publicly disclosed details of the NET Power system (such as the underlying Allam Cycle described in U.S. Patent 8,596,075, filed in 2010) consider the recovery of waste heat from process streams other than the turbine exhaust as a means to increase overall cycle performance. Recovery of waste heat from the ASU, as modelled in the Report, is just one embodiment of this process. Since 2010 there has been significant development and optimisation of heat recuperation within the cycle, including the implementation and refinement of an additional method of generating and recovering heat internal to the process itself.

The methods by which heat is recuperated by the cycle are also the product of extensive research and development by 8 Rivers. Additional percentage points of efficiency are achieved by taking into account the characteristics of CO_2 as a heat exchange fluid, in particular its changing specific heat values across the temperature and pressure ranges encountered in the system. NET Power and 8 Rivers have confirmed with equipment manufacturers that the performance required of the main heat exchanger train will be achievable with existing technology and manufacturing processes. Without these proprietary methods of heat recuperation, system performance (LHV efficiency) may drop by around two to three percentage points.

2.2. Cooling

Cooling throughout the plant is an area that has been fine-tuned by 8 Rivers. Because the NET Power process is a semi-closed, recuperative sCO_2 cycle, it departs from the typical considerations and limitations that apply in relation to cooling traditional steam-based systems. Optimising the cooling in the NET Power process is done with the aim of reducing the parasitic load of compression and pumping of the recycled CO_2 by lowering the temperature of the recycled CO_2 working fluid as much as is practical. Due to the unique properties of CO_2 at the pressures and temperatures in the cycle, temperature has a significant effect on the density and compressibility of the recycle stream. Cooling the CO_2 as much as is practical has the effect of (1) lowering the compressibility of the recycle CO_2, leading to more efficient pressurization, (2) reducing the pressure at which the CO_2 becomes dense enough for the pump suction, thereby reducing the compression load in favour of more efficient pumping, and (3) reducing the required size of the compressors and pumps due to the smaller volumetric flow rates.

Each of the above consequences of low cooling temperatures contributes to reducing parasitic loads and lowering the costs of re-pressurizing recycled CO_2. Since compression and pumping of the recycled CO_2

represents a significant portion of the parasitic load of the overall NET Power plant, reductions in cooling temperature will directly and significantly improve overall system efficiency and cost. By contrast, the temperature of the cooling water in a steam cycle has a limited to negligible impact on how parasitic loads are managed, since cooling water temperatures primarily impact cycle performance via the steam condensing pressures that they generate. The efficiency of steam turbine systems (namely turbine gross power output) is indirectly increased through lowering the LP turbine steam condensing pressure to a practical limit, which has a *de minimis* impact on overall performance.

There are two main aspects of cooling optimisation in the NET Power system, namely (1) achieving the lowest practicable cooling water temperature from the cooling water system and (2) effectively using that low temperature cooling water to cool the CO_2 working fluid.

2.2.1. Achieving Low Cooling Water Temperature

8 Rivers specifies the use of mechanical draft cooling towers with the NET Power cycle, because mechanical draft cooling is often a more cost effective cooling solution versus natural draft cooling given the specific characteristics of the Allam Cycle.

While both natural draft and mechanical draft systems can be operated with an approach temperature of approximately 7°C (as used by the natural draft cooling tower modelled for the based case in the Report), mechanical draft technology is capable of more effectively reducing the approach to the wet bulb temperature to a lower value than that of a natural draft system. Natural draft systems are inherently limited in the approaches they can achieve given their methods of construction and reliance on the buoyancy of upward currents through natural convection. Vendor data obtained by NET Power and 8 Rivers shows that an approach to the wet bulb temperature of approximately 4°C being achievable using today's mechanical draft cooling systems. Typically, the extra cost needed to achieve these closer temperatures is not justified, especially when the additional parasitic electrical load for the draft fans is considered. However, because of the direct link between cooling temperatures and compression parasitic loads in the NET Power system, the lower cooling water temperature results in an increase in net plant output as compared to using a natural draft cooling tower using a 7°C approach. This is confirmed by the mechanical draft cooling sensitivity case in the Report, which shows that AmecFWI were able to achieve a 0.7% increase in net plant output (6MWe) by using mechanical draft (4°C approach) instead of natural draft (7°C approach).

Additionally, the equipment and erection costs of modular mechanical draft cooling towers are considerably less than the cost of the bulk materials, equipment and construction for large-scale natural draft towers. According to a reputable international EPC firm consulted by NET Power, mechanical draft towers may cost about 80% less than natural draft at the required duty. In the cost estimation section of the Report, AmecFWI similarly estimate that their natural draft units are about three times the cost of the mechanical draft units.

These factors dictate that in most situations it is more advantageous for the NET Power system to use mechanical draft cooling.

2.2.2. Efficiently Cooling the Recycled CO_2

Given the high volumetric flow rates that occur during re-pressurization of the recycle CO_2, compact heat exchangers are specified for the heat transfer units for the CO_2 compressors and pumps in the NET Power system. An additional benefit of these exchangers is that they can more readily supply tighter approach temperatures than the tube and shell type technology often used in steam based applications.

Furthermore, the difference in specific heat between (x) the gaseous CO_2 that is re-pressurized at the cold end of the NET Power process and (y) the liquid water coolant supplied by the cooling water system is significantly larger than the difference in specifics heats of condensing steam and cooling water seen in conventional steam cycles. This larger difference results in a correspondingly larger log mean temperature difference (LMTD) for heat exchangers in the NET Power system for a given approach temperature. NET Power is therefore able to pursue more aggressive approach temperatures than those used in traditional steam system cooling designs while balancing practical considerations for heat exchanger heat transfer coefficients and cumulative duties.

2.3. Optimisation of Main Process Streams

As compression work is the second largest parasitic load in the NET Power system (after the air separation process), considerable effort has been put into optimising the flow rates and compositions for the main process streams so as to cost-effectively minimise this parasitic load.

An example of this is the specification for the oxygen supply technology and delivery pressure, where the trade-off of ASU power consumption and cost versus oxygen compression load were studied.

A further example is the combustor oxidant stream, where extensive testing of the combustor over the last two years has shown that the high recycle rate (leading to accumulation) and combustion conditions above the auto-ignition point for the fuel mixture reduce the required excess oxygen to a level below that modelled by AmecFWI. Reducing this level also significantly reduces the overall oxygen content in the system's CO_2 inventory and CO_2 export stream due to the high recycle rate and semi-closed nature of the process. These have positive implications for the downstream compression and purification processes.

In general, the purity of the CO_2 export stream from the NET Power plant is at least 95 mol%, and the possibility of additional purification is an integral feature of the NET Power system. AmecFWI sets the turbine outlet pressure at 34 bar in order to match the CPU inlet pressure required by the separate cryogenic CPU modelled by AmecFWI in the Report. This disproportionately sacrifices turbine output compared to using a lower turbine outlet pressure and supplementary CO_2 compression. Further, NET Power would utilise a CO_2 purification system designed by 8 Rivers that, because it is closely integrated with the core cycle, results in moderate efficiency improvements and greater production of purified CO_2. As shown in the table above, this system also reduces the parasitic load of the CO_2 purification process by a meaningful amount compared to the CPU modelled in the Report, at the same time increasing the amount of purified CO_2 output. It also allows the turbine to have a pressure ratio of about 10 (compared to a pressure ratio of about 9 as modelled in the Report) while still staying within the region where CO_2 has favourable thermodynamic properties and thus leading to greater power output.

2.4. Equipment Specification, Selection and Operation

In undertaking the design, engineering and specification of a commercial plant (specifically as part of a Pre-FEED Study commissioned by NET Power), NET Power and 8 Rivers have engaged in detailed discussions with vendors in order to confirm all equipment specifications, cost and performance. All of this information contributes to high degree of confidence in NET Power's design and modelling results.

In particular, when modelling the NET Power process, 8 Rivers uses data about the direct-fired sCO_2 turbine that have been supplied by the manufacturer, Toshiba. Toshiba is a major supplier of turbomachinery worldwide and has demonstrated expertise in the research, design, testing, manufacturing, sale, supply and maintenance of advanced turbines. While developing the NET Power turbine, Toshiba has provided its own proprietary data for the turbine in order for 8 Rivers to comprehensively and accurately model the turbine's performance in the system. This information leads to a greater gross turbine output than that modelled by AmecFWI. The difference in output may be accounted for by the Report's use of a turbine model developed for conventional, air-fed gas turbines, which specify cooling flows (rates, temperature and configuration) and seal configurations that are not the same as Toshiba's (confidential) design.

In addition, NET Power and 8 Rivers have methodically investigated the equipment offerings of vendors of compressors, pumps, ASUs and heat exchangers in order to identify the commercially available technologies that are suitable for use in the NET Power system. In each case, several cost-effective options have been identified. This review process has included producing bid packages for key plant equipment, confirming performance characteristics and obtaining budgetary estimates. As a result, 8 Rivers is able to model the system using data that are based on or corroborated by vendors (such as turbo-machinery performance maps) that have been provided specifically for the purposes of the NET Power system, and which produce more accurate results than can be obtained from generic performance tables. NET Power and 8 Rivers are also working with vendors on an ongoing basis to improve the integration and efficiency of equipment throughout the cycle, which is expected to yield overall plant performance improvements for nth-of-a-kind plants.

3. Conclusion

8 Rivers's proprietary techniques for optimising system performance will allow NET Power plants to achieve higher efficiency and better CO_2 purification performance than that modeled by AmecFWI in the Report (58.8% LHV compared to 55.4% LHV). 8 Rivers is already developing systems and methods that will guarantee at least 60% LHV efficiency for first generation nth-of-a-kind NET Power plants with CO_2 capture, ensuring that they outperform other proposed oxy-combustion power systems and are competitive with natural gas combined cycle whether with or without CO_2 capture.

May 2015

NET Power, LLC, 8 Rivers Capital, LLC, 406 Blackwell Street, Durham, North Carolina, 27701, USA

Email: info@netpower.com

8 Rivers Capital, LLC
NET Power, LLC
406 Blackwell Street
4th Floor, Crowe Building
Durham NC 27701

Commentary on Cost Estimate Summaries relating to the NET Power system in the IEAGHG Report on Oxy-Turbine Power Plants

1. Introduction

NET Power and its technology developer, 8 Rivers Capital, are pleased to have been given the opportunity by the IEAGHG and Amec Foster Wheeler Italiana (AmecFWI) to provide this commentary on the evaluation of the NET Power system contained in the draft IEAGHG Report on Oxy-Turbine Power Plants. This commentary has been prepared after reviewing a draft of the Report's cost estimate summaries for the NET Power system.

The NET Power system will achieve high-efficiency, cost-effective power generation from natural gas with near-zero emissions, including near-100% capture of all CO_2 produced by combustion. The NET Power system generates electricity at a cost comparable to non-capture natural gas combined cycles, while also capturing the CO_2, which is delivered at high quality and pressure. The system is the product of significant research and validation by 8 Rivers, together with EPC firm CB&I, power producer Exelon and equipment OEM Toshiba. A 50MWth small-scale plant (to be commissioned starting 2016) will demonstrate the NET Power system and its ability to produce electricity while inherently capturing CO_2.

AmecFWI has undertaken its evaluation for NET Power plants based on information that is publicly available, and thus AmecFWI's results do not completely convey the performance and cost benefits of commercial NET Power plants. Cost and performance estimates prepared for NET Power are based on considerable work undertaken by its partners (8 Rivers, CB&I, Exelon and Toshiba) that has generated proprietary and confidential information (in terms of system design, operation and optimisation). This information was not considered by AmecFWI in its evaluation.

However, even if these estimates were based on full information, there are several inconsistencies and skewed comparisons made in the Report that must be addressed. First, costs for new technologies in this Report must be considered estimates of "early commercial" plants, rather than "nth-of-a-kind", due to the nature of the Report's specifications for cost estimation. This distinction is critical when comparing to estimates for natural gas combined cycle (NGCC) technology, which are true "nth-of-a-kind" estimates benefitting from decades of process improvement, technology advancements and cost maturity, since those benefits are not considered for new technologies studied in the Report. The NET Power system is highly competitive when an equivalent "nth-of-a-kind" estimate is compared with those of mature (NGCC) plants. Second, there are several adjustments to AmecFWI's methodology that NET Power believes improve its consistency and conformity with the Report's specifications. These adjustments bring the capital cost figure determined by this Report to within 10% of NET Power's own estimates for similar "early commercial" plants. Third, costs for mature NGCC plants must be more carefully investigated, since this widely deployed technology is subject to competitive market forces and pricing strategies that are not represented in estimates for new technologies considered in the Report.

Each of these aspects is discussed in more detail below.

2. "Early Commercial" vs. "Nth-of-a-kind" Estimates

The IEAGHG's specification for the Report required cost estimates to be:

> *"… for 'nth plants' based on current knowledge of the technology, i.e. they are commercial plants built after the initial technology demonstration plants. Additional costs normally associated with 1st-of-a-kind commercial plants shall be excluded."*

In other words, cost estimates for new technologies, such as NET Power, are considered by the specification to be an "nth" cost if current knowledge and technology status are utilized and contingencies associated with "1st-of-a-kind" facilities are removed. For new technologies, however, NET Power considers this specification to be more representative of "early commercial" units, as opposed to an "nth-of-a-kind" estimates. A true "nth-of-a-kind" estimate would take into account technology developments, process improvements, and reduced pricing of major equipment and plants seen with increasing commercial deployment – consistent with the history of combustion turbines and combined cycle power generation (NGCC). It is therefore incorrect to describe these "early commercial" estimates for new technologies as "nth-of-a-kind", as is used throughout the Report, particularly when comparing to mature NGCC costs. As discussed in NET Power's commentary to Chapter D.2, the performance modeling of the NET Power system for the Report's purposes is also consistent with this "early commercial" specification, only considering what is possible with existing, commercially available equipment and materials and not any enhancements that are expected with further R&D and product improvements.

As has been mentioned previously, an "nth-of-a-kind" estimate typically accounts for performance increases and cost reductions resulting from technology developments and process improvements that will occur with increasing commercial deployment. The cost estimate for NGCC used in the Report constitutes a true "nth-of-a-kind" estimate, as the technology benefits from decades of continuous technology improvement, process refinement and cost maturity. These "nth-of-a-kind" factors, however, have not been considered in AmecFWI's estimates for NET Power or other new technologies, nor are they required or permitted to be considered by the study specification. Estimates for NGCC in the Report are therefore not directly comparable to "early commercial" estimates for NET Power.

A more direct comparison should instead be made with NET Power's median "nth-of-a-kind" estimate for a US plant ($1,047/kW; €830/kW) which, although improvements to current technology and process design are not taken into account, does account for competitive pricing (as opposed to single vendor quotes), large volume orders and learning curve reductions from "early commercial" estimates. One of the most recent publically available resources for non-capture NGCC costs[1] for a US location (and therefore more directly comparable to NET Power's estimate for a US location) shows an average cost of $1,061/kW (€842/kW) for "nth-of-a-kind" plants located in the deregulated PJM ISO territory on the US East Coast, which indicates that NET Power's estimate is highly competitive, even in the "early commercial" phase.

3. Analysis of Estimate for NET Power in the Report

Work undertaken by NET Power, as leader of the commercialisation programme, and 8 Rivers, as technology developer, has given NET Power and 8 Rivers a detailed understanding of the costs of the NET Power system at the commercial scale. NET Power and 8 Rivers have, together with EPC firm CB&I, completed design, engineering and costing for a commercial-scale plant based on a 500MWth supercritical CO_2 (sCO_2) turbine operating in ISO ambient conditions for a US location. This work has included:

- Preparing optimized material, energy, and heat balances for the plant.
- Preparing specifications for engineered equipment performance.
- Obtaining budgetary estimates and specified performance curves from equipment vendors.
- Defining required host site services and utilities.
- Developing a plant layout and required equipment supports and piping runs.

[1] Brattle for PJM Interconnection, "Cost of New Entry Estimates for Combustion Turbine and Combined Cycle Plants in PJM", 2014

- Developing plant operational requirements and associated “bottom-up” costs.
- Developing a project procurement, construction and commissioning schedule.
- Preparing a capex and opex cost estimate for the plant, using the above information.

Using this work, NET Power and its partners have prepared internal “bottom-up” capital and operating cost estimates for the “1st-of-a-kind”, “early commercial”, and “nth-of-a-kind” NET Power plants, as NET Power recognises that cost considerations and assumptions differ in each of these stages of a technologies development status.

Considering the study specification, and the data generated from prior investigation of the commercial-scale NET Power plant, NET Power offers the following observations concerning the accuracy of the “early commercial” costs estimated by AmecFWI for the NET Power plant:

- AmecFWI’s estimate for the direct material cost of “Unit 3000 Power & CO_2 Cycle”, which encompasses the main plant and equipment comprising the NET Power system (turbine, generator, main heat exchanger, CO_2 compression train, water separator, CO_2 and oxygen pumps, etc.), is about 10% higher (~€25M) than NET Power’s “early commercial” estimate. Since construction, EPC and contingency costs are determined as a function of direct material costs, this results in a total overestimation by AmecFWI of ~€49M for Unit 3000.

- The cost estimate for “Unit 5000 ASU” is higher than indicated by market data collected by NET Power and 8 Rivers. Furthermore, ASU costs calculated by AmecFWI are derived from a vendor quote that specifies significant savings on a second ASU train. These savings have been ignored by AmecFWI on the basis that production of 120 bar O_2 is not commonly practiced today. This is inconsistent with the study specification requiring that “additional costs normally associated with 1st-of-a-kind commercial plants shall be excluded”. NET Power estimates these savings to be ~€47M when the “1st-of-a-kind” costs are excluded and a consistent estimating methodology is employed.

- AmecFWI has not used a consistent estimating methodology to calculate maintenance costs (part of fixed O&M) for the NET Power system and the NGCC baseline. Costs for NET Power have been calculated as 2.5% of Unit 3000 direct material costs and 1.5% of all other Unit direct material costs. This is in contrast to maintenance costs for NGCC, which have been calculated as 1.5% of all Unit direct material costs. This is again inconsistent with the specification that “additional costs normally associated with 1st-of-a-kind commercial plants shall be excluded”, since there is no fundamental technology difference included in Unit 3000 (turbines, pumps, compressors, heat exchangers, etc.) that would account for higher intensity of maintenance costs as a proportion of direct material costs. Since no “bottom-up:” analysis has been done by AmecFWI to indicate this higher intensity is justified, it can be concluded that this is meant to address uncertainty with equipment on a “1st-of-a-kind” commercial facility. This should instead be brought in line with the 1.5% adopted for the other systems, resulting in ~€7.0M/year in fixed O&M cost savings (when combined with the capital cost reductions discussed previously).

- A similarly inconsistent approach has been applied to the operational staff required by a NET Power plant, which was estimated by AmecFWI to be about double that of an NGCC plant. A NET Power plant has fewer and smaller turbines (single turbine per train, as opposed to the two gas turbines and three steam turbines per train in the NGCC configuration in the Report), no complex HRSG system and a smaller footprint for the power train. Even large ASUs are today operated nearly completely remotely with few on-site staff required. There is therefore no expected fundamental difference in the operations or maintenance requirements between an “early commercial” NET Power plant and an NGCC plant. Assuming the same number of operators are required for the NET Power plant as for an NGCC plant, an additional €2.2M/yr. in fixed costs savings can be expected. Actual “bottom-up” O&M costs for a NET

Power plant indicate that costs will be even lower than NGCC, requiring fewer operators and less maintenance due to the factors mentioned above.

- As discussed in NET Power's commentary to Chapter D.2, actual efficiency of the NET Power process, in an "early commercial" plant configuration and modelled according to the study specifications, is 58.8% LHV. Despite an offer by 8 Rivers to disclose the confidential information that would enable validation of this performance, AmecFWI declined and subsequently used their lower modelled efficiency in their estimate. The actual performance of the system is a significant improvement over the 55.4% LHV as modelled by AmecFWI and has a proportionately large positive effect on the specific cost and LCOE of the NET Power plant.
- Incorporating the above reductions and increased performance of the NET Power system results in a specific cost of €1,068/kW ($1,345/kW) and an LCOE of €73.8/MWh for a system with full CO2 capture at pipeline conditions (including CO2 transport and storage costs). This is in close agreement with NET Power's estimated "early commercial" specific cost of €992/kW ($1,250/kW).

4. Estimates for Conventional NGCC

The NGCC estimate produced by AmecFWI is on the low end of the range of costs NET Power and its partners have observed for newly constructed plants. This exemplifies a trend of recent NGCC estimates reflecting market dynamics that balance aggressive equipment pricing with potential technical services revenue. In many cases, NGCC equipment vendors (particularly for the gas turbine) are lowering upfront equipment price in exchange for realising deferred revenue through long-term service agreements (LTSAs). This is particularly true for projects located in deregulated markets, such as the Netherlands which tend to see a higher cost of capital for new projects, favouring a lower equipment cost over longer term LTSA expenditures.

As the low capital cost estimate generated by AmecFWI is based on recent vendor quotes for a Netherlands location, it likely conforms to this trend. However, the calculation of operational costs in the Report is based on historical "rules-of-thumb", which would not capture the higher LTSA and other service agreements that would be tied to such a low capital estimate. This inconsistency is highly favourable for the NGCC estimate generated for this Report, as these costs would significantly increase the overall cost of the plant. No such inconsistency exists for the NET Power plant, where cost estimates are reflective of the true capital cost.

5. Conclusion

The study specification for the plant cost analysis undertaken in the Report requires estimation of "early commercial" plant costs for new technologies. However, estimates for the incumbent NGCC technology represent true "nth-of-a-kind" estimates given the decades of technological advancement and cost maturity this system has experienced. Furthermore, variability in the markets for new power generation construction include the balancing of equipment pricing with LTSA structures which result, in some cases, in understated pricing for new projects. Comparisons between AmecFWI's cost estimates for "early commercial" NET Power plants and the NGCC "nth-of-a-kind" estimates must be viewed in that context.

The detailed design, pre-FEED and cost estimation studies undertaken by NET Power, 8 Rivers and their partners, based on currently available equipment and technologies but considering "nth-of-a-kind" factors, indicate the specific total plant cost to be in the range of $829/kW to $1,250/kW (€658/kW to €992/kW, median €830/kW) for a US-based NET Power plant that captures approximately 1.6 million tonnes of export-ready high pressure CO_2 at a pressure of 150 bar. Based on reasonable market assumptions, NET Power views these estimates to be highly competitive with NGCC plants that do not have carbon capture, especially when considering the regulatory certainty NET Power provides to plant owners and/or the additional revenue potential from CO_2 sales (or avoided CO_2 tax), and even more so when sales of additional industrial gases produced by the ASU are considered. These costs also represent a significant improvement over all other carbon capture technologies.

May 2015

NET Power, LLC, 8 Rivers Capital, LLC, 406 Blackwell Street, Durham, North Carolina, 27701, USA

Email: info@netpower.com

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : CASE 3A: S-GRAZ
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 3a of the study, which is the S-GRAZ cycle based power plant, with cryogenic purification and separation of the carbon dioxide. The plant is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected S-GRAZ plant configuration is based on two parallel trains, each composed of one F-class equivalent oxy-fired gas turbine and one heat recovery steam generator (HRSG), generating steam at one pressure level for expansion in a back-pressure steam turbine, this latter providing the cooling medium for the turbine blades.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 3a – Unit arrangement

Unit	Description	Trains
3000	Power Island	
	Gas Turbine	2 x 50%
	Recycle gas re-compressor (included in GT package)	2 x 50%
	HRSG + Back pressure steam turbine	2 x 50%
	Gas Turbine low pressure expander	2 x 50%
	Flue gas condenser	2 x 50%
4000	CO_2 purification and compression	
	Raw gas compression	2 x 50%
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power Island

The unit is mainly composed by two trains, each including:

- One F-class equivalent oxy-fired gas turbine.
- One heat recovery steam generator (HRSG), generating steam at one pressure level.
- One back pressure steam turbine, expanding the steam to the pressure level required for turbine blades cooling.
- One low pressure expander.
- One flue gas condenser.
- Recycled gas compression train (part of the gas turbine package).

Technical information relevant to this unit is reported in chapter D, section 2.1.3, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.2.1. Gas Turbine expander design features

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the streams at gas turbine boundary are summarised in the following Figure 1.

The natural gas from the let-down and metering station is heated using heat available from the CO_2 compression in the CPU before entering the burners of the gas turbine at 100°C.

Oxygen is delivered from the oxygen compressor in the ASU at the required pressure level and enters the burners of the gas turbine at 150°C.

The fuel and oxidant streams are combined with the recycled water and CO_2 rich stream, which is around half of the expanded flue gas downstream the HRSG. Additional cooling is provided by the steam generated in the HRSG after expansion in a back-pressure steam turbine to the pressure level required by the turbine. The recycled gas flowrate to the combustor is set in order to control the combustion outlet temperature at 1533°C. The steam flowrate is the excess resulting from the steam generation after supplying the cooling steam to the turbine to control the blades metal temperature.

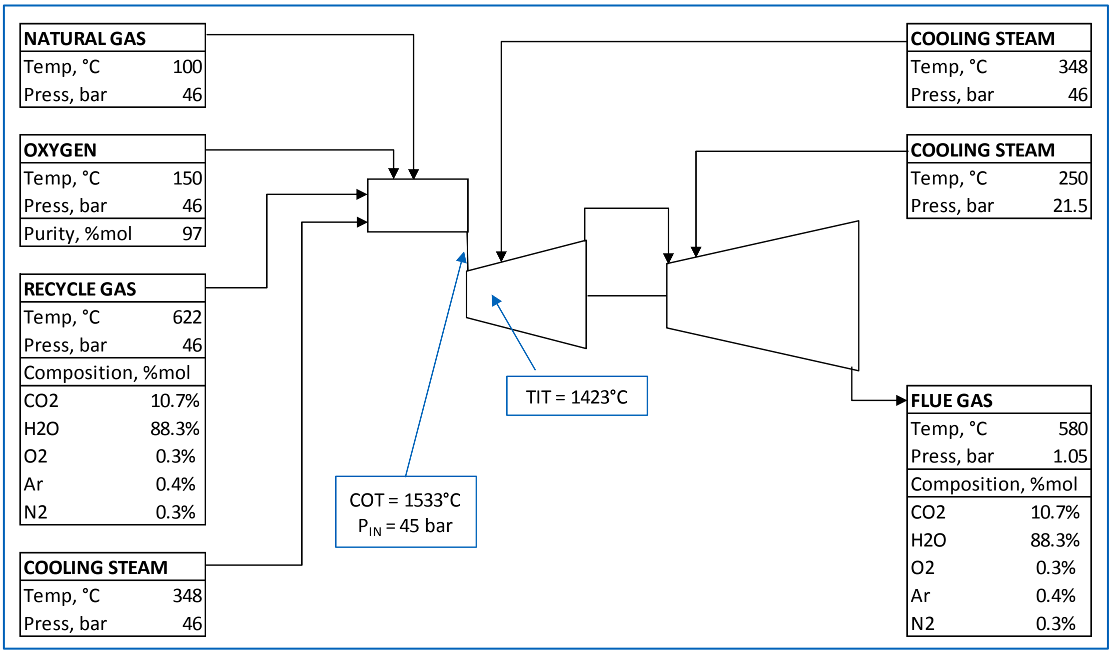


Figure 1. S-GRAZ gas turbine

As for the SCOC-CC, the turbine inlet pressure is set at 44.5 bar. A pressure slightly higher than the ambient pressure (to avoid leakages into the CO_2 loop) has been selected for the turbine discharge so as to keep the design of the turbomachines closer to the current standards.

The gas turbine expander includes two sections. The high pressure section has one stage, at a rotational speed of 4600 RPM, and it drivers one of the two recycle compressors. The second section has 5 stages at a rotational speed of 3000 RPM. Stages number and rotational speed are set so to have acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine.

2.2.2. *Heat recovery section*

The exhaust gases from the gas turbine enter the HRSG at 580°C. The HRSG recovers heat available from the exhaust gas by producing super-heated high pressure steam to be expanded in a back-pressure steam turbine. The boiler feed water from the deaerator enters the HRSG at around 105°C, while steam is sent at the steam turbine inlet at 170 bar and 555°C. The typical 25°C approach temperature between steam and exhaust gas temperature is considered to have an adequate heat transfer coefficient and limit the coils surface.

Steam is expanded in a back pressure steam turbine with an intermediate extraction at around 45 bar and discharge at around 21 bar. Steam from the controlled extraction is injected in the combustion chamber and in the first stage of the turbine, while the exhaust steam is used as cooling stream in lower pressure stages.

The cooling flowrate is set to control the gas turbine blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st stator	860°C
1st rotor	830°C
from 2nd stator	830°C
from 2nd rotor	800°C

Recovering steam from the HRSG, in addition to the power generation in the steam turbine, enhances the plant efficiency. In fact, acting as cooling medium, it allows reducing the amount of flue gas to be recycled and consequently the compressor power consumption.

2.2.3. *Flue gas low pressure expander and condenser*

A fraction (around 50%) of the flue gas from the HRSG not recycled back to the combustion chamber is expanded down to condenser pressure. At the temperature level allowed by the cooling water temperature (i.e. 29°C), the condensing pressure is optimised by considering the following opposite effects: a lower condensation pressure leads to a higher power output from the expander, but also to a higher consumption of the wet flue gas compressor, due to the lower inlet pressure and the higher water content.

As shown in the following Figure 2, at the condensing temperature of 29°C assumed for this study case, the optimum condensation pressure is 7 kPa.

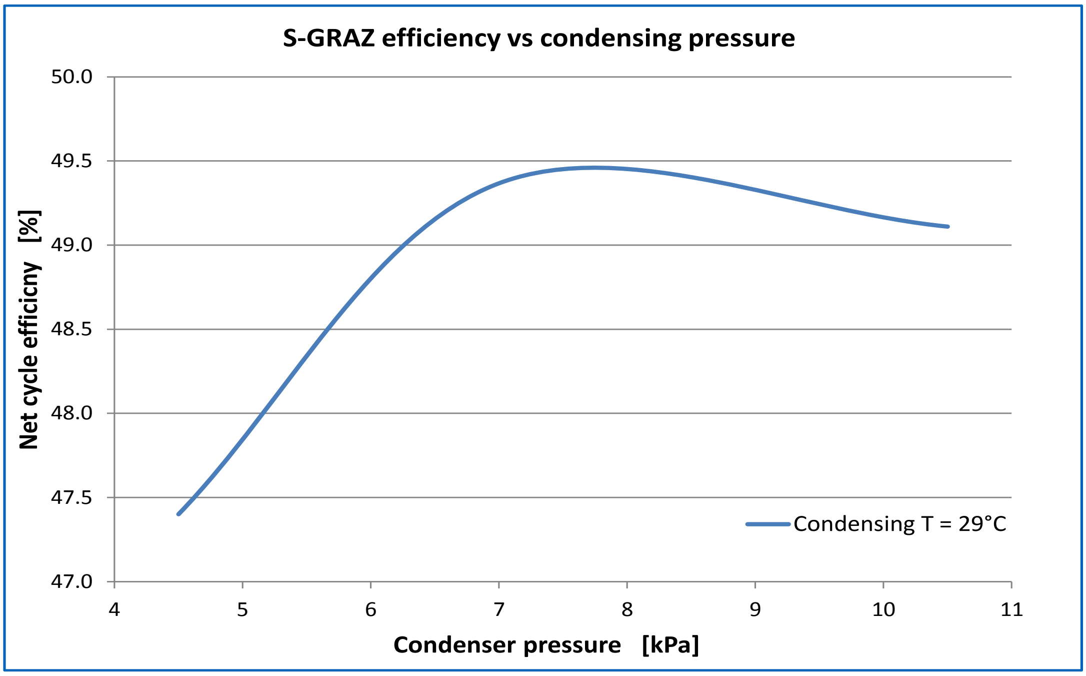


Figure 2. S-Graz cycle: optimum condensation pressure

The wet flue gas from the condenser is compressed to atmospheric pressure and sent to the CPU. The heat from the wet flue gas compressor intercoolers, as well as from the raw gas compressor intercoolers in the CPU, is recovered pre-heating the condensate for steam generation in the HRSG.

2.2.4. *Heat integration*

The oxy-fuel cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular, heat available at high temperature level from the oxy-turbine cycle is used to provide heat required in the CPU mainly for TSA regenerator heating and inert gas heating before expansion, while heat available at low temperature level in the CO_2 compressor intercoolers is used for oxygen and natural gas heating to enhance gas turbine efficiency.

The following interfaces have been considered:

- Natural gas is heated using as heating medium compressed CO_2 from the final compression before being sent to plant B.L.
- Heat downstream the first stage of the recycle gas compressor is used as heating medium in the TSA regenerator and inerts gas heaters of the CPU.
- Heat downstream raw gas compressor in the CPU and downstream wet flue gas compressor in the oxy-cycle is used for condensate pre-heating.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 16 bar. A dedicated compressor is installed to compress the oxygen up to the pressure and the temperature level required by the gas turbine combustor. As no heat is available from the process, oxygen pre-heating is performed by designing the ASU to produce oxygen at lower pressure and by taking the advantage of the heat of compression to increase oxygen temperature.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric requirement, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- Raw flue gas compression (0.07 - 34 bar);
- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

Basin diameter	120 m
Cooling tower height	210 m
Water inlet height	17 m

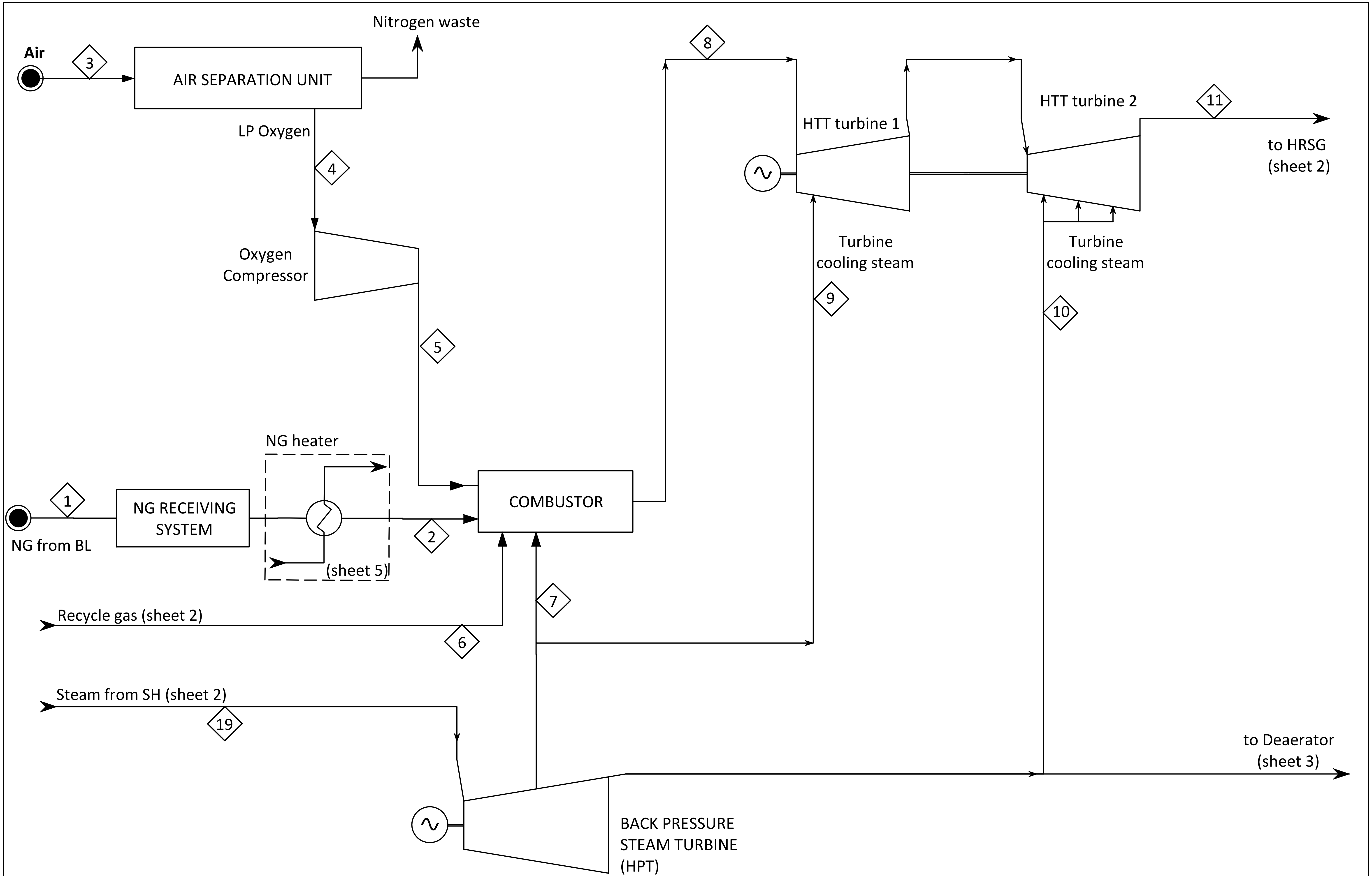
- Natural gas receiving station;
- Raw water system;

- Demineralised water plant;
- Waste Water Treatment
- Fire fighting system;
- Instrument and Plant air.

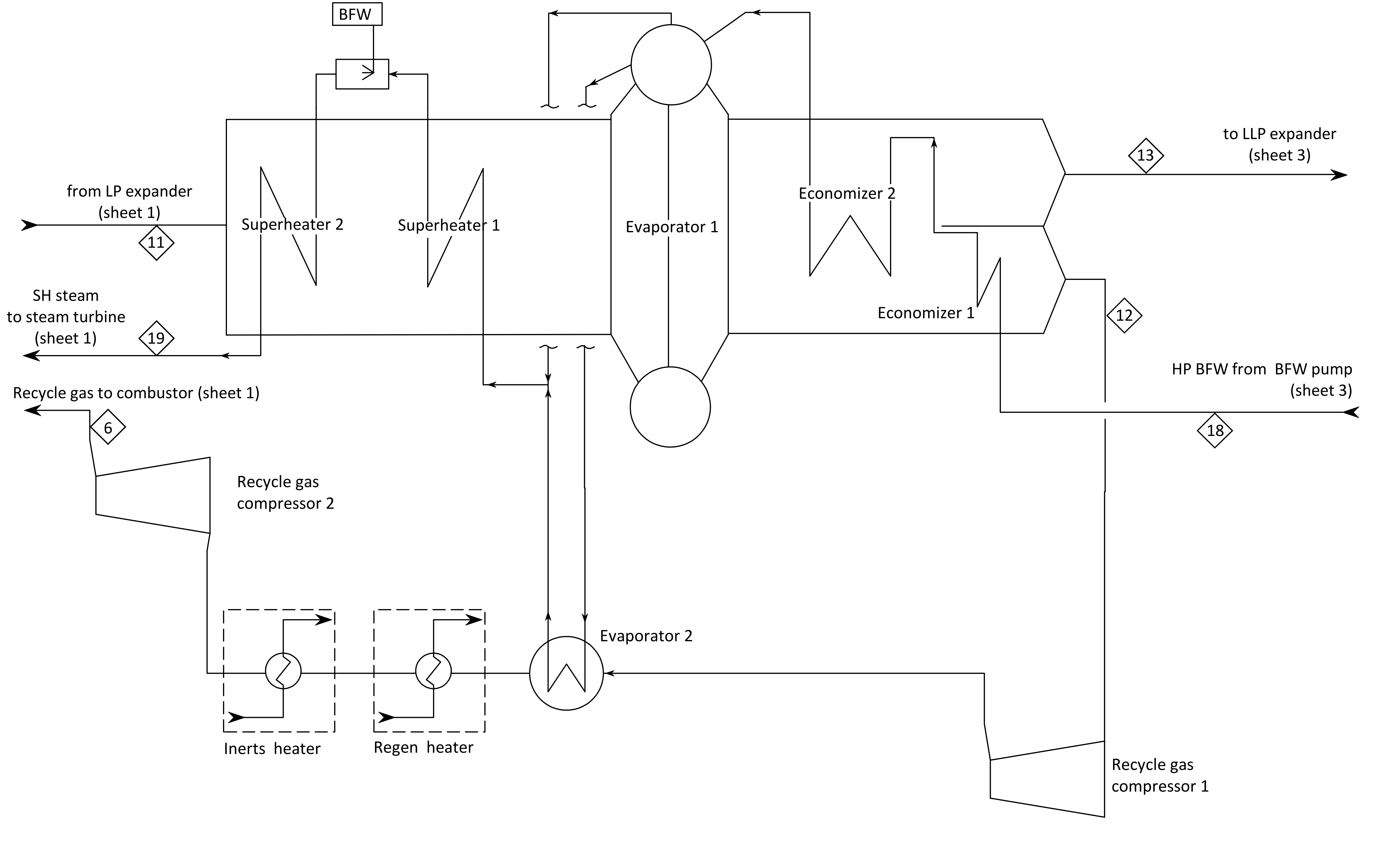
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

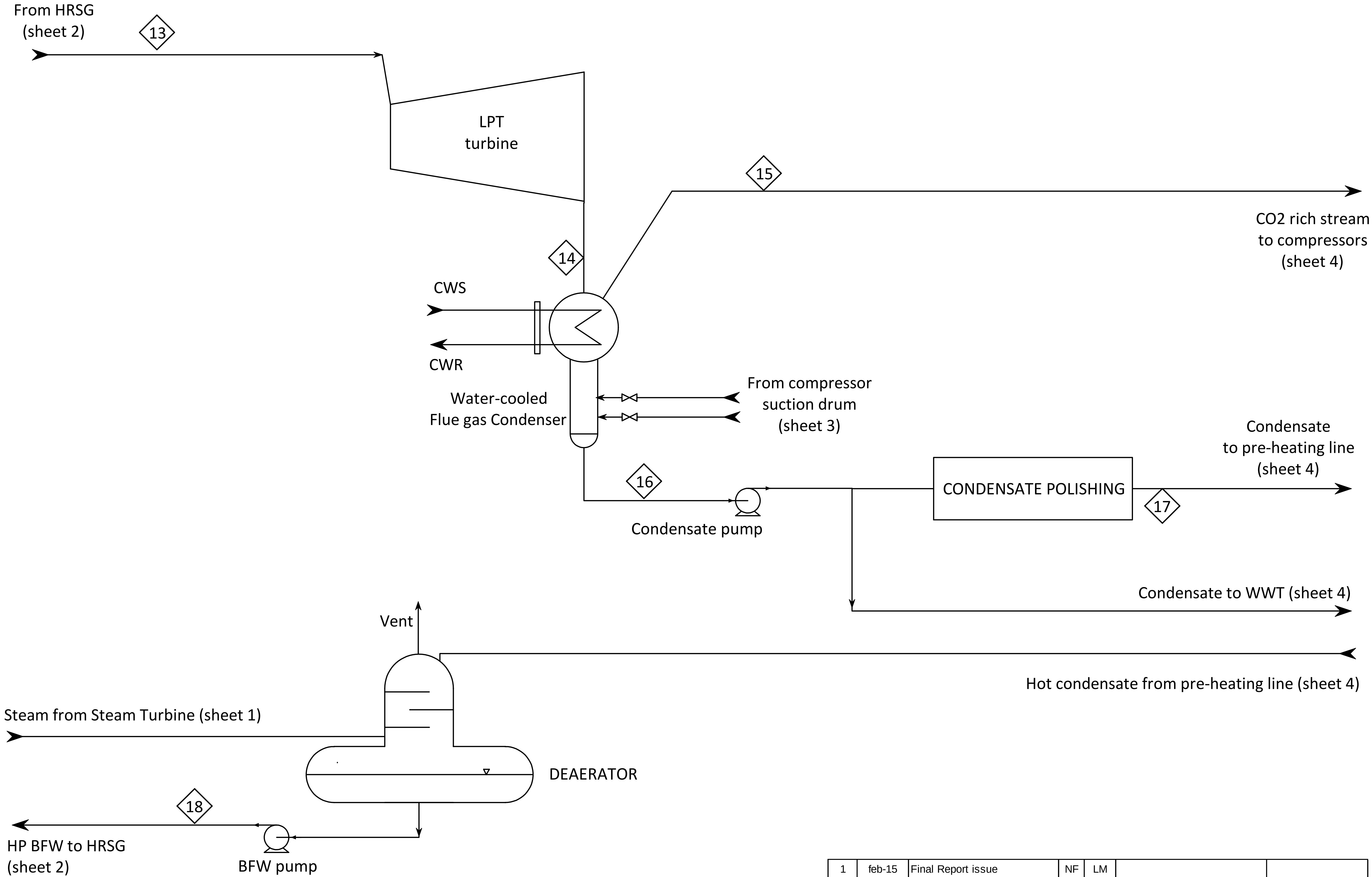
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



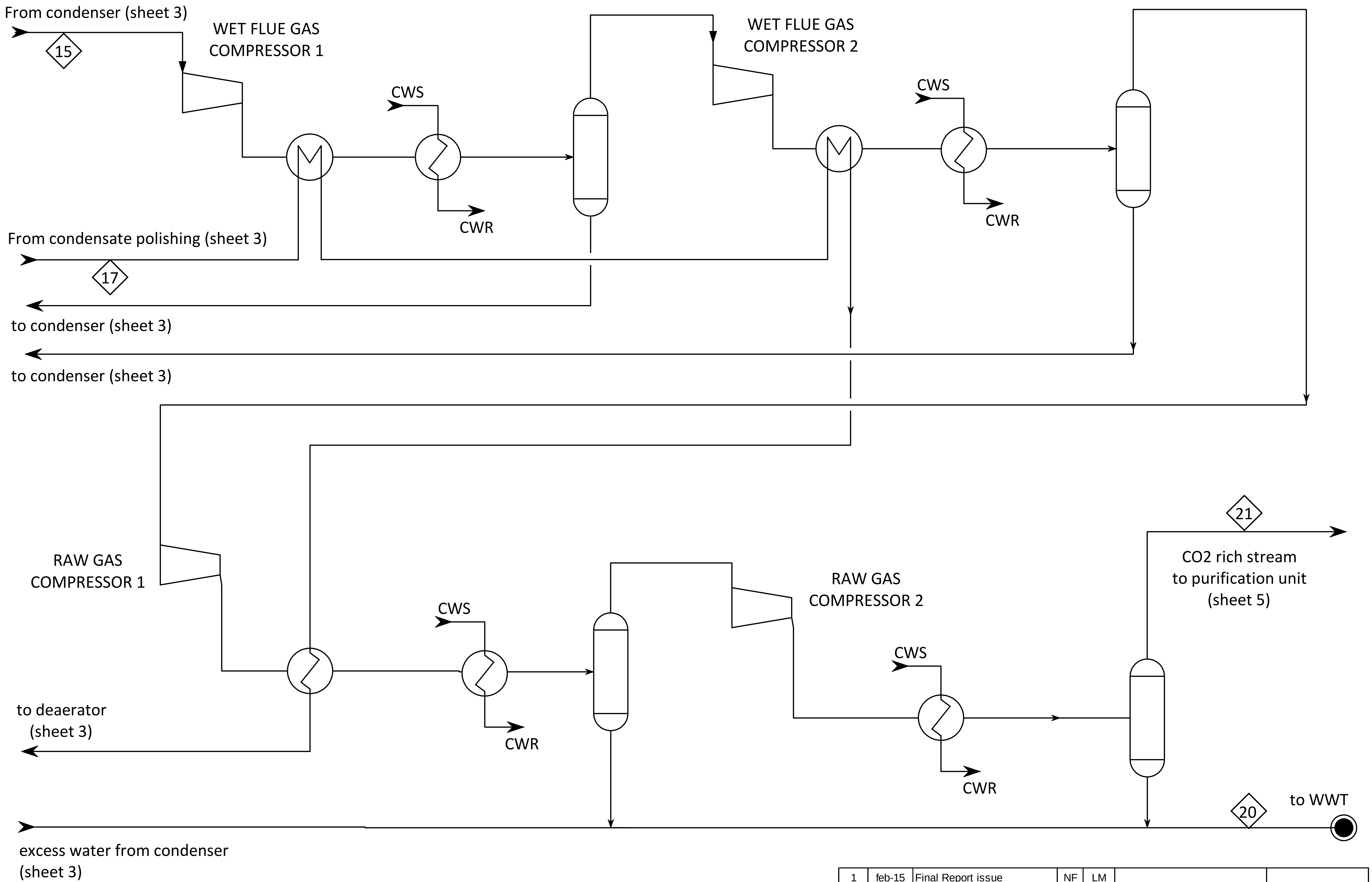
1	feb-15	Final Report issue	NF	LM		
0	Jun 2014	Draft report issue	NF	LM	**UNIT:** Oxyturbine (HHTandHPT)	
Rev.	Date	Comment	By	Appr	**CASE:** 3 - S-GRAZ CYCLE	Sheet 01 of 05



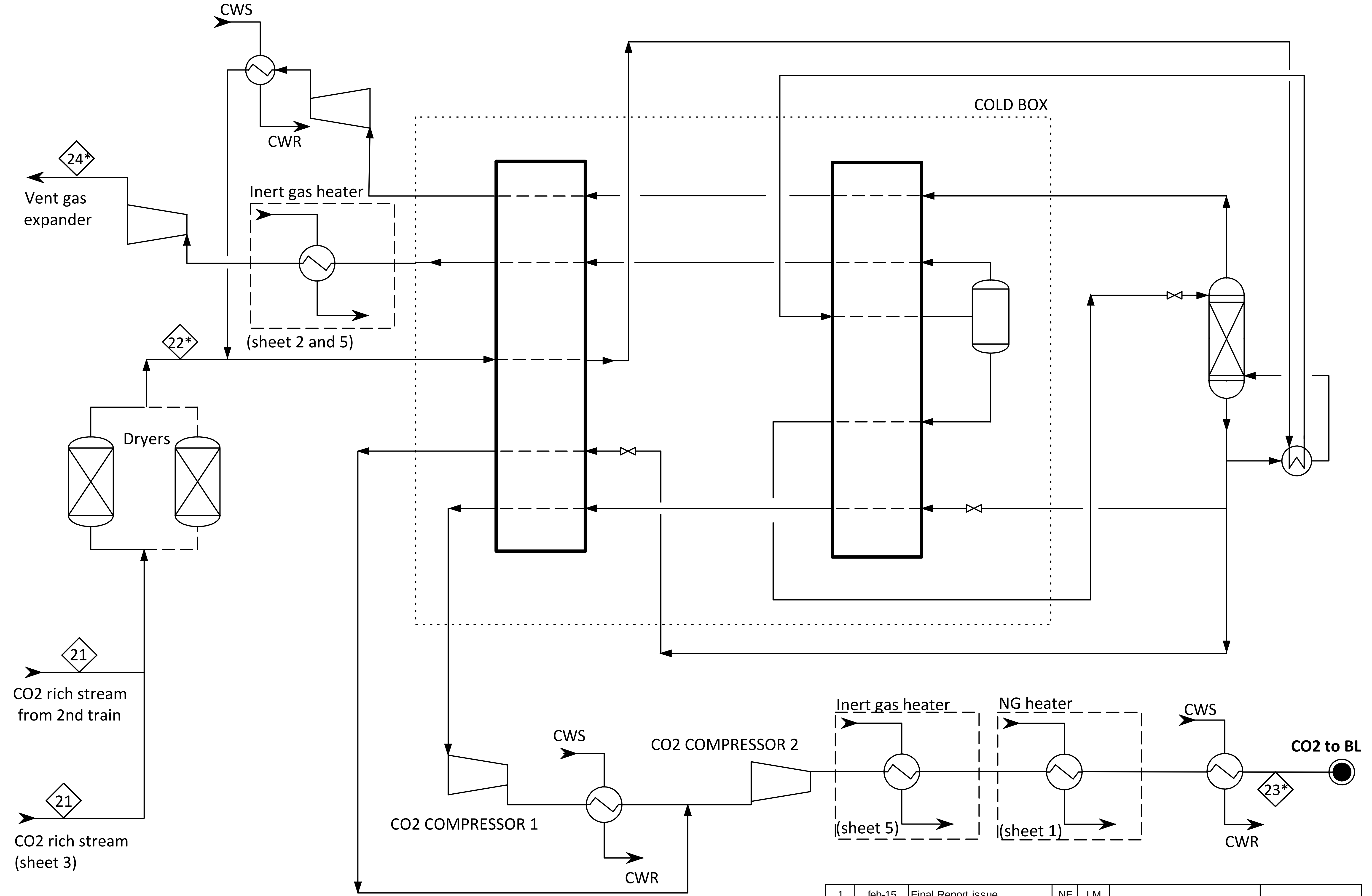
1	feb-15	Final Report issue	NF	LM		
0	Jun 2014	Draft report issue	NF	LM	**UNIT:** HRSG and recycle gas compressors	
Rev.	Date	Comment	By	Appr	**CASE:** 3 - S-GRAZ CYCLE	Sheet 02 of 05



1	feb-15	Final Report issue	NF	LM		
0	Jun 2014	Draft report issue	NF	LM	**UNIT:** LPT turbine	
Rev.	Date	Comment	By	Appr	**CASE:** 3 - S-GRAZ CYCLE	Sheet 03 of 05



1	feb-15	Final Report issue	NF	LM		
0	mag-14	Draft report issue	NF	LM	**UNIT:** CO_2 compression section	
Rev.	Date	Comment	By	Appr	**CASE:** 3 - S-GRAZ CYCLE	Sheet 04 of 05



1	feb-15	Final Report issue	NF	LM		
0	Jun 2014	Draft report issue	NF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** 3 - S-GRAZ CYCLE	Sheet 05 of 05

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

	Case 3a - S-GRAZ - HEAT AND MATERIAL BALANCE		REVISION	0	1	
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	FF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	NF	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	June 2014	February 2015	

HEAT AND MATERIAL BALANCE
UNIT 3000 - OXY-FUEL CYCLE

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	Heated NG to combustor	Air to ASU	Oxygen from ASU	Oxygen to combustor	Recycle gas	Cooling steam to combustor	Flue gas to HP turbine	Cooling steam to HP turbine	Cooling steam to LP turbine
Temperature (°C)	15	100	9	15	150	622	348	1533	348	248
Pressure (bar)	70	45.8	amb	16	45.8	45.8	46.2	44.5	46.20	21.00
TOTAL FLOW										
Mass flow (kg/h)	59,470	59,470	1,006,640	230,615	230,615	901,330	75,660	1,267,230	174,790	160,915
Molar flow (kmol/h)	3,300	3,300	34,880	7,180	7,180	43,000	4,200	57,840	9,700	8,930
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	59,470	59,470	1,006,640	230,615	230,615	901,330	75,660	1,267,230	174,790	160,915
Molar flow (kmol/h)	3,300	3,300	34,880	7,180	7,180	43,000	4,200	57,840	9,705	8,930
Molecular Weight (kg/kmol)	18.0	18.0	28.9	32.1	32.1	21.0	18.0	21.9	18.0	18.0
Composition (%mol)	as assigned	as assigned								
Ar			0.92%	2.00%	2.00%	0.43%	-	0.57%	-	-
CO_2			0.04%	0.00%	0.00%	10.70%	-	14.14%	-	-
H_2O			0.97%	0.00%	0.00%	88.28%	100.00%	84.51%	100.00%	100.00%
N_2			77.32%	1.00%	1.00%	0.30%	-	0.40%	-	-
O_2			20.75%	97.00%	97.00%	0.29%	-	0.38%	-	-
Total			100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 3a - S-GRAZ - HEAT AND MATERIAL BALANCE		REVISION	0	1	
	CLIENT :	IEAGHG	**PREP.**	FF	NF	
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	NF	LM	
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM	LM	
	LOCATION:	The Netherlands	**DATE**	June 2014	February 2015	

HEAT AND MATERIAL BALANCE
UNIT 3000 - OXY-FUEL CYCLE

STREAM	11	12	13	14	15	16	17	18	19	20
	Flue gas to HRSG	Flue gas recycle	Flue gas to LLP turbine	LLP tubine exhauts gas	Wet flue gas from condenser	Water from condenser	Condensate to pre-heating line	BFW to HRSG	HP steam to back-pressure steam turbine	Waste water to WWT
Temperature (°C)	580	292	292	67	29	29	29	90	555	29
Pressure (bar)	1.05	1.02	1.02	0.07	0.07	0.07	2.5	197	170	2.5
TOTAL FLOW										
Mass flow (kg/h)	1,602,935	901,300	701,635	701,635	269,155	529,650	411,370	411,370	411,370	120,890
Molar flow (kmol/h)	76,475	43,000	33,475	33,475	9,470	29,400	22,835	22,835	22,835	6,710
LIQUID PHASE										
Mass flow (kg/h)						529,650	411,370	411,370		120,890
GASEOUS PHASE										
Mass flow (kg/h)	1,602,935	901,300	701,635	701,635	269,155				411,370	
Molar flow (kmol/h)	76,475	43,000	33,475	33,475	9,470				22,835	
Molecular Weight (kg/kmol)	21.0	21.0	21.0	21.0	28.4				18.0	
Composition (%mol)										
Ar	0.43%	0.43%	0.43%	0.43%	1.52%	0.00%	0.00%	-	-	0.00%
CO_2	10.70%	10.70%	10.70%	10.70%	37.82%	0.00%	0.00%	-	-	0.00%
H_2O	88.28%	88.28%	88.28%	88.28%	58.57%	100.00%	100.00%	100.00%	100.00%	100.00%
N_2	0.30%	0.30%	0.30%	0.30%	1.07%	0.00%	0.00%	-	-	0.00%
O_2	0.29%	0.29%	0.29%	0.29%	1.02%	0.00%	0.00%	-	-	0.00%
Total	100%	100%	100%	100%	100%	100%	100%	100%	100%	100%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 3a - S-GRAZ - HEAT AND MATERIAL BALANCE		REVISION	0	1	
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	FF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	NF	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	June 2014	February 2015	

HEAT AND MATERIAL BALANCE
UNIT 4000 - CPU

STREAM	21	22*	23*	24*						
	CO2 rich gas to TSA	CO2 rich stream to purification unit	Purified CO2	Vent gas from CPU						
Temperature (°C)	28	30	29	83						
Pressure (bar)	34.50	34.00	110	1.1						
TOTAL FLOW										
Mass flow (kg/h)	169,370	338,550	284,450	54,100						
Molar flow (kmol/h)	3,930	7,845	6,465	1,380						
LIQUID PHASE										
Mass flow (kg/h)			284,450							
GASEOUS PHASE										
Mass flow (kg/h)	169,370	338,550		54,100						
Molar flow (kmol/h)	3,930	7,845		1,380						
Molecular Weight (kg/kmol)	43.1	43.2		39.2						
Composition (%mol)										
Ar	3.66%	3.67%	0.20%	19.85%						
CO_2	91.13%	91.29%	99.79%	51.69%						
H_2O	0.18%	0.00%	0.00%	0.00%						
N_2	2.58%	2.58%	0.00%	14.65%						
O_2	2.46%	2.46%	0.01%	13.81%						
Total	100%	100%	100%	100%						

NOTE

1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 3a – Water consumption summary

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	Jul-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 3a - S-GRAZ cycle

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water DT = 11°C [t/h]
3000	**OXY-TURBINE CYCLE**			
	Condenser			**47,520**
	Turbine and generator Auxiliaries		**10**	**4,170**
	Condenser compressor after coolers			**10,530**
5000	**AIR SEPARATION UNIT**			
	MAC intercoolers			**8,910**
	BAC intercoolers			**830**
4000	**CO_2 PURIFICATION UNIT**			
	CO_2 purification unit			**2,560**
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	**1,375**		
	Demineralized water unit	**15**	**-10**	
	Waste Water Treatment and Condensate Recovery	**-230**		
	Balance of plant			
	BALANCE	**1,160**	**0**	**74,520**

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 3a – Electrical consumption summary

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Jul-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 3a - S-GRAZ cycle

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**OXY-TURBINE CYCLE**	
	Condensate and recycle water system	**6,450**
	Flue gas compression (from condenser)	**35,410**
	Turbine Auxiliaries + generator losses	**5,600**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**123,200**
	Booster air compressor and miscellanea	**11,160**
	Oxygen compressor	**16,250**
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	**28,340**
	Autorefrigerated inerts removal unit compression consumption	**14,020**
	Autorefrigerated inerts removal unit expander production	**-3,250**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**9,330**
	Balance of plant	**1,460**
	BALANCE	**247,970**

6. Overall performance

The following table shows the overall performance of Case 3a, including CO_2 balance and removal efficiency.

FOSTER WHEELER

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Jul-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 3a - S-GRAZ cycle

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
HTT turbine power output	MWe	1292.0
HPT turbine power output	MWe	90.9
LPT turbine power output	MWe	148.0
Turbine recycle gas compressors	MWe	-522.2
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**1008.7**
Oxy-turbine cycle	MWe	47.5
Air separation unit + Oxygen compressor	MWe	150.6
CO2 purification and compression unit	MWe	39.1
Utility & Offsite Units	MWe	10.8
ELECTRIC POWER CONSUMPTION	**MWe**	**248.0**
NET ELECTRIC POWER OUTPUT	**MWe**	**760.8**
(Step Up transformer efficiency = 0.997%) (B)	MWe	758.5
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**65.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.4%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**59.3%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**44.7%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6452
CO_2 removal efficiency	**%**	**90.1**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**
CO_2 emission per net power production	**kg/MWh**	**41.5**

6.1. Comparison with literature performance data

Scope of this section is to compare the performance data of this Case 3a with those reported in the public domain, particularly with data of the paper presented at the ASME Turbo Expo 2005 ([1]), relevant to a plant with the same configuration of the present case.

The following Table 4 summarises the cycle performance, in terms of electrical efficiency for:

- This study Case 3a, considering the configuration and the design basis, in terms of combustion outlet temperature, turbine inlet temperature, natural gas composition, 97%mol oxygen purity and especially turbine cooling flow requirements, as described in previous sections.
- Literature data, published by the GRAZ university.

For each of the above listed conditions, main design features potentially affecting the performance are also highlighted.

Table 4. Case 3a – Performance comparison

	Case 3a	**Graz cycle (as per ASME Turbo Expo 2005, [1])**
Design parameters		
Condensation pressure	7 kPa	4 kPa
CO_2 purification	YES (O_2 content: 100 ppm)	NO
CO_2 compression	110 barg	100 barg
Performance		
Gross electrical efficiency (1)	63.4%	66.5%
Cycle net electrical efficiency (2)	61.9%	64.6%
Plant net electrical efficiency (3)	49.4%	52.6%

Notes

(1) Calculated considering net power production from the turbines, subtracting total compression power demand.

(2) Including power cycle auxiliaries and utility consumptions, no CPU and ASU included

(3) Including CO_2 compression and ASU consumptions. Only CO_2 compression is considered in the reference case, while CPU only foreseen in the study case to meet CO_2 purity specification (i.e. oxygen content)

[1] W.Sanz, H. Jericha, F. Luckel, E. Göttlich, F. Heitmeir, *A further step towards a Graz cycle power plant for CO_2 capture*, Proceeding of ASME Turbo Expo 2005, June 6-9, 2005, Nevada (US)

The efficiency loss of Case 3a with respect to the reference literature case is around three (3) percentage points, mainly due to the lower gross power production. The following considerations can be made in order to explain this difference:

- The condensation pressure is lower in the case presented at the ASME Turbo conference, leading to around one percentage point difference.
- The CO_2 purification requires around 25-30% addition power consumption with respect to the power demand of the CO_2 compression alone, which accounts for around 0.5 percentage point efficiency difference.
- The cooling stream to the second section of the HTT gas turbine is higher in the study Case 3a with respect to the reference literature case; consequently the efficiency is lower. Performance data obtained by considering the same turbine efficiency and cooling stream flowrate are summarized in the following Table 5, showing that the related efficiency loss is around the residual two percentage points.

Table 5. Case 3a – Performance comparison (same gas turbine)

	Case 3a (same gas turbine as reference case)	**Graz cycle (as per ASME Turbo Expo 2005)**
Performance		
Gross electrical efficiency	65.4%	66.5%
Cycle net electrical efficiency	64.0%	64.6%
Plant net electrical efficiency	51.4%	52.6%

7. Environmental impact

The oxy-combustion gas turbine plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 6 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power island and in the CO_2 purification unit.

Table 6. Case 3a – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	54,230
Flow, Nm^3/h	31,000
Composition	(%mol)
Ar	19.85%
N_2	14.65%
O_2	13.81%
CO_2	51.69%
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from the flue gas condenser/ compressor intercoolers and CO_2 purification unit) are treated to recover water, so the main liquid effluent is the cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 320 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 3a - S-GRAZ cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	OXY TURBINE PACKAGE							
PK- 3101-1/2	**Oxy Turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
T- 3101	HTT turbine 1		141 MWe Pin: 44.5 bar; Pout: 21 bar					*Including:* *Lube oil system* *Cooling system*
T- 3102	HTT turbine 2		506 MWe Pin: 21 bar; Pout: 1.05 bar					*Hydraulic control system* *Seals system*
T- 3103	LPT turbine		74 MWe Pin: 1.05 bar; Pout: 7 kPa					*Drainage system*
F- 3101	Combustor		770 MWth					
K- 3101	Recycle gas compressor							
	1st stage		140 MWe					
	2nd stage		70 MWe					
G- 3101	Oxy turbine generator (HP+LP stage)		540 MVA					*Including relevant auxiliaries*
G- 3102	Oxy turbine generator (LLP stage)		90 MVA					*Including relevant auxiliaries*
	HEAT RECOVERY STEAM GENERATOR							
PK- 3201-1/2	**Heat recovery steam generator**	Horizontal, Natural Circulated, 1 Pressure Level						***2 x 50% HRSG package***
D- 3201	HP steam drum							
E- 3201	HP Economizer 1st section							
E- 3202	HP Economizer 2nd section							
E- 3203	HP Evaporator							
E- 3205	HP superheater							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	BFW pump	Centrifugal	450 m3/h x 2165 m	3400 kW				*One operating one spare, per each train*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3a - S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	HEAT RECOVERY STEAM GENERATOR							
	DRUM							
	Deaerator							
	Steam generator blowdown drum (continuous)							
	Intermittent blowdown drum							
	EXCHANGER							
	Blowdown cooler							
	PACKAGES (common to both trains)							
PK- 3202	**Fluid Sampling Package**							
PK- 3203	**Phosphate Injection Package**							
	Phosphate storage tank							
	Phosphate dosage pumps							*One operating one spare*
PK- 3204	**Oxygen scavenger Injection Package**							
	Oxygen scavenger storage tank							
	Oxygen scavenger dosage pumps							*One operating one spare*
PK- 3204	**Amine Injection Package**							
	Amine storage tank							
	Amine dosage pumps							*One operating one spare*
	BACK-PRESSURE STEAM TURBINE							
PK- 3301-1/2	**Back pressure steam turbine and Generator Package**							
	Including:							
ST- 3301	Back-pressure steam turbine (HPT turbine)		46 MWe HP steam inlet: 170 bar MP steam extraction: 45 bar Exhaust pressure: 21 bar					*Including:* *Lube oil system* *Cooling* *Hydraulic control system* *Seals system (including gland condenser and vacuum system)* *Drainage system*
G- 3301	Steam turbine generator		60 MVA					*Including relevant auxiliaries*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3a - S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
FOSTER WHEELER		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
FLUE GAS CONDENSER PACKAGE and COMPRESSION PACKAGE								
PK- 3302-1/2	**Flue gas Condenser Package**							***2 x 50% condenser package***
	Each including:							
	Flue gas condenser		310 MWth					*Including condenser hotwell*
	PUMPS		**Q [m3/h] x H [m]**					
P- 3301 A/B	Flue gas condensate pump	Centrifugal	585 m3/h x 50 m	110 kW				*One operating one spare, per each train*
PK- 3303-1/2	**Wet flue gas compressors**							***2 x 50% compression train***
	- *Wet flue gas compressor #1*	axial	Flowrate: 2 x 110,000 Nm3/h Pin: 7 kPa; Pout : 20 kPa Compression ratio: 2.9	*2 x 5,100 kW*				*Two operating per each train*
	- *Wet flue gas compressor #2*	axial	Flowrate: 2 x 55,000 Nm3/h Pin: 19.4 kPa; Pout : 105 kPa Compression ratio: 5.4	*2 x 4,650 kW*				*Two operating per each train*
	Condesate separators							
	Intercoolers							
	Condensate pre-heater #1							
	Condensate pre-heater #1							
	CW cooler #1							
	CW cooler #2							

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3a - S-GRAZ cycle			REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
				DATE	Jun-14	Feb-15		
				ISSUED BY	NF	NF		
				CHECKED BY	LM	LM		
				APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 4000 - CO2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	PACKAGES							
PK - 4001	**CO2-rich gas compression**							
	Including:							
	Raw flue gas compressors							
	- Raw flue gas compressor #1	axial	Flowrate: 2 x 92,000 Nm3/h Pin: 1.02 bar; Pout : 15 bar Compression ratio: 14.7	*2 x 12.5 MWe*				*2x50%*
	- Raw flue gas compressor #2	axial	Flowrate: 2 x 98,000 Nm3/h Pin: 14.4 bar; Pout : 35 bar Compression ratio: 2.44	*2 x 3.0 MWe*				*2x50%*
	Condesate separators							
	Intercoolers							
	Condensate pre-heater #1							
	Condensate pre-heater #1							
	CW cooler #1							
	CW cooler #2							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO2 liquid separator							
	- CO2 distillation column							
	- CO2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MWe*				*2x50%*
	- CO2 compressor 1	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		2900 kW					*1x100%*
	- Overhead recycle compressors (optional)							
	- Intercoolers							
	Inert gas heater							
	NG heater							
	Cooling water intercoolers							

FOSTER WHEELER	CLIENT: IEAGHG		REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2	
	LOCATION: The Netherlands		DATE	Jun-14	Feb-15			
	PROJ. NAME: Oxy-turbine power plants		ISSUED BY	NF	NF			
	CONTRACT N.: 1-BD-0764 A		CHECKED BY	LM	LM			
	CASE.: 3a - S-GRAZ cycle		APPROVED BY	LM	LM			

EQUIPMENT LIST

Unit 5000 - Air Separation Unit (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5500 t/d					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 34.5 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		5.0 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
	- Oxygen Compressors	Centrifugal		9,000 kWe				
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3a - S-GRAZ cycle				REVISION DATE ISSUED BY CHECKED BY APPROVED BY	Rev.: Draft Jun-14 NF LM LM	Rev.: 0 Feb-15 NF LM LM	Rev.1 / Rev.2

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower** including: Cooling water basin	Natural draft	960 MWth Diameter: 115 m, Height: 180 m, Water inlet: 17 m				concrete	
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/.../G	**Cooling Water Pumps (primary system)**	Centrifugal	13,500 x 40	1750				*Six in operation, one spare*
P- 6003 A/B	**Cooling tower make up pumps**	Centrifugal	1450 x 30	160				*One in operation, one spare*
	PACKAGES							
	Cooling Water Filtration Package Cooling Water Sidestream Filters		7700 m3/h					
	Sodium Hypochlorite Dosing Package Sodium Hypochlorite storage tank Sodium Hypochlorite dosage pumps							
	Antiscalant Package Dispersant storage tank Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
PK- 6002	Let down station							

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 3a - S-GRAZ cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	RAW WATER SYSTEM							
PK- 6003	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Portable water pumps	centrifugal						*One operating, one spare*
	DEMINERALIZED WATER SYSTEM							
PK- 6004	Demin Water Package, including:							
	- Multimedia filter							
	- Reverse Osmosis (RO) Cartidge filter							
	- Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : CASE 3B: MODIFIED S-GRAZ
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 3b of the study, which is the Modified S-GRAZ cycle based power plant, with cryogenic purification and separation of the carbon dioxide. The plant is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected Modified S-GRAZ plant configuration is based on two parallel trains, each composed of one F-class equivalent oxy-fired gas turbine and one heat recovery steam generator (HRSG), generating steam at one pressure level for expansion in a back-pressure steam turbine, this latter providing the cooling medium for the turbine blades.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 3b – Unit arrangement

Unit	Description	Trains
3000	Power Island	
	Gas Turbine	2 x 50%
	Recycle gas re-compressor (included in GT package)	2 x 50%
	HRSG + Back pressure steam turbine	2 x 50%
	Wet flue gas compression and low pressure steam cycle	2 x 50%
4000	CO_2 purification and compression	
	Raw gas compression	2 x 50%
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power Island

The unit is mainly composed by two trains, each including:

- One F-class equivalent oxy-fired gas turbine.
- One heat recovery steam generator (HRSG) generating steam at one pressure level.
- One back pressure steam turbine expanding the steam to the pressure level required for turbine blades cooling.
- Wet flue gas compression and low pressure steam cycle.
- Recycled gas compression train (part of the gas turbine package).

Technical information relevant to this unit is reported in chapter D, section 2.3.1, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.2.1. Gas Turbine expander design features

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the streams at gas turbine boundary are summarised in the following Figure 1.

The natural gas from the let-down and metering station is heated using heat available from the CO_2 compression in the CPU before entering the burners of the gas turbine at 100°C.

Oxygen is delivered from the ASU at the required pressure level and heated using heat available from the raw gas compressor in the CPU before entering the burners of the gas turbine at 140°C.

The fuel and oxidant streams are combined with the recycled water and CO_2 rich stream, which is around half of the expanded flue gas downstream the HRSG. Additional cooling is provided by the steam generated in the HRSG, after expansion in a back-pressure steam turbine to the pressure level required by the turbine. The recycled gas flowrate to the combustor is set in order to control the combustion outlet temperature at 1533°C. The steam flowrate is the excess resulting from the steam generation after supplying the cooling steam to the turbine to control the blades metal temperature.

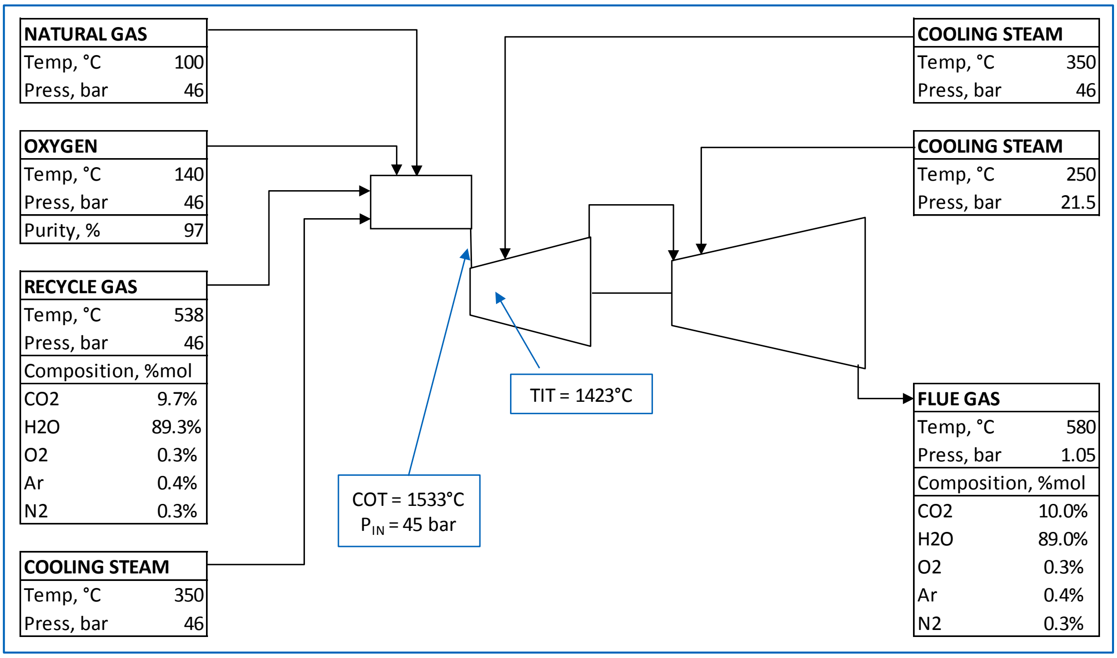


Figure 1. Modified S-GRAZ gas turbine

As for the SCOC-CC, the turbine inlet pressure is set at 44.5 bar. A pressure slightly higher than the ambient pressure (to avoid leakages into the CO_2 loop) has been selected for the turbine discharge so as to keep the design of the turbomachines closer to the current standards.

The gas turbine expander includes two sections. The high pressure section has one stage, at a rotational speed of 4600 RPM, and it drives one of the two recycle compressors. The second section has 5 stages at a rotational speed of 3000 RPM. Stages number and rotational speed are set so as to have acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine.

2.2.2. <u>*Heat recovery section*</u>

The exhaust gases from the gas turbine enter the HRSG at 580°C. The HRSG recovers heat available from the exhaust gas by producing super-heated high pressure steam to be expanded in a back-pressure steam turbine. The boiler feed water from the deaerator enters the HRSG at around 105°C, while steam is sent at the steam turbine inlet at 170 bar and 555°C. The typical 25°C approach temperature between steam and exhaust gas temperature is considered to have an adequate heat transfer coefficient and limit the coils surface.

Steam is expanded in a back pressure steam turbine with an intermediate extraction at around 45 bar and discharge at around 21 bar. Steam from the controlled extraction is injected in the combustion chamber and in the first stage of the turbine, while the exhaust steam is used as cooling stream in lower pressure stages.

The cooling flowrate is set to control the gas turbine blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st stator	860°C
1st rotor	830°C
from 2nd stator	830°C
from 2nd rotor	800°C

Recovering steam from the HRSG, in addition to the power generation in the steam turbine, enhances the plant efficiency. In fact, acting as cooling medium, it allows reducing the amount of flue gas to be recycled and consequently the compressor power consumption.

2.2.3. *Flue gas compression and low pressure steam cycle*

A fraction (around 50%) of the flue gas from the HRSG, not recycled back to the combustion chamber, is compressed in a two-stages compressor. Heat available from compression is used for low pressure steam generation. Final cooling is provided with cooling water. Net CO_2-rich product is sent to the CPU, while water separated in the compression process is sent to the WWT for treatment and recovery.

Condensate from the condenser is pumped and preheated against flue gas from the second compressor stage. After being degassed using steam extracted from the steam turbine, the BFW is pre-heated against flue gas from the second compressor stage. Steam is generated against both the flue gas from first and second compressor stage, mixed and sent to a steam turbine, condensing type.

2.2.4. *Heat integration*

The oxy-fuel cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular heat available at high temperature level from the oxy-turbine cycle is used to provide heat required in the CPU, mainly for TSA regenerator heating and inert gas heating before expansion, while heat available at low temperature level in the CO_2 compressor intercoolers is used for oxygen and natural gas heating to enhance gas turbine efficiency.

The following interfaces have been considered:

- Natural gas is heated using as heating medium compressed CO_2 from the final compression before being sent to plant B.L.
- Oxygen is heated using as heating medium raw flue gas from the first compression stage (1-15 bar) in the CPU.

- Heat downstream the first stage of the recycle gas compressor is used as heating medium in the TSA regenerator and inerts gas heaters of the CPU.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 50 bar. Oxygen pressure is set by the requirement of the gas turbine combustor.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric requirement, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- Raw flue gas compression (1 - 34 bar);
- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

Basin diameter	120 m
Cooling tower height	210 m
Water inlet height	17 m

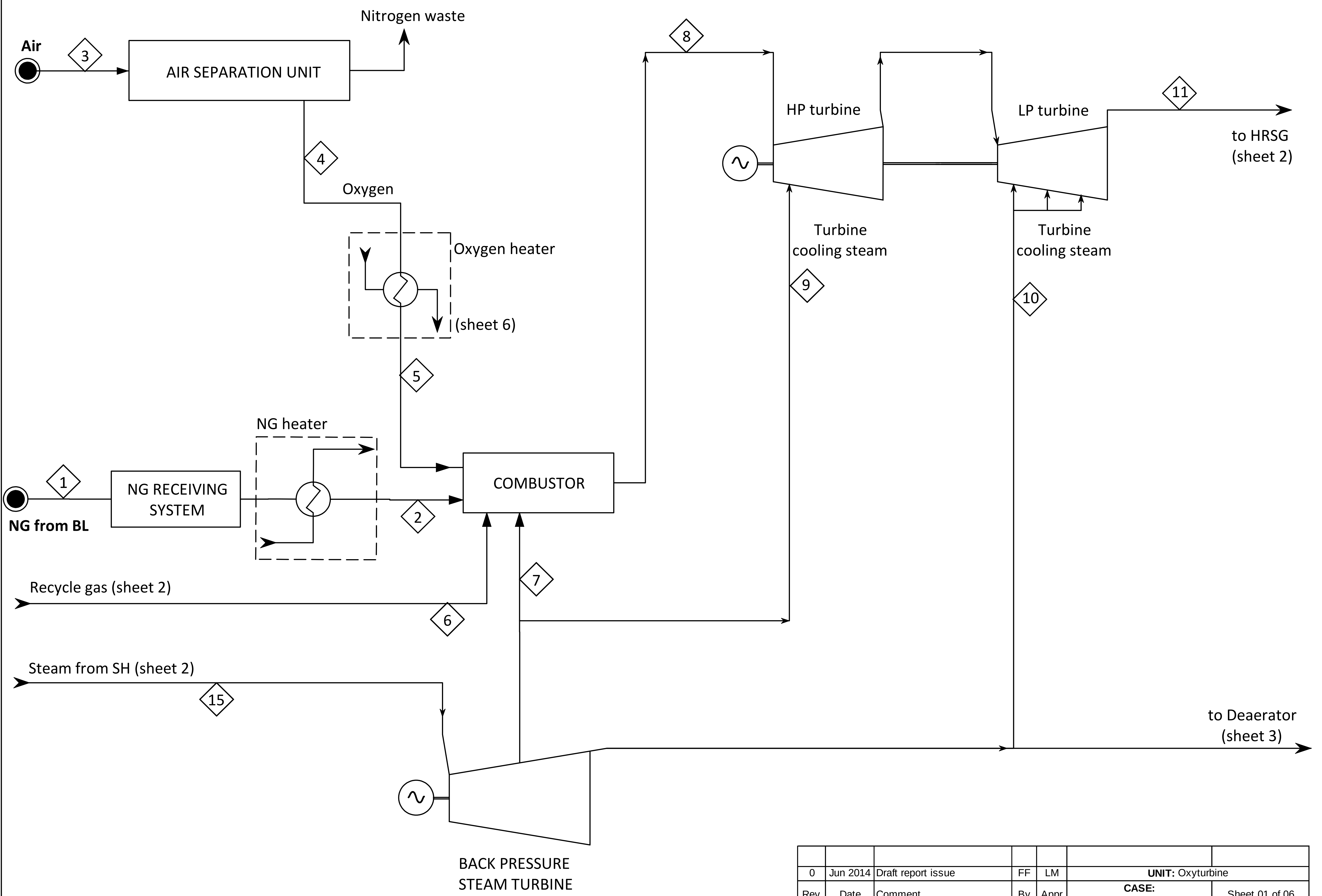
- Natural gas receiving station;
- Raw water system;
- Demineralised water plant;

- Waste Water Treatment
- Firefighting system;
- Instrument and Plant air.

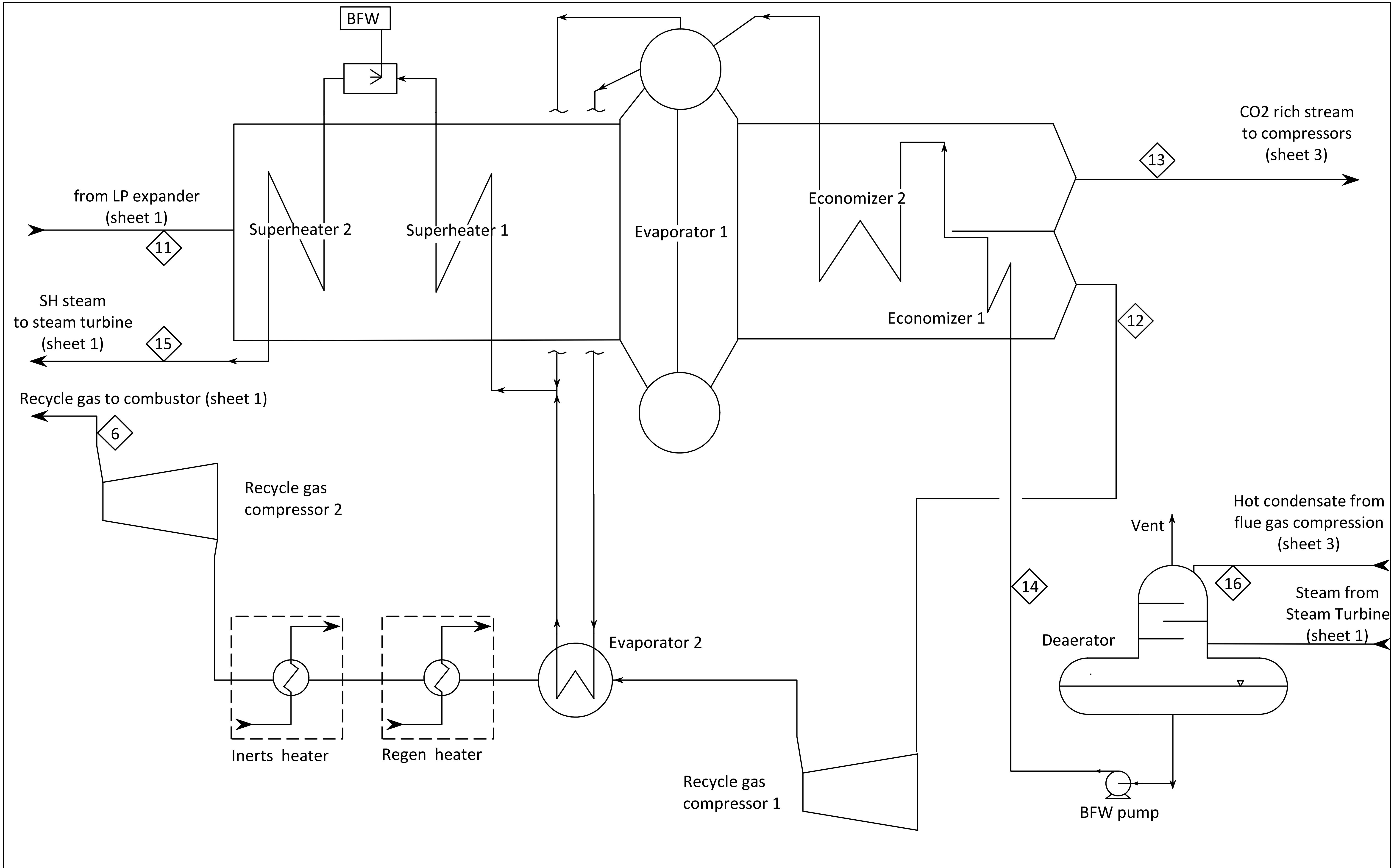
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

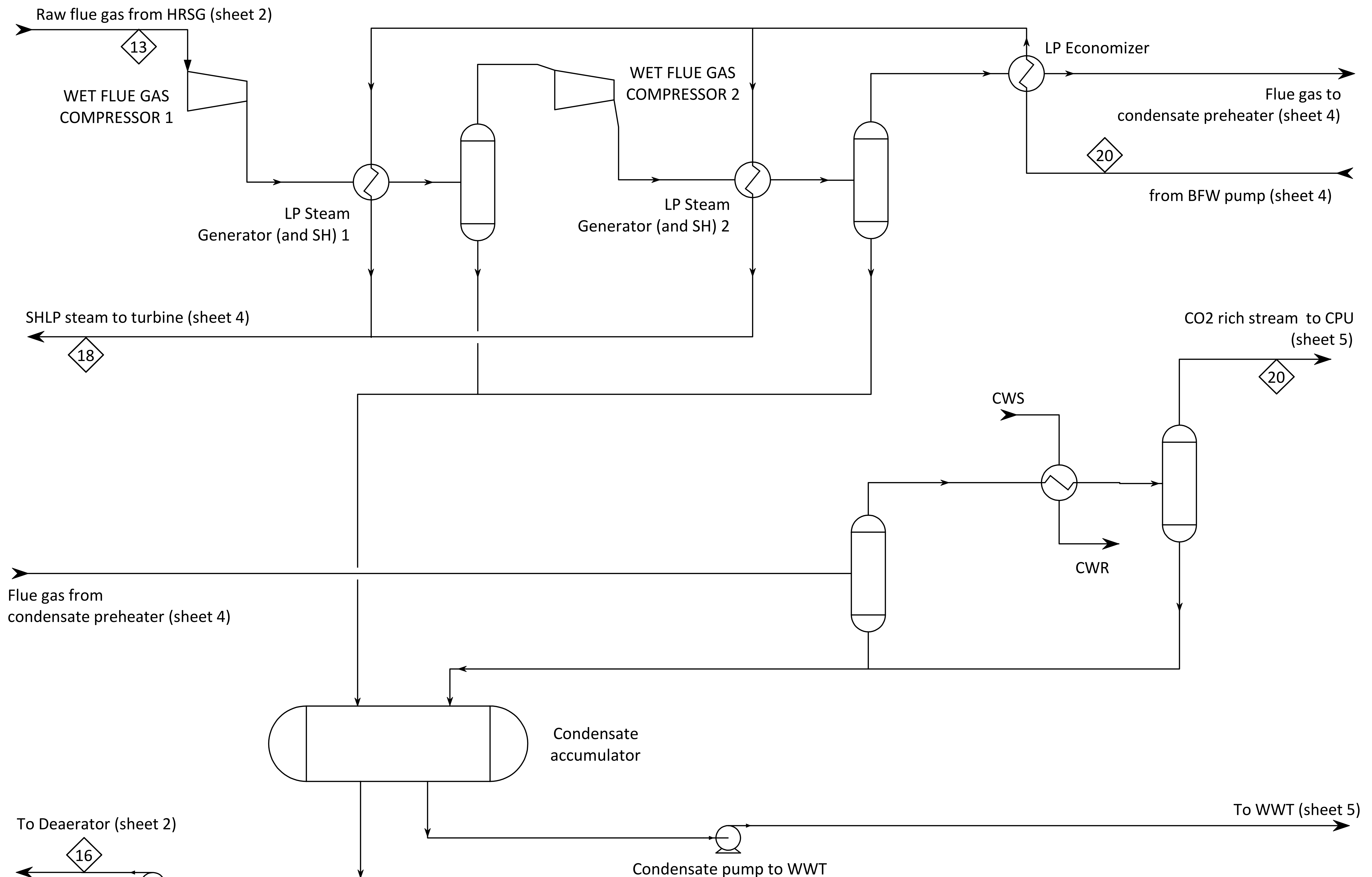
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



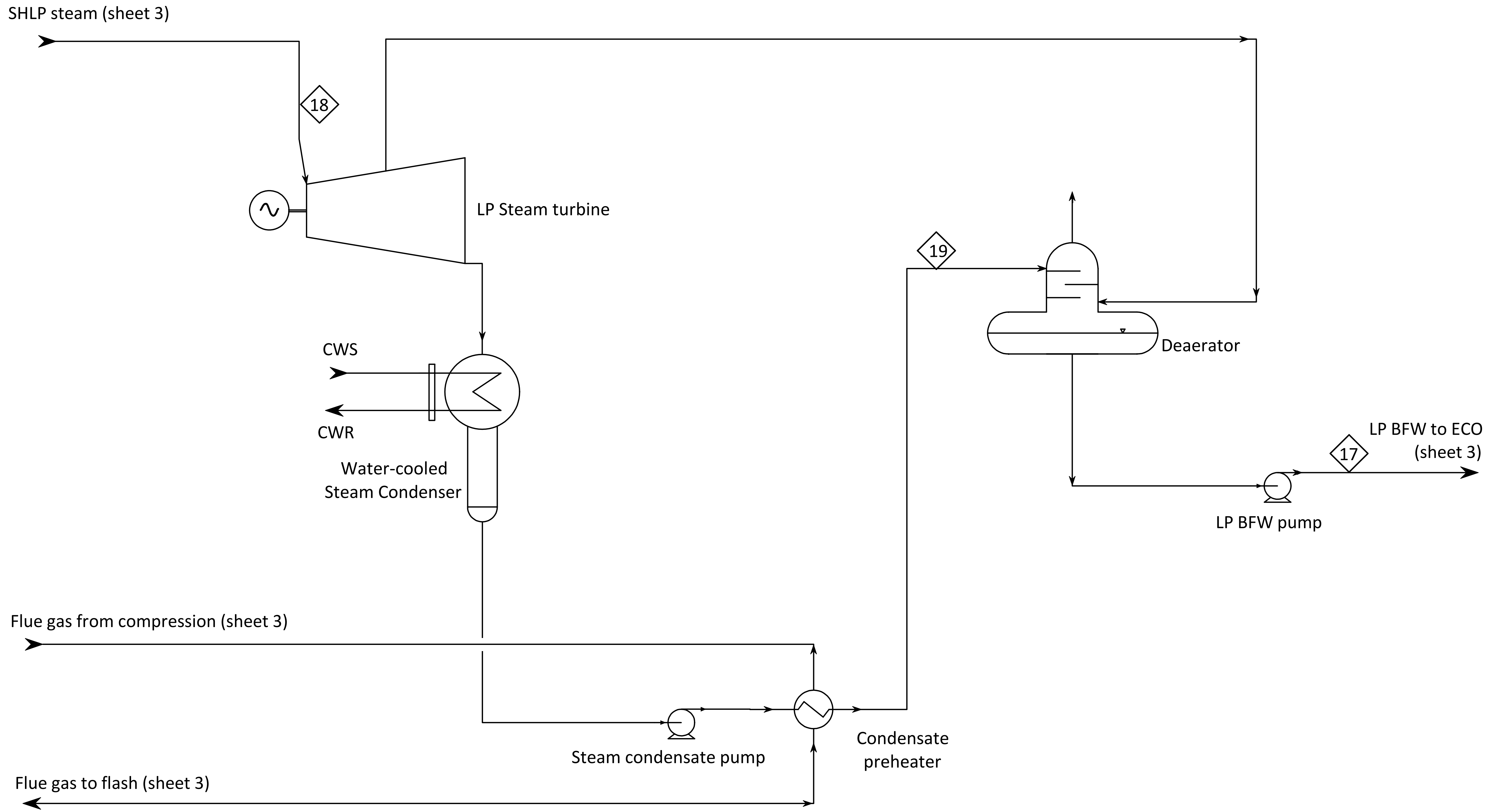
0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Oxyturbine	
Rev.	Date	Comment	By	Appr	**CASE:** 3b - modified S-GRAZ CYCLE	Sheet 01 of 06



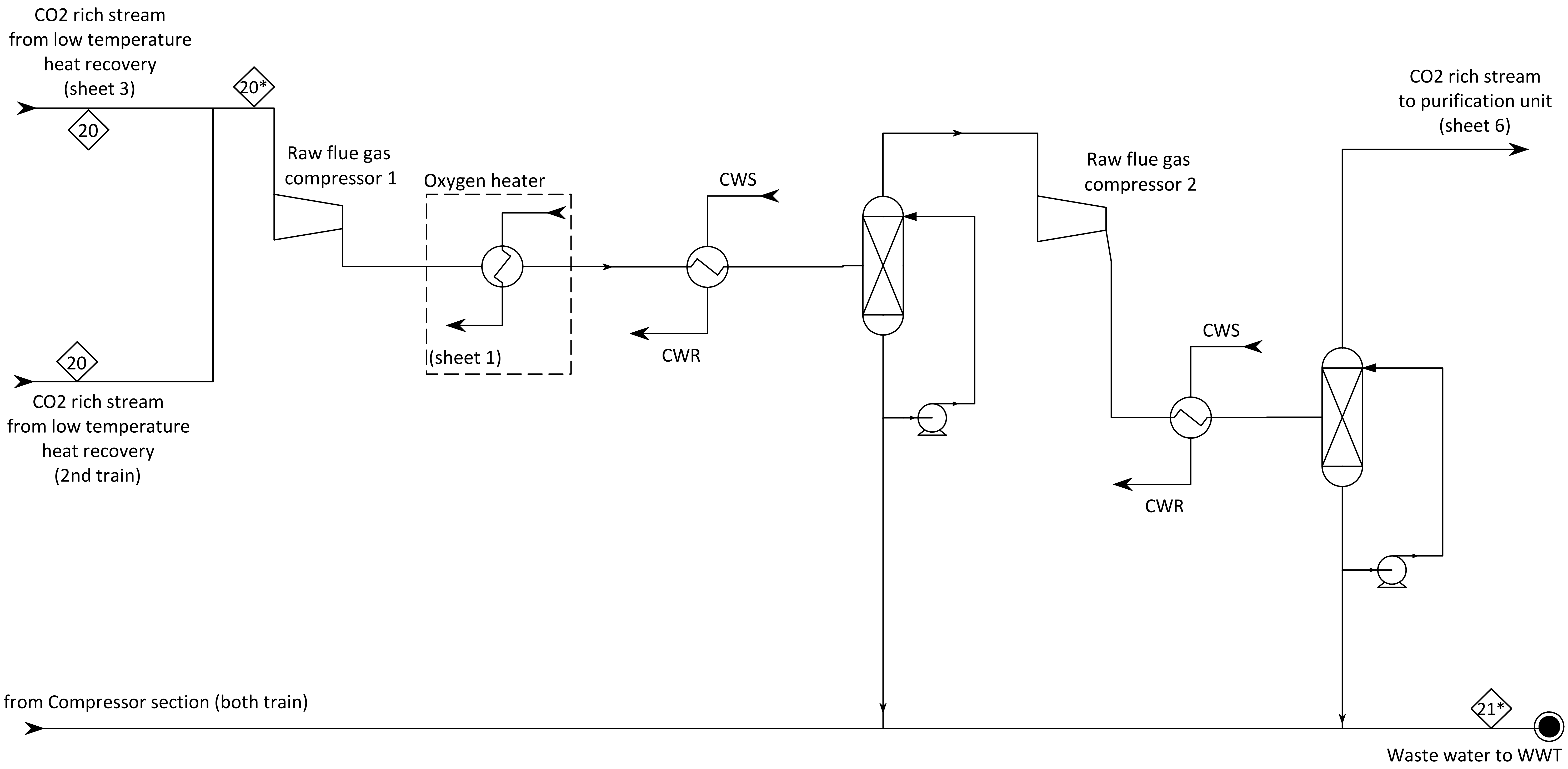
0	Jun 2014	Draft report issue	FF	LM	UNIT: HRSG and recycle gas compressors	
Rev.	Date	Comment	By	Appr	CASE: 3b - modified S-GRAZ CYCLE	Sheet 02 of 06



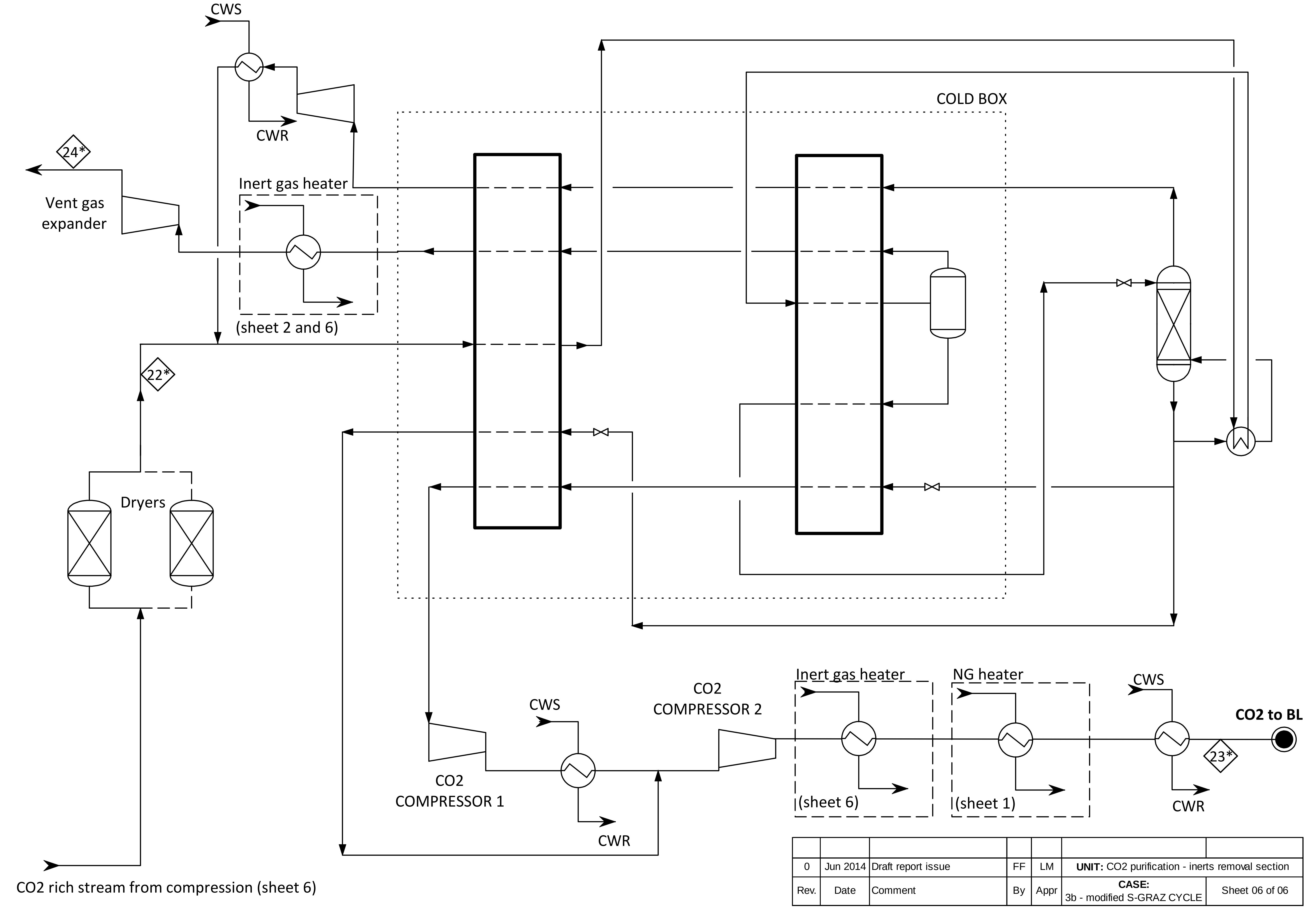
0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Wet flue gas compression	
Rev.	Date	Comment	By	Appr	**CASE:** 3b - modified S-GRAZ CYCLE	Sheet 03 of 06



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Low temperature steam cycle	
Rev.	Date	Comment	By	Appr	**CASE:** 3b - modified S-GRAZ CYCLE	Sheet 04 of 06



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** CO2 purification - compression section	
Rev.	Date	Comment	By	Appr	**CASE:** 3b - modified S-GRAZ CYCLE	Sheet 05 of 06



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** 3b - modified S-GRAZ CYCLE	Sheet 06 of 06

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

	Case 3b - modified S-GRAZ - HEAT AND MATERIAL BALANCE		REVISION	0		
	CLIENT :	IEAGHG	**PREP.**	FF		
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	NF		
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM		
	LOCATION:	The Netherlands	**DATE**	June 2014		

HEAT AND MATERIAL BALANCE
UNIT 3000 - OXY TURBINE

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	Heated NG to combustor	Air to ASU	Oxygen from ASU	Oxygen to combustor	Recycle gas	Cooling stream to combustor	Flue gas to HP turbine	Cooling stream to HP turbine	Cooling stream to LP turbine
Temperature (°C)	15	100	9	15	140	538	350	1533	350	251
Pressure (bar)	70	45.8	amb	46.7	45.8	45.8	46.2	44.5	46.2	21.5
TOTAL FLOW										
Mass flow (kg/h)	59,470	59,470	1,007,045	230,770	230,770	752,500	127,915	1,171,130	161,330	148,360
Molar flow (kmol/h)	3,300	3,300	34,895	7,185	7,185	36,350	7,100	54,105	8,955	8,235
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	59,470	59,470	1,007,045	230,770	230,770	752,500	127,915	1,171,130	161,330	148,360
Molar flow (kmol/h)	3,300	3,300	34,895	7,185	7,185	36,350	7,100	54,105	8,955	8,235
Molecular Weight (kg/kmol)	18.0	18.0	28.9	32.1	32.1	20.7	18.0	21.6	18.0	18.0
Composition (%mol)	as assigned	as assigned								
Ar			0.92%	2.00%	2.00%	0.39%	-	0.53%	-	-
CO_2			0.04%	0.00%	0.00%	9.74%	-	13.16%	-	-
H_2O			0.97%	0.00%	0.00%	89.29%	100.00%	85.53%	100.00%	100.00%
N_2			77.32%	1.00%	1.00%	0.28%	-	0.37%	-	-
O_2			20.75%	97.00%	97.00%	0.30%	-	0.41%	-	-
Total			100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 3b - modified S-GRAZ - HEAT AND MATERIAL BALANCE		REVISION	0		
	CLIENT :	IEAGHG	**PREP.**	FF		
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	NF		
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM		
	LOCATION:	The Netherlands	**DATE**	June 2014		

HEAT AND MATERIAL BALANCE

UNIT 3000 - OXY TURBINE and LOW PRESSURE STEAM CYCLE

STREAM	11	12	13	14	15	16	17	18	19	20
	Flue gas to HRSG	Flue gas recyle	Wet flue gas to compressors	BFW to HRSG	Superheated steam to back-pressure ST	Flue gas condensate to deaerator	LP BFW to ECO	SH LP steam to steam turbine	Steam condensate to deaerator	CO2 rich stream to CPU
Temperature (°C)	580	194	194	106	555	96	47	170	40	28
Pressure (bar)	1.05	1.05	1.05	195	170	2.5	0.81	0.75	0.11	1.95
TOTAL FLOW										
Mass flow (kg/h)	1,480,770	736,310	744,460	443,460	443,460	454,000	548,090	548,090	541,940	171,060
Molar flow (kmol/h)	71,290	35,450	35,840	24,615	24,615	25,200	30,424	30,424	30,080	4,016
LIQUID PHASE										
Mass flow (kg/h)				443,460		454,000	548,090		541,940	
GASEOUS PHASE										
Mass flow (kg/h)	1,480,770	736,310	744,460		443,460			548,090		171,060
Molar flow (kmol/h)	71,290	35,450.00	35,840.00		24,615			30,424		4,016
Molecular Weight (kg/kmol)	20.8	20.8	20.8		18.0			18.0		42.6
Composition (%mol)										
Ar	0.40%	0.40%	0.40%	-	-	-	-	-	-	3.59%
CO_2	9.98%	9.98%	9.98%	-	-	-	-	-	-	89.09%
H_2O	89.02%	89.02%	89.02%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	2.03%
N_2	0.28%	0.28%	0.28%	-	-	-	-	-	-	2.52%
O_2	0.31%	0.31%	0.31%	-	-	-	-	-	-	2.77%
Total	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE

1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 3b - modified S-GRAZ - HEAT AND MATERIAL BALANCE		REVISION	0		
	CLIENT :	IEAGHG	**PREP.**	FF		
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	NF		
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM		
	LOCATION:	The Netherlands	**DATE**	June 2014		

HEAT AND MATERIAL BALANCE
UNIT 4000 - CPU

STREAM	**21***	**22***	**23***	**24***						
	Waste water to WWT	CO2 rich stream to purification unit	Purified CO2 to BL	Vent gas from CPU						
Temperature (°C)	25	28	30	82						
Pressure (bar)	2.5	33.0	110	1.1						
TOTAL FLOW										
Mass flow (kg/h)	241,390	339,060	283,850	55,210						
Molar flow (kmol/h)	13,400	7,865	6,450	1,415						
LIQUID PHASE										
Mass flow (kg/h)	241,390		283,850							
GASEOUS PHASE										
Mass flow (kg/h)		339,060		55,210						
Molar flow (kmol/h)		7,865		1,415						
Molecular Weight (kg/kmol)		43.1		39.0						
Composition (%mol)										
Ar	0.00%	3.66%	0.19%	19.51%						
CO_2	0.00%	90.94%	99.80%	50.52%						
H_2O	100.00%	0.00%	0.00%	0.00%						
N_2	0.00%	2.58%	0.00%	14.33%						
O_2	0.00%	2.82%	0.01%	15.64%						
Total	100.00%	100.00%	100.00%	100.00%						

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 3b – Water consumption summary

CLIENT: IEAGHG		**REVISION**	0
PROJECT NAME: Oxy-turbine power plant		**DATE**	May-14
PROJECT No. : 1-BD-0764A		**MADE BY**	NF
LOCATION : The Netherlands		**APPROVED BY**	LM

Case 3b - Modified S-GRAZ cycle

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water 2° syst. [DT = 11°C] [t/h]
3000	**OXY-TURBINE CYCLE**			
	Condenser			**54,370**
	Turbine and generator Auxiliaries		**5**	**4,120**
	Condenser vent compressor after cooler			**1,930**
5000	**AIR SEPARATION UNIT**			
	MAC intercoolers			**9,070**
	BAC intercoolers			**850**
4000	**CO_2 PURIFICATION UNIT**			
	CO2 purification unit			**2,600**
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	**1,310**		
	Demineralized water unit	**8**	**-5**	
	Waste Water Treatment and Condensate Recovery	**-230**		
	Balance of plant			
	BALANCE	**1,088**	**0**	**72,940**

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 3b – Electrical consumption summary

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	May-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 3b - Modified S-GRAZ cycle

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**OXY-TURBINE CYCLE**	
	Condensate and recycle water system	**6,530**
	Flue gas compression	**34,320**
	Turbine Auxiliaries + generator losses	**5,480**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**128,000**
	Booster air compressor and miscellanea	**19,100**
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	**22,440**
	Autorefrigerated inerts removal unit compression consumption	**13,920**
	Autorefrigerated inerts removal unit expander production	**-2,900**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**9,130**
	Balance of plant	**1,460**
	BALANCE	**237,480**

6. Overall performance

The following table shows the overall performance of Case 3b, including CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	May-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 3b - Modified S-GRAZ cycle

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
HTT turbine power output	MWe	1206.1
HPT turbine power output	MWe	95.5
LPT turbine power output	MWe	117.4
Turbine recycle gas compressors	MWe	-423.9
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**995.2**
Oxy-turbine cycle	MWe	46.3
Air separation unit	MWe	147.1
CO2 purification and compression unit	MWe	33.5
Utility & Offsite Units	MWe	10.6
ELECTRIC POWER CONSUMPTION	**MWe**	**237.5**
NET ELECTRIC POWER OUTPUT	**MWe**	**757.7**
(Step Up transformer efficiency = 0.997%) (B)	MWe	755.5
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**64.8%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.2%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**58.5%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**44.6%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6449
CO_2 removal efficiency	**%**	**90.1**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**
CO_2 emission per net power production	**kg/MWh**	**41.5**

6.1. Comparison with literature performance data

Scope of this section is to compare the performance data of this Case 3b with those reported in the public domain, particularly with data of the paper presented at the ASME Turbo Expo 2008 ([1]), relevant to a plant with the same configuration of the present case.

The following Table 4 summarises the cycle performance, in terms of electrical efficiency for:

- This study Case 3b, considering the configuration and the design basis, in terms of combustion outlet temperature, turbine inlet temperature, natural gas composition, 97% oxygen purity and especially turbine cooling flow requirements, as described in previous sections.
- Literature data, published by the GRAZ university.

For each of the above listed conditions, main design features potentially affecting the performance are also highlighted.

Table 4. Case 3b – Performance comparison

	Case 3a	**Graz cycle (as per ASME Turbo Expo 2005, [1])**
Design parameters		
Condensation pressure	4.0 kPa	2.1 kPa
CO_2 purification	YES (O2 content: 100 ppm)	NO
CO_2 compression	110 barg	100 barg
Performance		
Gross electrical efficiency (1)	62.5%	67.6%
Cycle net electrical efficiency (2)	61.1%	65.7%
Plant net electrical efficiency (3)	49.2%	54.1%

Notes

(1) Calculated considering net power production from the turbines, subtracting total compression power demand.

(2) Including power cycle auxiliaries and utility consumptions, no CPU and ASU included

(3) Including CO_2 compression and ASU consumptions. Only CO_2 compression is considered in the reference case, while CPU only foreseen in the study case to meet CO_2 purity specification (i.e. oxygen content)

[1] H. Jericha, W.Sanz, E. Göttlich, F. Neumayer, *Design details of a 600 MW Graz cycle power plant for CO_2 capture*, Proceeding of ASME Turbo Expo 2008, June 9-13, 2008, Berlin (D)

The efficiency loss of Case 3b with respect to the reference literature case is around five (5) percentage points, mainly due to the lower gross power production. The following considerations can be made in order to explain this difference:

- The condensation pressure is lower in the case presented at the ASME Turbo conference, leading to around two-three percentage point difference. Reference shall also be made to section 6.2.1, where sensitivity to the condensation pressure is further assessed.
- A lower efficiency is considered for the low pressure steam cycle of Case 3b, due to the very low steam turbine inlet pressure and temperature.
- The CO_2 purification requires around 25-30% addition power consumption with respect to the power demand of the CO_2 compression alone, which accounts for around 0.5 percentage point efficiency difference.
- The cooling stream to the medium pressure stage of the gas turbine is higher in the study Case 3b with respect to the reference literature case; consequently the efficiency is lower. Performance data obtained by considering the same turbine efficiency and cooling stream flowrate are summarized in the following Table 5, showing that the related efficiency loss is around the residual three percentage points.

Table 5. Case 3b – Performance comparison (same gas turbine)

	Case 3b (same gas turbine as reference case)	**Graz cycle (as per ASME Turbo Expo 2008)**
Performance		
Gross electrical efficiency	64.0%	67.6%
Cycle net electrical efficiency	62.6%	65.7%
Plant net electrical efficiency	50.9%	54.1%

6.2. Performance sensitivity to key design parameters

In addition to the base case, the plant performance has been evaluated after modification of some of its key design parameters. The following Table 6 summarises the sensitivity cases selected for the Modified S-Graz cycle, highlighting the key features and performance parameters affected by each modification.

Table 6. Modified S-Graz cycle – Sensitivity cases

Case 3b – Sensitivity to ambient temperature		
Base case figure	Sensitivity	Impact
T_{ref} = 9°C	T = 25°C	• Cooling water temperature: 30-41°C • Condenser operating conditions: 44°C, 9.2 kPa • Higher ASU and CPU compressor consumptions • Lower power generation from the steam turbine
Case 3b – Sensitivity to fuel nitrogen content		
Base case figure	Sensitivity	Impact
Mole fraction ref = 0.89%	Mole fraction = 14%	• High raw gas flowrate to the CPU, leading to higher compressors consumptions and expander production • Minor impact on recycled flue gas composition, as nitrogen content is negligible with respect to water content
Case 3b – Sensitivity to oxygen purity		
Base case figure	Sensitivity	Impact
O_2 purity ref: 97%mol	O_2 purity 95%mol	• Lower ASU consumptions • High raw gas flowrate to the CPU, leading to higher compressors consumptions and expander production • Minor impact on recycled flue gas composition
O_2 purity ref: 97%mol	O_2 purity 99.5%mol	• Higher ASU consumptions • Low raw gas flowrate to the CPU, leading to lower compressors consumptions and expander production • Minor impact on recycled flue gas composition

6.2.1. *Sensitivity to ambient temperature*

Table 7 shows the impacts on key design parameters and performance when ambient temperature varies from reference (9°C) to a higher value (25°C).

Table 7. Modified S-Graz cycle – Sensitivity to ambient temperature

			BASE CASE	SENSITIVITY CASE
SENSITIVITY				
Ambient temperature		**°C**	**9**	**25**
DESIGN FEATURES				
Cooling water temperature	Supply	°C	15	30
	Return	°C	26	41
Condensing temperature		°C	29	44
Condensing pressure		kPa	4.0	9.1
PERFORMANCES COMPARISON				
Natural Gas flow rate (A.R.)		t/h	118.9	118.9
Natural Gas LHV		kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)		**MWth**	**1536**	**1536**
HTT turbine power output		MWe	1206.1	1206.1
HPT turbine power output		MWe	95.5	95.5
LPT turbine power output		MWe	117.4	89.4
Turbine recycle gas compressors		MWe	-423.9	-423.9
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)		**MWe**	**995.2**	**967.2**
Oxy-turbine cycle		MWe	46.3	46.3
Air separation unit		MWe	147.1	153.9
CO2 purification and compression unit		MWe	33.5	35.6
Utility & Offsite Units		MWe	10.6	11.0
ELECTRIC POWER CONSUMPTION		**MWe**	**237.5**	**246.8**
NET ELECTRIC POWER OUTPUT		**MWe**	**757.7**	**720.5**
(Step Up transformer efficiency = 0.997%) (B)		MWe	755.5	718.3
Gross electrical efficiency (C/A x 100) (based on LHV)		**%**	**64.8%**	**63.0%**
Net electrical efficiency (B/A x 100) (based on LHV)		**%**	**49.2%**	**46.8%**
Fuel Consumption per net power production		**MWth/MWe**	**2.03**	**2.14**
CO_2 emission per net power production		**kg/MWh**	**41.5**	**43.6**

Based on the above data, the following considerations can be drawn:

- Gross power production is reduced by around two (2) percentage points, due to the lower power production of the steam turbine.
- Net power production is also affected by the higher compressor power demand, both in the ASU and in the CPU, due to the higher cooling level and consequently the higher compressor inlet temperature downstream each intercooling stage. Resulting penalty is more than two (2) percentage points on the net electrical efficiency.

As a more general consideration, it has to be noted that the impact of hot ambient conditions is less significant in the semi-closed oxy-fuel cycle as the gas turbine performance are not (or less) affected by ambient temperature, which only affects the performance of the equipment impacted by the achievable heat rejection value (i.e. cooling medium temperature level) such as the steam turbine and the intercooled compressors.

6.2.2. *Sensitivity to fuel nitrogen content*

Table 8 shows the impacts on key design parameters and performance when the inert (nitrogen) content in the fuel to the gas turbine increases from reference (0.9%mol) to 14%mol.

Table 8. Modified S-Graz cycle – Sensitivity to fuel nitrogen content

			BASE CASE	SENSITIVITY CASE
SENSITIVITY				
Fuel nitrogen content		**%mol**	**0.89%**	**14%**
DESIGN FEATURES				
Raw flue gas to CPU		kmol/h	8032	9052
Inert gas		kmol/h	1594	2415
PERFORMANCES COMPARISON				
Natural Gas flow rate (A.R.)		t/h	118.9	147.1
Natural Gas LHV		kJ/kg	46502	37599
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)		**MWth**	**1536**	**1536**
HTT turbine power output		MWe	1206.1	1205.7
HPT turbine power output		MWe	95.5	95.0
LPT turbine power output		MWe	117.4	115.3
Turbine recycle gas compressors		MWe	-423.9	-419.0
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)		**MWe**	**995.2**	**997.1**
Oxy-turbine cycle		MWe	46.3	48.3
Air separation unit		MWe	147.1	147.1
CO2 purification and compression unit	Compressor	MWe	36.4	40.3
	Expander	MWe	-2.9	-4.9
Utility & Offsite Units		MWe	10.6	10.6
ELECTRIC POWER CONSUMPTION		**MWe**	**237.5**	**246.8**
NET ELECTRIC POWER OUTPUT		**MWe**	**757.7**	**755.6**
(Step Up transformer efficiency = 0.997%) (B)		MWe	755.5	753.3
Gross electrical efficiency (C/A x 100) (based on LHV)		**%**	**64.8%**	**64.9%**
Net electrical efficiency (B/A x 100) (based on LHV)		**%**	**49.2%**	**49.0%**
Fuel Consumption per net power production		**MWth/MWe**	**2.03**	**2.04**
CO_2 emission per net power production		**kg/MWh**	**41.5**	**41.7**

Based on the above data, the following considerations can be drawn:

- The main design parameter related to higher inert (nitrogen) content in the fuel is the higher flowrate entering the CPU, resulting in larger compressor and inert gas expander and related power demand/production.
- Difference in the gross power production is related to the lower recycle compressor consumption. A lower recycled flowrate is required since a moderated cooling effect is associated to the higher natural gas flowrate.
- Resulting net power production is slightly lower than the base case (0.2 percentage point), due to the higher CPU net power demand.

6.2.3. *Sensitivity to oxygen purity*

Table 9 shows the impacts on key design parameters and plant performance when the purity of the oxygen from the ASU varies from reference (97%mol) to higher and lower values (respectively 99.5%mol and 95%mol).

Table 9. Modified S-Graz cycle – Sensitivity to oxygen purity

			BASE CASE	SENSITIVITY CASE (lower purity)	SENSITIVITY CASE (higher purity)
SENSITIVITY					
Oxygen purity		**%mol**	**97%**	**95%**	**99.5%**
DESIGN FEATURES					
Raw flue gas to CPU		kmol/h	8032	8340	7666
Inert gas		kmol/h	1594	1710	1057
PERFORMANCES COMPARISON					
Natural Gas flow rate (A.R.)		t/h	118.9	118.9	118.9
Natural Gas LHV		kJ/kg	46502	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)		**MWth**	**1536**	**1536**	**1536**
HTT turbine power output		MWe	1206.1	1206.4	1204.9
HPT turbine power output		MWe	95.5	95.2	95.7
LPT turbine power output		MWe	117.4	116.8	118.1
Turbine recycle gas compressors		MWe	-423.9	-422.9	-424.2
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)		**MWe**	**995.2**	**995.6**	**994.6**
Oxy-turbine cycle		MWe	46.3	47.0	45.6
Air separation unit		MWe	147.1	146.4	153.9
CO2 purification and compression unit	Compressor	MWe	36.4	37.8	34.5
	Expander	MWe	-2.9	-3.5	-2.2
Utility & Offsite Units		MWe	10.6	10.6	10.6
ELECTRIC POWER CONSUMPTION		**MWe**	**237.5**	**238.3**	**242.4**
NET ELECTRIC POWER OUTPUT		**MWe**	**757.7**	**757.3**	**752.2**
(Step Up transformer efficiency = 0.997%) (B)		MWe	755.5	755.0	749.9
Gross electrical efficiency (C/A x 100) (based on LHV)		**%**	**64.8%**	**64.8%**	**64.7%**
Net electrical efficiency (B/A x 100) (based on LHV)		**%**	**49.2%**	**49.1%**	**48.8%**
Fuel Consumption per net power production		**MWth/MWe**	**2.03**	**2.03**	**2.05**
CO_2 emission per net power production		**kg/MWh**	**41.5**	**41.7**	**41.7**

In an oxy-combustion power plant, oxygen purity has to be selected in order to minimise the combined consumption of the ASU and the CPU. In fact, the higher the oxygen purity, the higher the ASU compressor consumptions but the lower the CPU power demand, as a lower flowrate has to be treated by this unit.

Based on the above data, the following considerations can be drawn:

- Increasing oxygen purity from 95%mol to 97%mol results in a minor power penalty in the ASU, as the distillation to produce the oxygen stream is essentially an oxygen/nitrogen separation [1] only.
- As purity is increased above 97%mol, power demand increases more considerably as the distillation changes in an oxygen/argon separation, which is characterised by a lower relative volatility.
- CPU net power demand (compression consumptions after considering expander production) increases linearly with the inert content in the oxygen from the ASU.
- As for the case with different nitrogen content in the fuel, marginal difference in the recycle compressor consumptions is related to the higher/lower inert content in the fuel, which has an impact on the recycle flowrate required for combustion temperature control.

The above considerations are graphically shown in the following Figure 2, which shows that the optimum oxygen purity in an oxy-combustion power plant based on Graz cycle is around 97%mol.

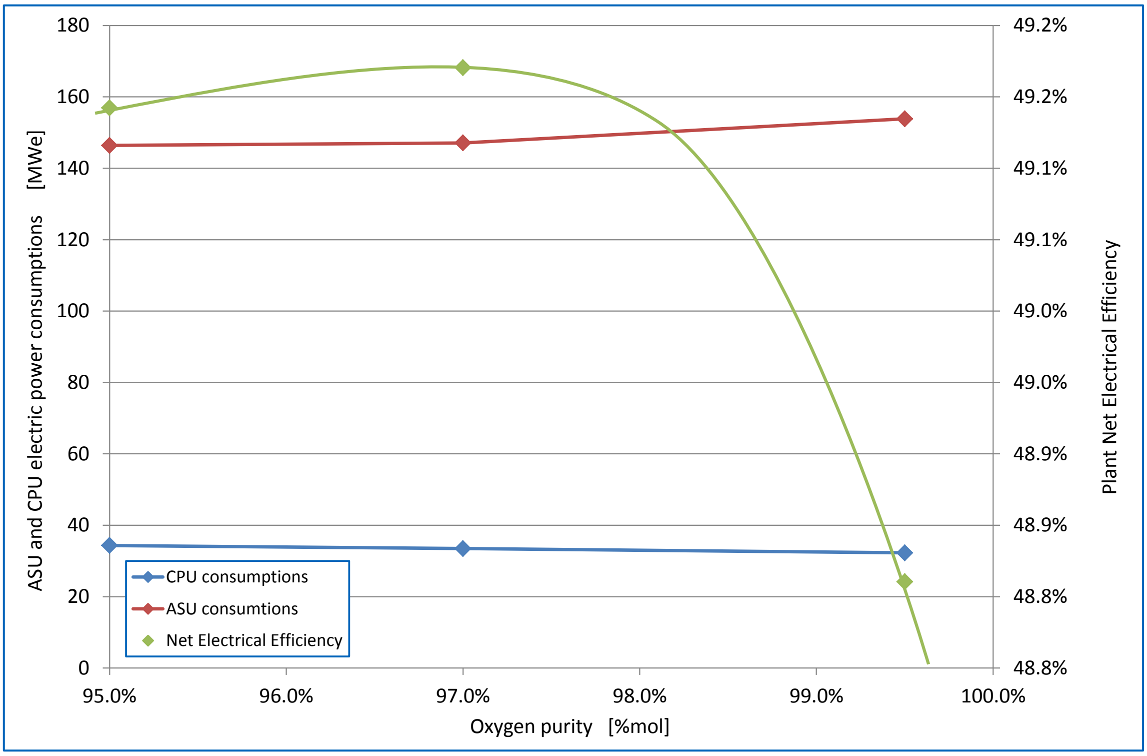


Figure 2. Modified S-GRAZ cycle – Impact of O_2 purity

[1] Simon Gibson, *Oxygen plant for gasification*, ICheme2014 conference proceeding

7. Environmental impact

The oxy-combustion gas turbine plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 10 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power island and in the CO_2 purification unit.

Table 10. Case 3b – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	55,400
Flow, Nm^3/h	31,830
Composition	(%mol)
Ar	19.50%
N_2	14.32%
O_2	15.68%
CO_2	50.50%
H_2O	0.00%
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from the flue gas compressor intercoolers and CO_2 purification unit) are treated to recover water, so the main liquid effluent is the cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 320 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3b - Modified S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	OXY TURBINE PACKAGE							
PK- 3101-1/2	**Oxy Turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
T- 3101	HP expander		131 MWe Pin: 44.5 bar; Pout: 21 bar					*Including:* *Lube oil system* *Cooling system*
T- 3102	LP expander		472 MWe Pin: 21 bar; Pout: 1.05 bar					*Idraulic control syste* *Seals system* *Drainage system*
F- 3101	Combustor		770 MWth					
K- 3101	Recycle gas compressor							
	1st stage		140 MWe					
	2nd stage		70 MWe					
G- 3101	Oxy turbine generator							*Including relevant auxiliaries*
	HEAT RECOVERY STEAM GENERATOR							
PK- 3201-1/2	**Heat recovery steam generator**	Horizontal, Natural Circulated, 1 Pressure Level						***2 x 50% HRSG package***
D- 3201	HP steam drum							
E- 3201	HP Economizer 1st section							
E- 3202	HP Economizer 2nd section							
E- 3203	HP Evaporator							
E- 3204	HP superheater							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	BFW pump	Centrifugal	450 m3/h x 2165 m	3400 kW				*One operating one spare, per each train*

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 3b - Modified S-GRAZ cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	HEAT RECOVERY STEAM GENERATOR							
	DRUM							
	Deaerator							
	Steam generator blowdown drum (continuous)							
	Intermittent blowdown drum							
	EXCHANGER							
	Blowdown cooler							
	PACKAGES (Common to both train)							
PK- 3202	**Fluid Sampling Package**							
PK- 3203	**Phosphate Injection Package**							
	Phosphate storage tank							
	Phosphate dosage pumps							*One operating one spare*
PK- 3204	**Oxygen scavenger Injection Package**							
	Oxygen scavenger storage tank							
	Oxygen scavenger dosage pumps							*One operating one spare*
PK- 3204	**Amine Injection Package**							
	Amine storage tank							
	Amine dosage pumps							*One operating one spare*
	BACK-PRESSURE STEAM TURBINE							
PK- 3301-1/2	**Back pressure steam turbine and Generator Package**							***2 x 50% package***
	Including:							
ST- 3301	Back-pressure steam turbine		48 MWe HP steam inlet: 170 bar MP steam extraction: 45 bar Exhaust pressure: 21 bar					*Including:* *Lube oil system* *Cooling* *Idraulic control system* *Seals system (including gland condenser and vacuum system)* *Drainage system*
G- 3301	Steam turbine generator		60 MVA					*Including relevant auxiliaries*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3b - Modified S-GRAZ cycle		REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2	
			DATE	Jun-14	Feb-15			
			ISSUED BY	NF	NF			
			CHECKED BY	LM	LM			
			APPROVED BY	LM	LM			

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	FLUE GAS COMPRESSION PACKAGE							
PK- 3303-1/2	**Wet flue gas compressors**							***2 x 50% compression train***
	- Wet flue gas compressor #1	axial	Flowrate: 2 x 400,000 Nm3/h Pin: 104 kPa; Pout : 127 kPa Compression ratio: 1.22	*2 x 5,100kW*				
	- Wet flue gas compressor #2	axial	Flowrate: 2 x 190,000 Nm3/h Pin: 125 kPa; Pout : 195 kPa Compression ratio: 1.56	*2 x 4,200kW*				
	Condensate separators							
	Intercoolers							
	Condensate pre-heater							
	BFW pre-heater							
	Steam generator #1							
	Steam generator #2							
	Steam superheater #1							
	Steam superheater #2							
	CW cooler							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3301 A/B	Flue gas condensate pump	Centrifugal	610 m3/h x 30 m	75 kW				*One operating one spare, per each train*
	STEAM TURBINE							
PK- 3401-1/2	**Steam turbine and Generator Package**							***2 x 50% package***
	Including:							
ST- 3401	Steam turbine		60 MWe					*Including:* *Lube oil system* *Cooling system* *Idraulic control system* *Seals system* *Drainage system*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3b - Modified S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	STEAM TURBINE							
G- 3401	Steam turbine generator		75 MVA					*Including relevant auxiliaries*
	Inlet/after condenser							
	Gland Condenser							
PK- 3402-1/2	**Steam Condenser Package**							***2 x 50% condenser package***
	Each including:							
	Steam condenser		350 MWth					*Including:*
								Condenser hotwell
								Ejector
								Start-up Ejector
PK- 3403-1/2	**Steam Turbine by-pass system**							***2 x 50% package***
	PUMPS		**Q [m3/h] x H [m]**					
P- 3401 A/B	Condensate pumps	Centrifugal	700 m3/h x 10 m	3 kW				*One operating one spare*
P- 3402 A/B	BFW pump	Centrifugal	500 m3/h x 15 m	5 kW				
	DRUM							
	Deaerator							

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 3b - Modified S-GRAZ cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 4000 - CO_2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	PACKAGES							
PK - 4001	**CO_2-rich gas compression**							
	Including:							
	Raw flue gas compressors							
	- *Raw flue gas compressor #1*	axial	Flowrate: 2 x 90,000 Nm3/h Pin: 1.9 bar; Pout : 15 bar Compression ratio: 7.9	*2 x 9 MWe*				*2x50%*
	- *Raw flue gas compressor #2*	axial	Flowrate: 2 x 98,000 Nm3/h Pin: 14.4 bar; Pout : 35 bar Compression ratio: 2.44	*2 x 3.0 MWe*				*2x50%*
	Condesate separators							
	Intercoolers							
	Oxygen heater							
	CW cooler #1							
	CW cooler #2							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO_2 liquid separator							
	- CO_2 distillation column							
	- CO_2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MWe*				*2x50%*
	- CO_2 compressor 1	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		3000 MWe					*1x100%*
	- Overhead recycle compressors (optional)							
	- Intercoolers							
	Inert gas heater							
	NG heater							
	Cooling water intercoolers							

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3b - Modified S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 5000 - Air Separation Unit (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5500 t/d					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 35.5 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		9.5 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3b - Modified S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower** including: Cooling water basin	Natural draft	940 MWth Diameter: 115 m, Height: 180 m, Water inlet: 17 m				concrete	
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/B/C/D	**Cooling Water Pumps (primary system)**	Centrifugal	13,000 x 40	1750				*Four in operation*
P- 6003 A/B	**Cooling tower make up pumps**	Centrifugal	1500 x 30	160				*One in operation, one spare*
	PACKAGES							
	Cooling Water Filtration Package Cooling Water Sidestream Filters		7700 m3/h					
	Sodium Hypochlorite Dosing Package Sodium Hypochlorite storage tank Sodium Hypochlorite dosage pumps							
	Antiscalant Package Dispersant storage tank Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
PK- 6002	Let down station							
	RAW WATER SYSTEM							
PK- 6003	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Portable water pumps	centrifugal						*One operating, one spare*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 3b - Modified S-GRAZ cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	DEMINERALIZED WATER SYSTEM							
PK- 6004	Demin Water Package, including:							
	- Multimedia filter							
	- Reverse Osmosis (RO) Cartidge filter							
	- Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : CASE 4A: CES CYCLE
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 4a of the study, which is the CES cycle based power plant, with cryogenic purification and separation of the carbon dioxide. The plant is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected CES plant configuration is based on two parallel trains, each including one F-class equivalent oxy-fired gas turbine, composed of two combustors and three expansion sections with intermediate heat recovery section and final condenser.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 4a – Unit arrangement

Unit	Description	Trains
3000	Power Island	
	NG compressor	2 x 50%
	Gas Turbine	2 x 50%
	Flue gas condenser	2 x 50%
	Wet flue gas compression	2 x 50%
	Water/steam system (including low pressure steam cycle)	2 x 50 %
4000	CO_2 purification and compression	
	Raw gas compression	2 x 50%
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power Island

The unit is mainly composed of two trains, each including:

- One F-class equivalent oxy-fired gas turbine, composed of:
  - HP combustor and expansion section (not cooled),
  - MP combustor and expansion section,
  - LP expansion section.
- One flue gas condenser.
- Wet flue gas compressor.
- Injected BFW/Steam system.
- Low pressure steam cycle.

Technical information relevant to this unit is reported in chapter D, section 2.1.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.2.1. Gas Turbine expander design features

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the stream at gas turbine boundary are summarised in the following Figure 1 and Figure 2.

The natural gas from the metering station is diverted into two streams. Around 55% of the total overall flowrate is compressed to 170 bar, and fed to the high pressure combustor of the gas turbine at around 100°C. The remaining fraction is first let down from the grid pressure to the required pressure level of the second combustor through the let-down station and then injected in the combustor at ambient temperature.

Oxygen is delivered from the ASU at the pressure level required by the second combustor (around 30 bar). The flowrate required for the combustion in the first reactor is compressed to 170 bar in a dedicated compressor and injected in the combustor at around 135°C. The remaining portion is heated up against compressed CO_2 in the CPU before being fed to the second combustor at around 150°C.

The fuel and oxidant stream to the first combustor are combined with cold boiler water and high pressure saturated steam generated in the downstream heat recovery section. All the steam generated in the power plant is injected in the combustion chamber while the boiler water flowrate is set in order to control the combustion outlet temperature at 900°C.

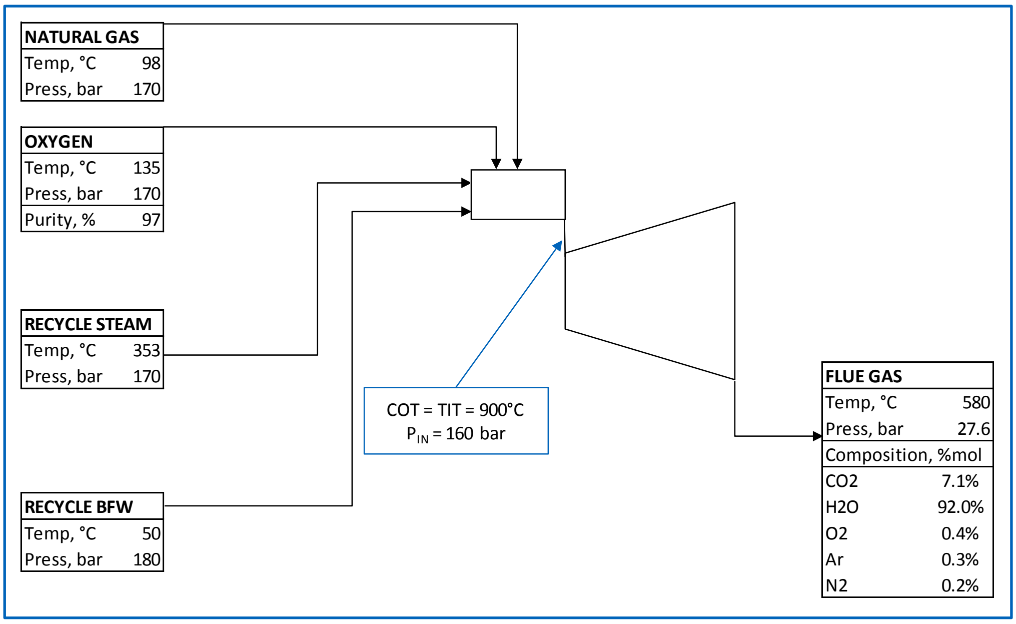


Figure 1. CES gas turbine – HP section

The HP turbine inlet conditions are set at 160 bar and 900°C. The temperature is lower than the maximum allowed for cooled gas turbine, as the first stage of the CES gas turbine is not cooled.

Heat available from the flue gas exhaust from the HP section is used for HP steam generation to be used for combustion temperature control.

Part of the cooled exhaust flue gas is fed to the second combustor, while the remaining fraction bypasses the combustor to be used for gas turbine cooling. The cooling flowrate is set to control the gas turbine blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st stator	860°C
1st rotor	830°C
from 2nd stator	830°C
from 2nd rotor	800°C

The MP turbine inlet conditions are set at 26 bar and 1533°C. Combustion outlet temperature is controlled acting on the natural gas flowrate to the second combustor. Natural gas flow to the first combustor is consequently fixed.

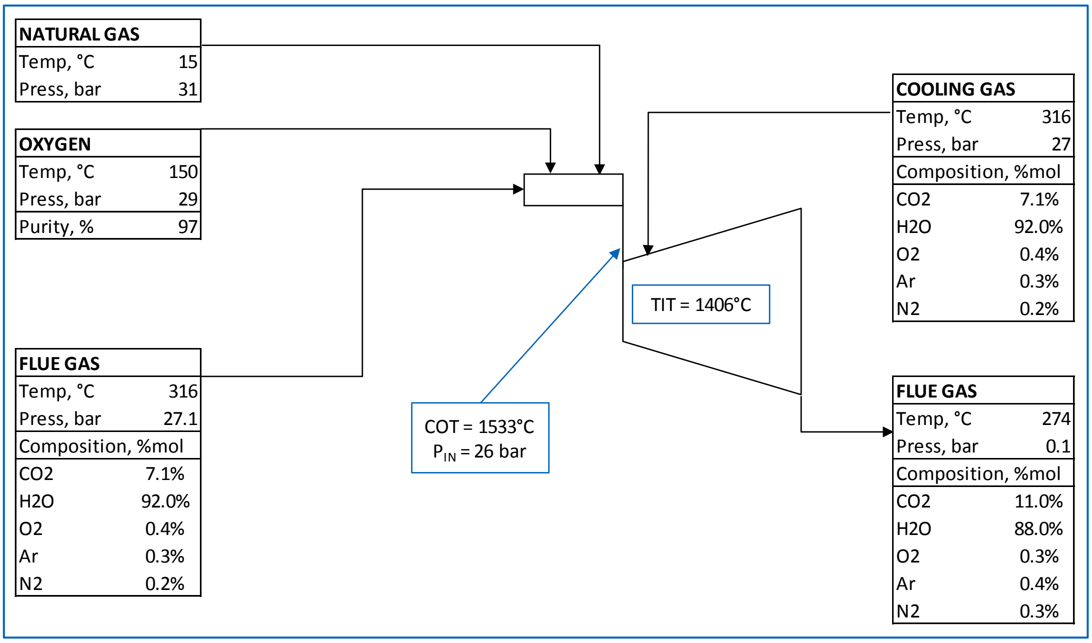


Figure 2. CES gas turbine – MP/LP section

The MP turbine section expands the flue gas down to atmospheric pressure. The final LP section expands the flue gas down to condenser pressure, which is set to 0.1 bar.

The MP turbine expander has 5 stages, so as to have acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine, at the rotational speed of 3600 RPM.

2.2.2. *Heat recovery section*

The exhaust gases from the HP turbine enter the heat recovery section at 580°C. Heat is recovered to produce high-pressure saturated steam to be injected in the HP combustor, at 170 bar and 355°C. Inert gas and regenerator preheating in the CPU is also achieved recovering heat from the flue gas.

The exhaust gases exit the LP turbine at 275°C. Before final cooling with cooling water in the condenser, heat is recovered from the flue gas generating low pressure steam to be expanded in a low pressure steam turbine, condensing type. Saturated steam from the compressor intercoolers downstream the flue gas condenser is mixed with the generated saturated steam. The whole steam flow is superheated against hot flue gas from the expander before being sent to a low pressure steam turbine.

2.2.3. *Flue gas condenser and compression section*

Cooled flue gas from the heat recovery section is cooled and partially condensed in a flue gas condenser. To be consistent with the scheme proposed in literature[1], condensing pressure is set at 0.1 bar while temperature is set at 29°C due to cooling medium conditions.

The wet flue gas from the condenser is compressed up to atmospheric pressure and sent to the CPU. Heat from wet flue gas compressor intercoolers is recovered generating low pressure steam for the condensing steam turbine.

Condensed water from the flue gas is recycled back to the condenser hotwell. Most of the flowrate is pumped to a condensate polishing section, in order to be treated and recycled back to the oxy-cycle to be used as injection water or for steam generation. The net water production from the combustion process is sent to the waste water treatment.

2.2.4. *Heat integration*

The oxy-fuel cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular heat available at high temperature level from the oxy-turbine cycle is used to provide heat required in the CPU mainly for TSA regenerator heating and inert gas heating before expansion, while heat available at low temperature level in the compressor intercoolers is used for oxygen and BFW pre-heating in order to enhance gas turbine efficiency.

The following interfaces have been considered:

- Oxygen to the second combustor is heated against compressed CO_2 from the final compression before being sent to plant B.L.
- Heat available from the flue gas exhaust from the HP turbine is used as heating medium in the TSA regenerator and inerts gas heaters of the CPU.
- Heat downstream raw gas compressor in the CPU is used for injection boiler water pre-heating.
- Heat available from the flue gas exhaust from the LP turbine and downstream the wet flue gas compressor in the oxy-cycle is used for LP steam generation.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 30 bar. Oxygen pressure is set by the requirement of the second gas turbine combustor.

A dedicated compressor is installed to compress the oxygen up to the pressure and the temperature level required by the first gas turbine combustor.

[1] DOE/NETL, 2010. *Carbon Capture Approaches for Natural Gas Combined Cycle Systems*. Report DOE/NETL-2011/1470, Revision 2, December 20, 2010.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- Raw flue gas compression (1 - 34 bar);
- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

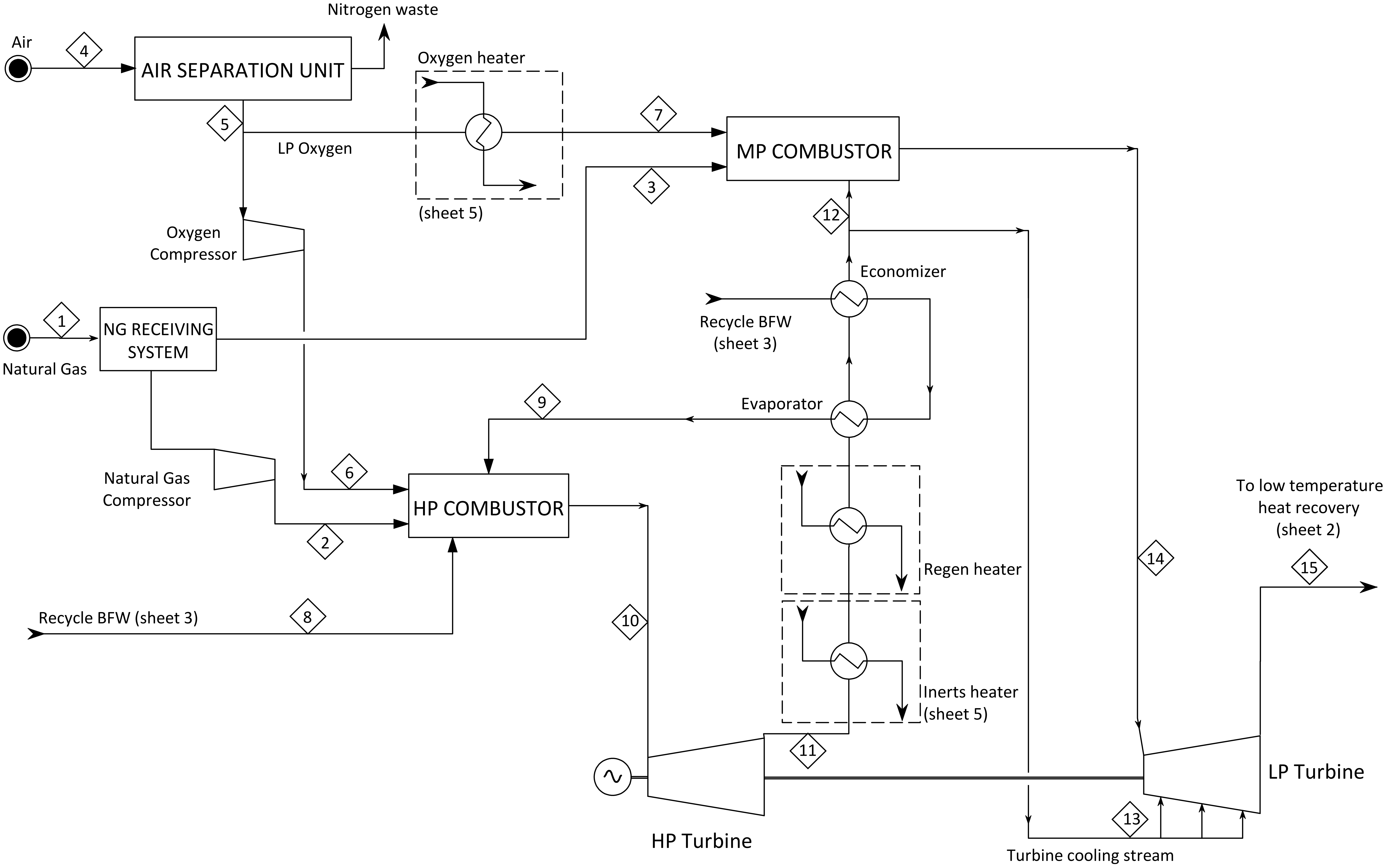
Basin diameter	120 m
Cooling tower height	180 m
Water inlet height	17 m

- Natural gas receiving station;
- Raw water system;
- Demineralised water plant;
- Waste Water Treatment
- Fire fighting system;
- Instrument and Plant air.

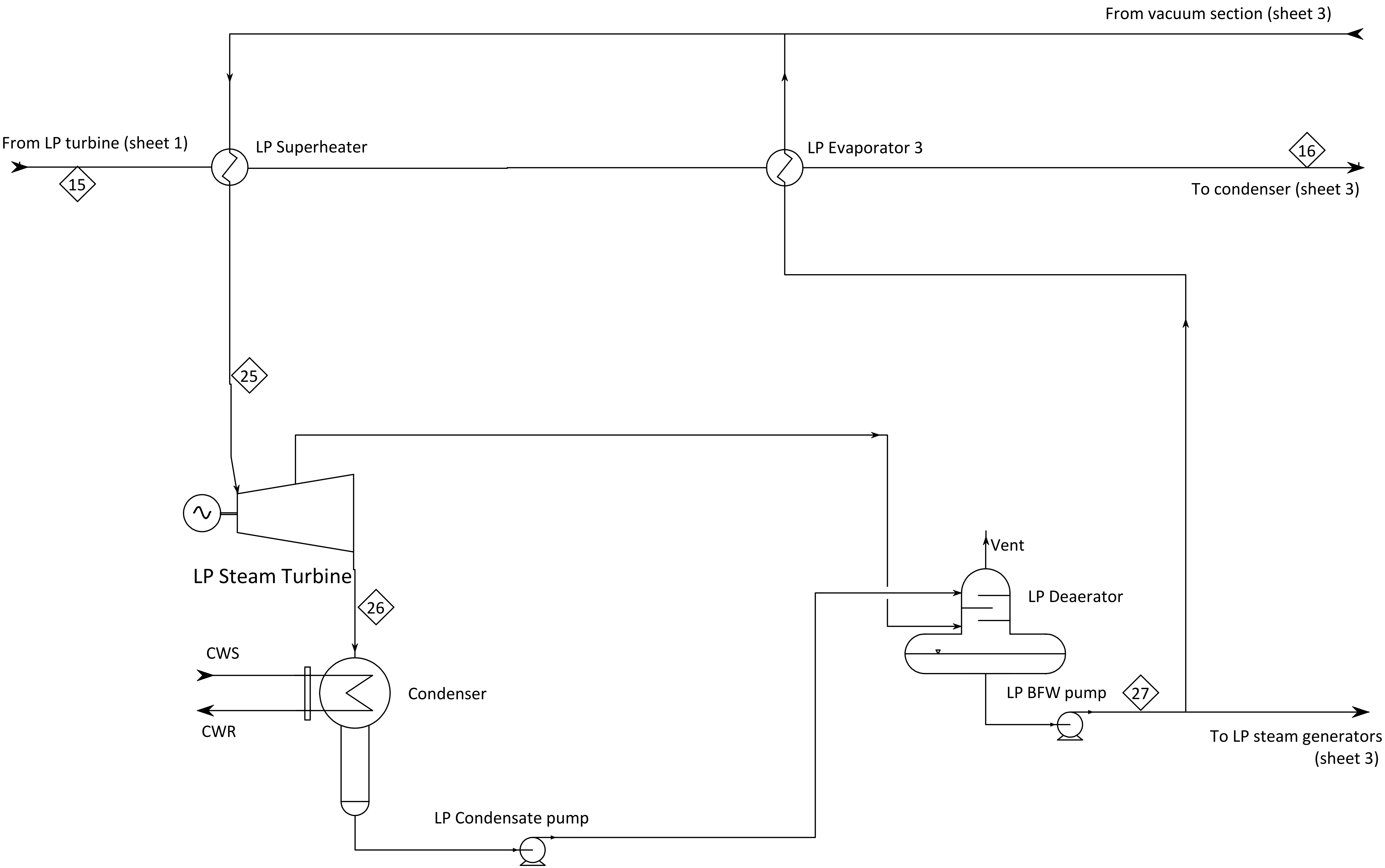
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

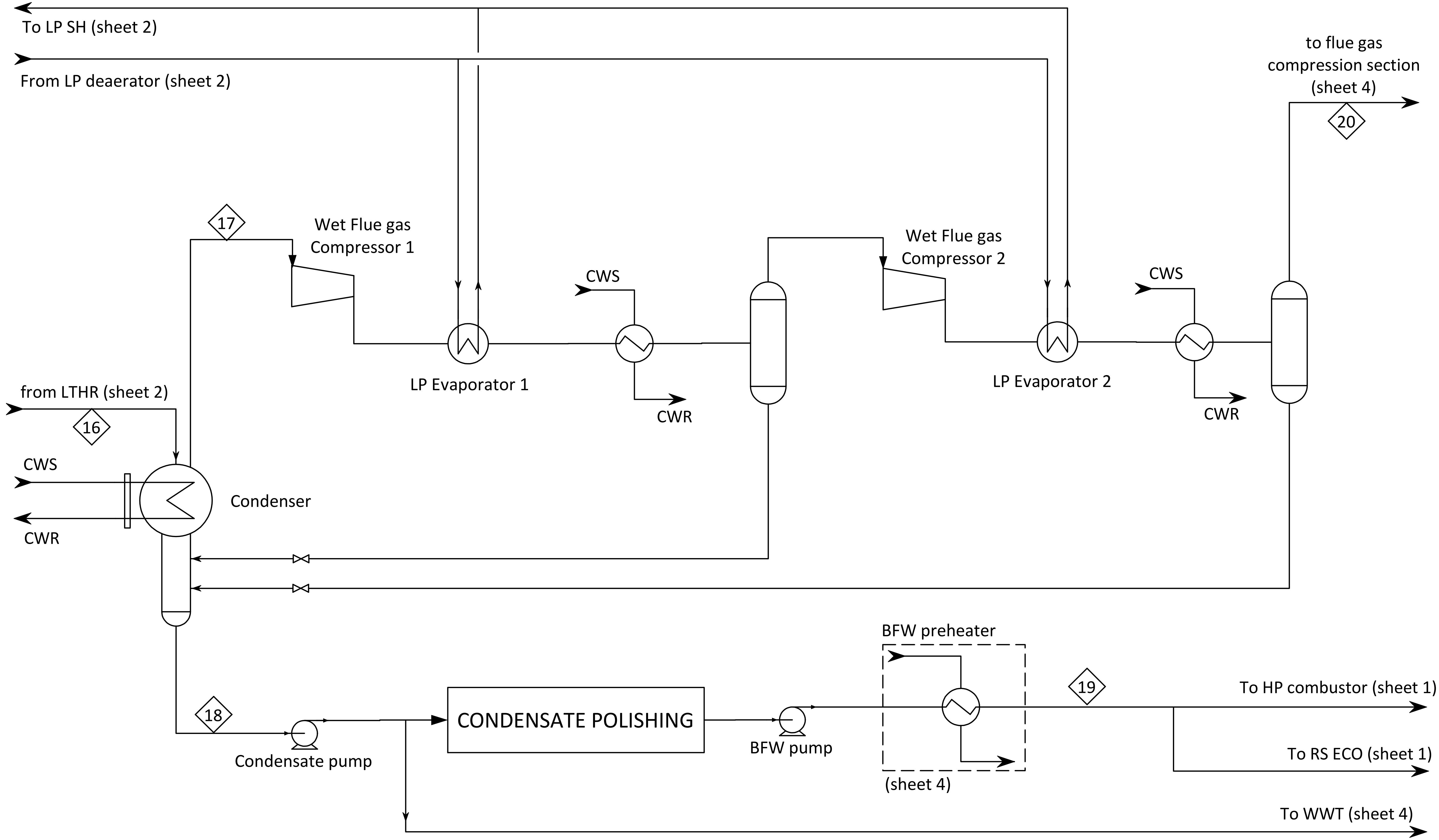
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



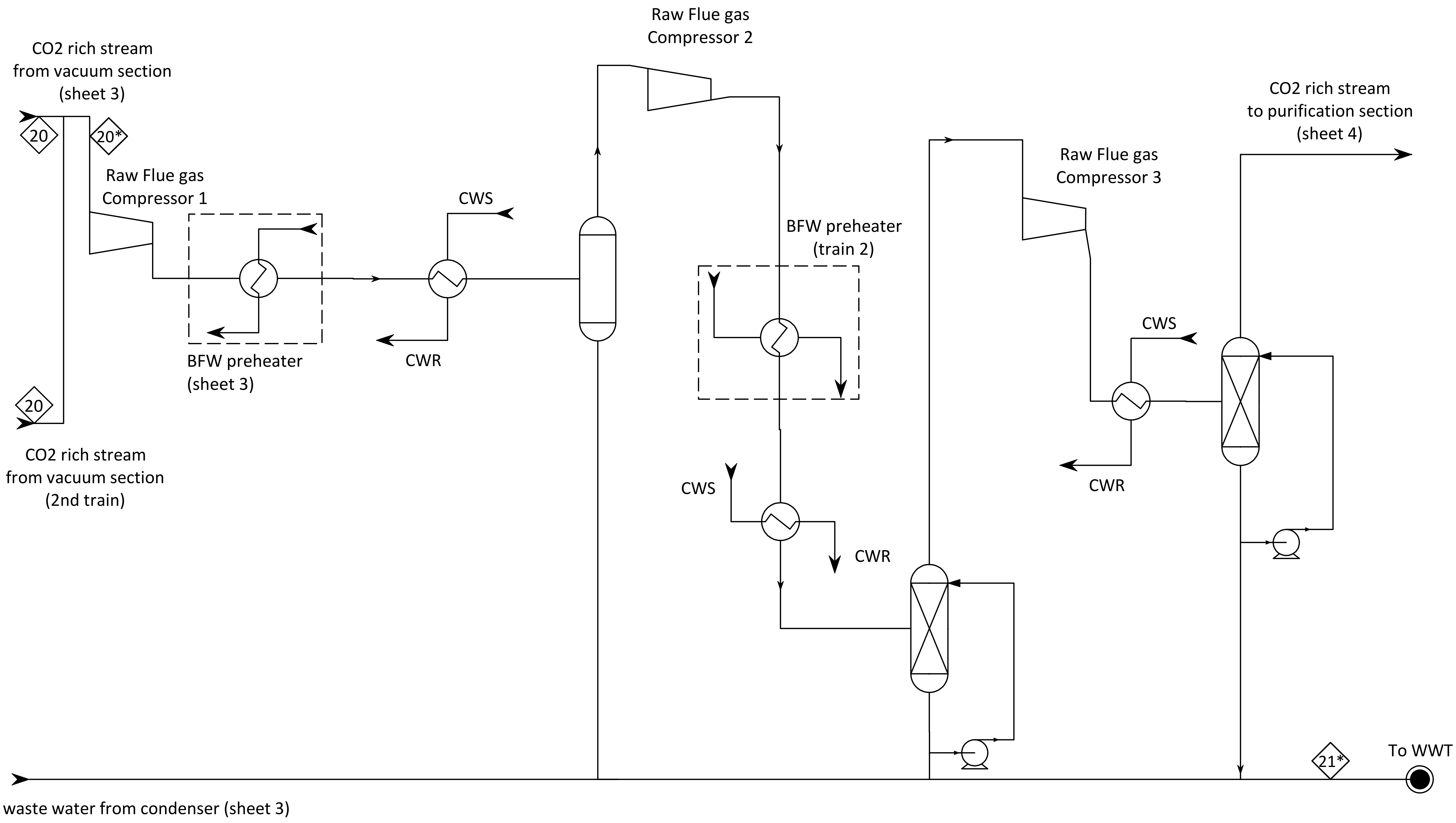
0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Oxy-turbine	
Rev.	Date	Comment	By	Appr	**CASE:** 4a - CES CYCLE	Sheet 01 of 05



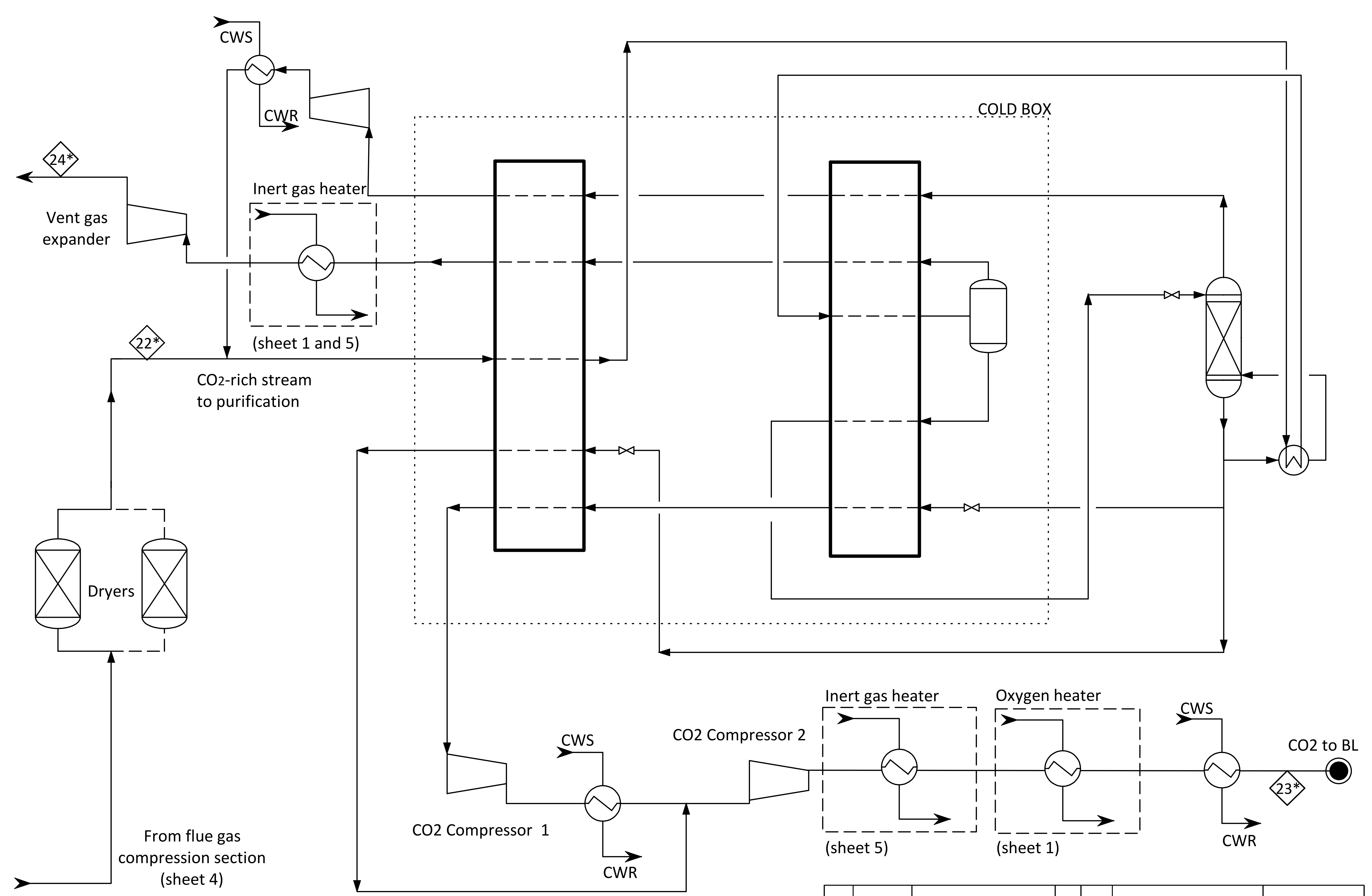
0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Low temperature heat recovery	
Rev.	Date	Comment	By	Appr	**CASE:** 4a - CES CYCLE	Sheet 02 of 05



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** Vacuum section and recycle water system	
Rev.	Date	Comment	By	Appr	**CASE:** 4a - CES CYCLE	Sheet 03 of 05



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** CO2 purification - compression section	
Rev.	Date	Comment	By	Appr	**CASE:** 4a - CES CYCLE	Sheet 04 of 05



0	Jun 2014	Draft report issue	FF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** 4a - CES CYCLE	Sheet 05 of 05

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

	Case 4a - CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0	1	
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	NF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	June 2014	February 2015	

HEAT AND MATERIAL BALANCE
UNIT 3000 - OXY TURBINE

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	NG to HP combustor	NG to MP combustor	Air to ASU	Oxygen from ASU	Oxygen to HP combustor	Oxygen to MP combustor	Recycle BFW to HP combustor	steam to HP combustor	Flue gas to HP turbine
Temperature (°C)	15	98	15	9	15	135	150	50	354	900
Pressure (bar)	70	170	30	amb	30	173	30	178	173	162.5
TOTAL FLOW										
Mass flow (kg/h)	59,470	33,185	26,285	1,013,130	232,100	130,770	101,330	278,700	120,705	563,365
Molar flow (kmol/h)	3,300	1,840	1,460	35,105	7,225	4,070	3,155	15,470	6,700	28,170
LIQUID PHASE										
Mass flow (kg/h)								278,700		
GASEOUS PHASE										
Mass flow (kg/h)	59,470	33,185	26,285	1,013,130	232,100	130,770	101,330		120,705	563,365
Molar flow (kmol/h)	3,300	1,840	1,460	35,105	7,225	4,070	3,155		6,700	28,170
Molecular Weight (kg/kmol)	18.0	18.0	18.0	28.86	32.1	32.1	32.1		18.0	20.0
Composition (%mol)	as assigned	as assigned	as assigned							
Ar				0.92%	2.00%	2.00%	2.00%	-	-	0.29%
CO_2				0.04%	0.00%	0.00%	0.00%	-	-	7.09%
H_2O				0.97%	0.00%	0.00%	0.00%	100.0%	100.0%	92.01%
N_2				77.32%	1.00%	1.00%	1.00%	-	-	0.20%
O_2				20.75%	97.00%	97.00%	97.00%	-	-	0.41%
Total				100.00%	100.00%	100.00%	100.00%	100.0%	100.0%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 4a - CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0	1	
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	NF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	June 2014	February 2015	

HEAT AND MATERIAL BALANCE
UNIT 3000 - OXY TURBINE

STREAM	11	12	13	14	15	16	17	18	19	20
	HP turbine exhaust flow	HP turbine exhaust flow to combustor	MP turbine Cooling stream	Flue gas to LP turbine	LP turbine exhaust flow	Exhaust flow to condeser	Wet flue gas from condenser	Condensate from condenser	BFW recycle	Flue gas to CPU
Temperature (°C)	580	316	316	1533	274	74	29	29	50	28
Pressure (bar)	27.5	27.0	27.0	26.5	0.10	0.10	0.10	0.10	200	1.05
TOTAL FLOW										
Mass flow (kg/h)	563,365	332,375	230,990	459,990	690,980	690,980	220,175	517,665	399,405	173,300
Molar flow (kmol/h)	28,170	16,620	11,550	21,300	32,850	32,850	6,720	28,735	22,170	4,115
LIQUID PHASE										
Mass flow (kg/h)								517,665	399,405	
GASEOUS PHASE										
Mass flow (kg/h)	563,365	332,375	230,990	459,990	690,980	690,980	220,175			173,300
Molar flow (kmol/h)	28,170	16,620	11,550	21,300	32,850	32,850	6,720			4,115
Molecular Weight (kg/kmol)	20.0	20.0	20.0	21.6	21.0	21.0	32.8			42.1
Composition (%mol)										
Ar	0.29%	0.29%	0.29%	0.52%	0.44%	0.44%	2.15%	0.00%	-	3.51%
CO_2	7.09%	7.09%	7.09%	12.96%	10.90%	10.90%	53.28%	0.00%	-	86.96%
H_2O	92.01%	92.01%	92.01%	85.73%	87.94%	87.94%	41.02%	100.00%	100.0%	3.73%
N_2	0.20%	0.20%	0.20%	0.37%	0.31%	0.31%	1.51%	0.00%	-	2.47%
O_2	0.41%	0.41%	0.41%	0.42%	0.42%	0.42%	2.04%	0.00%	-	3.33%
Total	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 4a - CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0	1	
	CLIENT :	IEAGHG	**PREP.**	NF	NF	
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	LM	LM	
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM	LM	
	LOCATION:	The Netherlands	**DATE**	June 2014	February 2015	

HEAT AND MATERIAL BALANCE

UNIT 4000 - CPU and Low pressure steam cycle

STREAM	**21***	**22***	**23***	**24***	**25***	**26***	**27***			
	Waste water to WWT	CO2-rich stream to purification	CO2 to BL	Inert gas stream	SH LP steam to steam turbine	Exhaust steam to condenser	LP BFW to steam generator			
Temperature (°C)	30	28	30	80	260	29	90			
Pressure (bar)	2.5	33	110	1.1	1.00	0.04	2.5			
TOTAL FLOW										
Mass flow (kg/h)	241,955	341,260	284,635	56,625	87,350	87,350	87,350			
Molar flow (kmol/h)	13,430	7,930	6,470	1,460	4,850	4,850	4,850			
LIQUID PHASE										
Mass flow (kg/h)	241,955					87,350	87,350			
GASEOUS PHASE										
Mass flow (kg/h)		341,260	284,635	56,625	87,350					
Molar flow (kmol/h)		7,930	6,470	1,460	4,850					
Molecular Weight (kg/kmol)		43.0	44.0	38.8	18.0					
Composition (%mol)										
Ar	0.00%	3.65%	0.16%	19.10%	-	-	-			
CO_2	0.00%	90.33%	99.83%	48.23%	-	-	-			
H_2O	100.00%	0.00%	0.00%	0.00%	100.0%	100.0%	100.0%			
N_2	0.00%	2.56%	0.00%	13.94%	-	-	-			
O_2	0.00%	3.46%	0.01%	18.74%	-	-	-			
Total	100.00%	100.00%	100.00%	100.00%	100.0%	100.0%	100.0%			

NOTE

1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 4a – Water consumption summary

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	0
	PROJECT NAME: Oxy-turbine power pl:	DATE	May-14
	PROJECT No. : 1-BD-0764A	MADE BY	NF
	LOCATION : The Netherlands	APPROVED BY	LM

Case 4 - CES cycle

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water [DT = 11°C] [t/h]
	OXY-TURBINE CYCLE			
	Condensate and recycle water system			
	Condenser		**10**	**51,770**
	Turbine and generator Auxiliaries			**6,000**
	Condenser vent compressor after cooler			**6,260**
	AIR SEPARATION UNIT			
	MAC intercoolers			**9,150**
	BAC intercoolers			**890**
	Oxygen compressor intercoolers			**550**
4000	**CO_2 PURIFICATION UNIT**			
	CO2 purification unit			**2,980**
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	**1,450**		
	Demineralized water unit	**15**	**-10**	
	Waste Water Treatment and Condensate Recovery	**-230**		
	Balance of plant			
	BALANCE	**1,235**	**0**	**77,600**

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 4a – Electrical consumption summary

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	May-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 4 - CES cycle

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**OXY-TURBINE CYCLE**	
	Natural gas compressor	2,770
	Condensate and recycle water system	5,760
	Flue gas compression	25,115
	Turbine Auxiliaries + generator losses	4,980
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	127,160
	Booster air compressor and miscellanea	13,330
	HP Oxygen compressor	14,160
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	27,230
	Autorefrigerated inerts removal unit compression consumption	14,000
	Autorefrigerated inerts removal unit expander production	-2,980
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	9,710
	Balance of plant	1,460
	BALANCE	**242,695**

6. Overall performance

The following table shows the overall performance of Case 4a, including CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	May-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 4 - CES cycle

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
HP turbine power output (@ gen terminals)	MWe	199.3
MP turbine power output (@ gen terminals)	MWe	472.9
LP turbine power output (@ gen terminals)	MWe	205.0
LP steam turbine (@ gen terminals)	MWe	24.5
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**901.6**
Oxy-turbine cycle (including NG compressor)	MWe	38.6
Air separation unit + Oxygen compressor	MWe	154.7
CO2 purification and compression unit	MWe	38.3
Utility & Offsite Units	MWe	11.2
ELECTRIC POWER CONSUMPTION	**MWe**	**242.7**
NET ELECTRIC POWER OUTPUT	**MWe**	**658.9**
(Step Up transformer efficiency = 0.997%) (B)	MWe	656.9
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**58.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**42.8%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**53.0%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**38.7%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6444
CO_2 removal efficiency	**%**	**90.0**
Fuel Consumption per net power production	**MWth/MWe**	**2.34**
CO_2 emission per net power production	**kg/MWh**	**47.1**

6.1. Comparison with literature performance data

Scope of this section is to compare the performance data of the case study with the performance data reported in the public domain, in particular the publication from DOE/NETL for the plant having same configuration as the present study case.

The following Table 4 summarises the cycle performance, in terms of electrical efficiency for:

- This study case, considering the configuration and the design basis, in terms of combustion outlet temperature, turbine inlet temperature, natural gas composition, 97%mol oxygen purity and especially turbine cooling flow requirements, as described in previous sections.
- Literature data, published by the DOE/NETL[1].
- In-house simulation considering the same turbine design conditions of the plant described in by the DOE/NETL.

For each of the above listed conditions, main design features potentially affecting the performance are also reported.

Table 4. Case 4a – Performance comparison

	Case 4a	**CES cycle** (as per DOE/NETL publication, [1])	**CES cycle** [(1)]
Performance			
Plant net electrical efficiency	42.8% (w/o LP steam cycle 40.6%)	49.6%	49.2%
Design parameters			
Cooling stream (fraction of flue gas from 1st combustor)	41%	9%	9%
NG to 1st reactor	56%	40%	40%
LP turbine outlet temperature	274°C	422°C	423°C
Condensation pressure	10 kPa	10 kPa	10 kPa
CO_2 purification	YES (O_2 content: 100 ppm)	NO	YES (O_2 content: 100 ppm)
CO_2 compression	110 barg	150 barg	110 barg

Notes

(1) Same gas turbine of the DOE/NETL publication, other basis as per present study

[1] DOE/NETL, 2010. *Carbon Capture Approaches for Natural Gas Combined Cycle Systems*. Report DOE/NETL-2011/1470, Revision 2, December 20, 2010.

Based on the above results, it is evident that the electrical efficiency drops while increasing the cooling requirements for gas turbine blades.

The amount of cooling stream in the study case is set in order to control the metal temperature below maximum allowed figure (assumed equal to 860°C for the 1st turbine stator). The available cooling medium is part of the cooled flue gas exhaust from the first expansion stage. The higher cooling flowrate estimated with respect to the gas turbine of the DOE/NETL publication, implies a higher portion of flue gas bypassing the second combustor.

Increasing this flowrate has the following negative effects on the cycle efficiency:

- The turbine inlet temperature is reduced and consequently both the turbine power output and the turbine outlet temperature decrease.
- As a consequence of the lower turbine outlet temperature, HP steam generation is not possible downstream LP turbine section.
- The CO_2 purification requires around 25-30% addition power consumption with respect to the power demand of the CO_2 compression alone, which accounts for around 0.4 percentage point efficiency difference (also including the difference relevant to the final CO_2 pressure).
- The fraction of HP turbine exhaust gas fed to the second combustor is smaller and consequently the amount of combustion gas to be heated-up in the second combustor is reduced. As a consequence, the combustor outlet temperature being fixed, a lower amount of natural gas can be burnt in the second combustor. This leads to a lower overall plant efficiency.

7. Environmental impact

The oxy-combustion gas turbine plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 5 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power island and in the CO_2 purification unit.

Table 5. Case 4a – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	56,520
Flow, Nm^3/h	32,700
Composition	(%mol)
Ar	19.10%
N_2	13.94%
O_2	18.74%
CO_2	48.23%
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from the flue gas condenser / compressor intercoolers and CO_2 purification unit) are treated to recover water, so the main liquid effluent is the cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 335 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

		REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
FOSTER WHEELER	CLIENT: IEAGHG	DATE	Jun-14	Feb-15		
	LOCATION: The Netherlands	ISSUED BY	NF	NF		
	PROJ. NAME: Oxy-turbine power plants	CHECKED BY	LM	LM		
	CONTRACT N.: 1-BD-0764 A	APPROVED BY	LM	LM		
	CASE.: 4a - CES cycle					

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	GAS TURBINE PACKAGE							
PK- 3101-1/2	**Gas Turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
T- 3101	HP expander		100 MWe Pin: 162.4 bar; Pout 27.6					*Including:* *Lube oil system*
T- 3102	MP expander		240 MWe Pin: 26.4 bar; Pout: 1 bar					*Cooling system* *Idraulic control system* *Seals system*
T- 3102	LP expander		105 MWe Pin: 1 bar; Pout: 0.1 bar					*Drainage system*
G- 3101	Oxy turbine generator							*Including relevant auxiliaries*
F- 3101	HP Combustor		*430 MWt*					*One per train, two in total*
F- 3102	MP Combustor		*340 MWt*					*One per train, two in total*
K- 3102-1/2	**NG compressor**		Flowrate: 42,000 Nm3/h Pin: 70 bar; Pout: 180 bar Compression ratio: 2.6	*1,600 kWe*				*One per train, two in total*
	HEAT RECOVERY SECTION and BFW SYSTEM							
PK- 3201-1/2	**Heat recovery section**							***2 x 50% package***
	Each including:							
E- 3201 A/B/C/D	BFW economisers							
E- 3202	Steam generator							
E- 3203	Inert gas heater							
E- 3204	Regenerator heater							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	BFW pumps	Centrifugal	450 m3/h x 1900 m	3000 kW				*One operating one spare, per each train*
	DRUM							
D- 3201	Deaerator							

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4a - CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
FLUE GAS CONDENSER PACKAGE and COMPRESSION PACKAGE								
PK- 3301-1/2	**Condenser Package**							***2 x 50% condenser package***
	Each including:							
	Flue gas condenser		185 MWth					*Including condenser hotwell*
	PUMPS		**Q [m3/h] x H [m]**					
P- 3301 A/B	Flue gas condensate pump	Centrifugal	570 m3/h x 50 m	110 kW				*One operating one spare, per each train*
PK- 3303-1/2	**Wet flue gas compressors**							***2 x 50% compression train***
	- *Wet flue gas compressor #1*	axial	Flowrate: 2 x 80,000 Nm3/h Pin: 10 kPa; Pout : 33 kPa Compression ratio: 3.3	*2 x 4,100kW*				*Two operating per each train*
	- *Wet flue gas compressor #2*	axial	Flowrate: 2 x 55,000 Nm3/h Pin: 30 kPa; Pout : 105 kPa Compression ratio: 3.5	*2 x 2,500 kW*				*Two operating per each train*
	Condesate separators							
	Intercoolers							
	Steam condensate pre-heater #1							
	Steam condensate pre-heater #2							
	CW cooler #1							
	CW cooler #2							
	LP STEAM CYCLE							
PK- 3401-1/2	**Heat recovery steam generation**							***2 x 50% package***
	Each including:							
E- 3401	BFW economisers							
E- 3402	Steam generator							
E- 3403	Steam superheater							

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4a - CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2		
		DATE	Jun-14	Feb-15				
		ISSUED BY	NF	NF				
		CHECKED BY	LM	LM				
		APPROVED BY	LM	LM				

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	LP STEAM CYCLE							
PK- 3402-1/2	**Steam turbine and Generator Package**							
	Including:							
ST- 3401	Steam turbine		12 MWe					*Including: Lube oil system Cooling system Idraulic control system Seals system Drainage system*
G- 3401	Steam turbine generator		15 MVA					*Including relevant auxiliaries*
	Inlet/after condenser Gland Condenser							
PK- 3403-1/2	**Steam Condenser Package**							***2 x 50% condenser package***
	Each including:							
	Steam condenser		60 MWth					*Including: Condenser hotwell Ejector Start-up Ejector*
PK- 3404-1/2	**Steam Turbine by-pass system**							***2 x 50% package***

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 4a - CES cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	LP STEAM CYCLE							
	PUMPS							
P- 3401 A/B	BFW pump	Centrifugal						*One operating one spare, per each train*
P- 3402 A/B	Steam condensate pumps	Centrifugal						*One operating one spare, per each train*
	DRUM							
	Deaerator							
	Steam generator blowdown drum (continuous)							
	Intermittent blowdown drum							
	EXCHANGER							
	Blowdown cooler							
	PACKAGES (Common to both train)							
PK- 3001	**Fluid Sampling Package**							
PK- 3002	**Phosphate Injection Package**							
	Phosphate storage tank							
	Phosphate dosage pumps							*One operating one spare*
PK- 3003	**Oxygen scavenger Injection Package**							
	Oxygen scavenger storage tank							
	Oxygen scavenger dosage pumps							*One operating one spare*
PK- 3004	**Amine Injection Package**							
	Amine storage tank							
	Amine dosage pumps							*One operating one spare*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4a - CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 4000 - CO2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	PACKAGES							
PK - 4001	**CO2-rich gas compression**							
	Including:							
	Raw flue gas compressors							
	- Raw flue gas compressor #1	axial	Flowrate: 2 x 92,500 Nm3/h Pin: 1.02 bar; Pout : 4.2 bar Compression ratio: 4.1	*2 x 5.6 MWe*				*2x50%*
	- Raw flue gas compressor #2	axial	Flowrate: 2 x 92,500 Nm3/h Pin: 4 bar; Pout : 15 bar Compression ratio: 3.8	*2 x 5.8 MWe*				*2x50%*
	- Raw flue gas compressor #3	axial	Flowrate: 2 x 100,000 Nm3/h Pin: 14.4 bar; Pout : 35 bar Compression ratio: 2.44	*2 x 3.5 MWe*				*2x50%*
	Condesate separators							
	Intercoolers							
	BFW pre-heater #1/#2							
	CW cooler #1 / #2							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO2 liquid separator							
	- CO2 distillation column							
	- CO2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MWe*				*2x50%*
	- CO2 compressor 2	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		3100 kW					*1x100%*
	- Overhead recycle compressors (optional)							
	- Intercoolers							
	Inert gas heater							
	NG heater							
	Cooling water intercoolers							

	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 4a - CES cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 5000 - Air Separation Unit (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5570 t/d					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 35.0 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		6.5 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
	- Oxygen Compressors	Centrifugal		6.0 MWe				*Two stages, intercooled*
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Jun-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: 4a - CES cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower**	Natural draft	990 MWth				concrete	
	including:		Diameter: 120 m,					
	Cooling water basin		Height: 180 m,					
			Water inlet: 17 m					
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/…/F	**Cooling Water Pumps**	Centrifugal	14,000 m3/h x 40 m	1780 kW				*Six in operation, one spare*
P- 6003 A/B	**Cooling tower make up pumps**	Centrifugal	1,500 m3/h x 30 m	160 kW				
	PACKAGES							
	Cooling Water Filtration Package							
	Cooling Water Sidestream Filters		7760 m3/h					
	Sodium Hypochlorite Dosing Package							
	Sodium Hypochlorite storage tank							
	Sodium Hypochlorite dosage pumps							
	Antiscalant Package							
	Dispersant storage tank							
	Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
PK- 6002	Let down station							
	RAW WATER SYSTEM							
PK- 6003	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Portable water pumps	centrifugal						*One operating, one spare*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4a - CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jun-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	DEMINERALIZED WATER SYSTEM							
PK- 6004	Demin Water Package, including: - Multimedia filter - Reverse Osmosis (RO) Cartidge filter - Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	CASE 4B: REVISED CES CYCLE
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	N. FERRARI
CHECKED BY	:	L. MANCUSO
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 4b of the study, which is the revised CES cycle based power plant, with cryogenic purification and separation of the carbon dioxide. The plant is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected revised CES plant configuration is based on two parallel trains, each including one F-class equivalent oxy-fired gas turbine, composed by two combustors and two expansion section and one heat recovery steam generator (HRSG), generating steam at one pressure level to be expanded in a back-pressure steam turbine to provide cooling medium for the turbine blades. As anticipated in chapter D, the revised CES cycle configuration is based on a LP turbine discharging at atmospheric pressure, not at vacuum conditions as in the base CES cycle.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 4a – Unit arrangement

Unit	Description	Trains
3000	Power Island	
	NG compressor	2 x 50%
	Gas Turbine	2 x 50%
	Recycled gas compression (included in GT package)	2 x 50%
	HRSG + Backpressure steam turbine	2 x 50%
	Water/steam system (including low pressure steam cycle)	2 x 50 %
4000	CO_2 purification and compression	
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power Island

The unit is mainly composed by two trains, each including:

- One F-class equivalent oxy-fired gas turbine, composed by:
    - HP combustor and expansion section (not cooled),
    - MP combustor and expansion section,
    - Recycle gas compressor.
- Heat recovery steam generator.
- Back pressure steam turbine.
- Injected BFW/Steam system.
- Low pressure steam cycle.

Technical information relevant to this unit is reported in chapter D, section 2.1.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.2.1. Gas Turbine expander design features

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the stream at gas turbine boundary are summarised in the following Figure 1 and Figure 2.

The natural gas from the metering station is diverted into two streams. Around 25% of the total overall flowrate is compressed to 120 bar and fed to the high pressure combustor of the gas turbine at around 65°C. The remaining fraction is first let down from the grid pressure to the required pressure level of the second combustor through the let-down station and, after being heated up against compressed CO_2 to 115°C in the CPU, is injected in the combustor.

Oxygen is delivered from the ASU at the pressure level required by the second combustor (around 35 bar). The flowrate required for the combustion in the first reactor is compressed to 120 bar in a dedicated compressor and injected in the combustor at around 150°C. The remaining portion is fed to the second combustor at ambient temperature.

The combustion outlet temperature of the HP combustor is controlled at 900°C injecting hot boiler water from the downstream heat recovery steam generator.

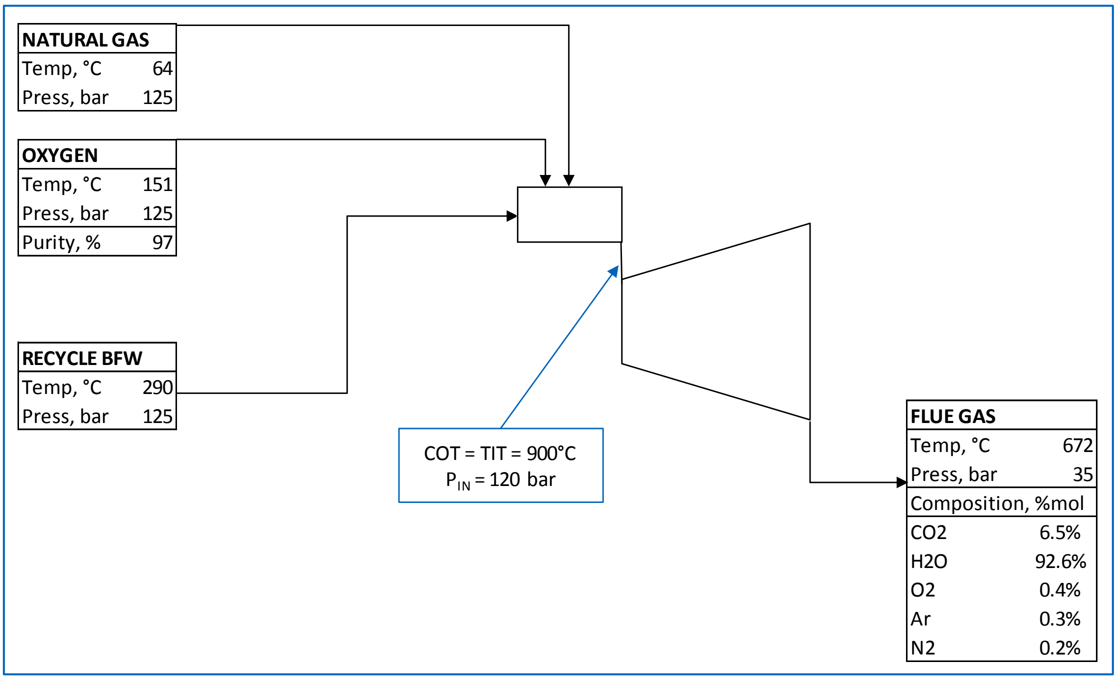


Figure 1. CES gas turbine – HP section

The HP turbine inlet conditions are set at 120 bar and 900°C. The temperature is lower than the maximum allowed for a cooled gas turbine, as the first stage of the CES gas turbine is not cooled.

The cooled exhaust flue gas from the HP turbine is fed to the second combustor. In this reactor, the exhaust flue gas, the natural gas and the oxygen stream are mixed with a flue gas recycled stream. The recycled gas flowrate to the combustor is set in order to control the combustion outlet temperature at 1533°C.

The HP turbine outlet pressure, also determining the MP turbine inlet pressure, is set at 35 bar. As the net CO_2 product is extracted downstream the recycle gas compressor, this figure is selected in order to have a CO_2-rich stream to the CPU at the pressure value required by the auto-refrigerated removal section, a pressure slightly higher than the ambient pressure to avoid leakages into the CO_2 loop.

The MP turbine expander has 5 stages, so as to have an acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine, at the rotational speed of 3600 RPM.

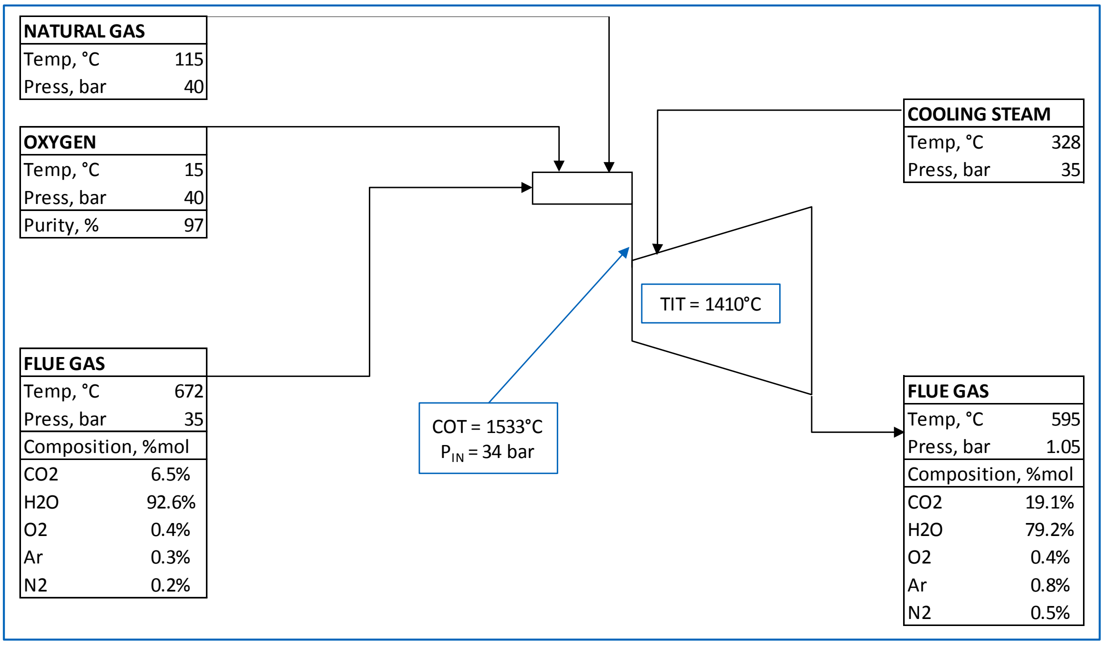


Figure 2. CES gas turbine – MP/LP section

2.2.2. *Heat recovery section*

The exhaust gases from the MP turbine enter the heat recovery section at 595°C.

The HRSG recovers heat available from the exhaust gas for super-heated high pressure steam generation and hot boiler feed water production. The HRSG is designed to generate the whole amount of steam required for MP turbine cooling and to pre-heat the boiler feed water injected in the HP combustor for combustion temperature control purpose. The cold boiler feed water from the deaerator enters the HRSG at around 90°C and steam is available at the steam turbine inlet at 170 bar and 570°C. The typical 25°C approach temperature between steam and exhaust gas temperature is considered to have an adequate heat transfer coefficient and limit the coils surface. Hot boiler feed water is injected into the HP combustor at 290°C.

High pressure steam is expanded in a back pressure steam turbine down to the pressure level required for MP turbine cooling, i.e. 35 bar, to enhance plant efficiency.

The cooling flowrate is set to control the gas turbine blade metal temperature as detailed below.

Turbine section	Maximum allowable temperature
1^{st} stator	860°C
1^{st} rotor	830°C
from 2^{nd} stator	830°C
from 2^{nd} rotor	800°C

2.2.3. *Flue gas recycle and low pressure steam cycle*

The cooled flue gas at 95°C from the HRSG is recycled back to the gas turbine through a double-stage intercooled compressor.

Heat available downstream the first compression stage is used for TSA regeneration and inert gas heating and for generating low pressure steam. Net CO_2 product stream is taken downstream the second compression stage at the pressure level required by the CPU.

This stream is cooled down with partial water condensation in two exchangers in series arrangement: the first heat exchanger is the low pressure steam superheater and the second is the LP BFW economiser.

2.2.4. *Heat integration*

The oxy-fuel cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular heat available at high temperature level from the oxy-turbine cycle is used to provide heat required in the CPU, while low grade heat available in the compressor intercoolers is used in the oxy-turbine cycle f in order to enhance gas turbine efficiency.

The following interfaces have been considered:

- Natural gas to the second combustor is heated against compressed CO_2 from the final compression before being sent to plant B.L.
- Heat available from the recycle gas compressor intercoolers is used as heating medium in the TSA regenerator and inerts gas heaters of the CPU.
- Heat available from the recycle gas compressor intercoolers in the oxy-cycle is used for LP steam generation.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 35 bar. Oxygen pressure is set by the requirement of the second gas turbine combustor.

A dedicated compressor is installed to compress the oxygen up to the pressure and the temperature level required by the first gas turbine combustor.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Due to the design feature of the Revised CES cycle and in particular the selection of the second combustor pressure in line with the feeding pressure to the auto refrigerated inerts removal section, the CPU for this particular case does not include the raw gas compression section.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

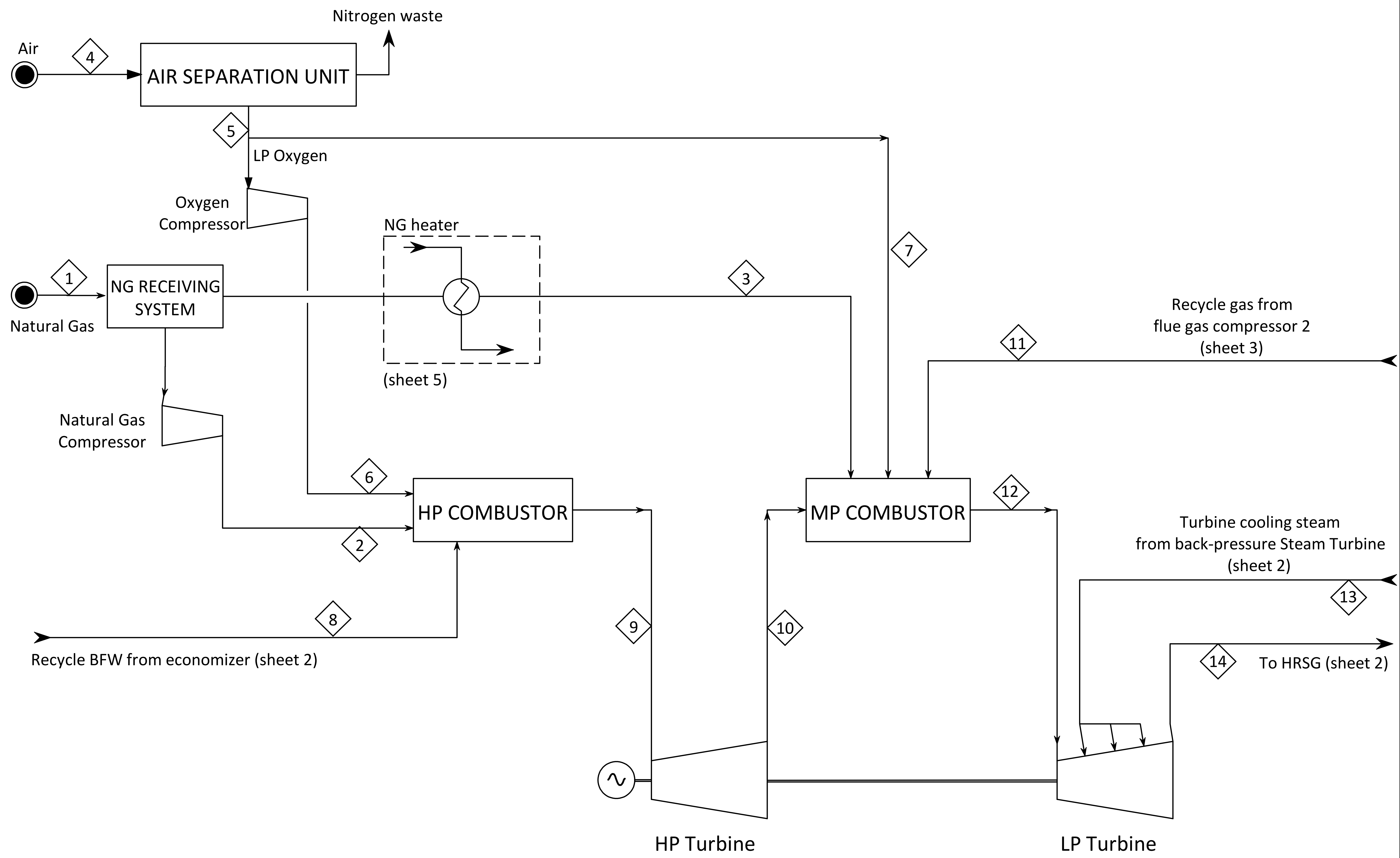
Basin diameter	130 m
Cooling tower height	180 m
Water inlet height	17 m

- Natural gas receiving station;
- Raw water system;
- Demineralised water plant;
- Waste Water Treatment
- Fire fighting system;
- Instrument and Plant air.

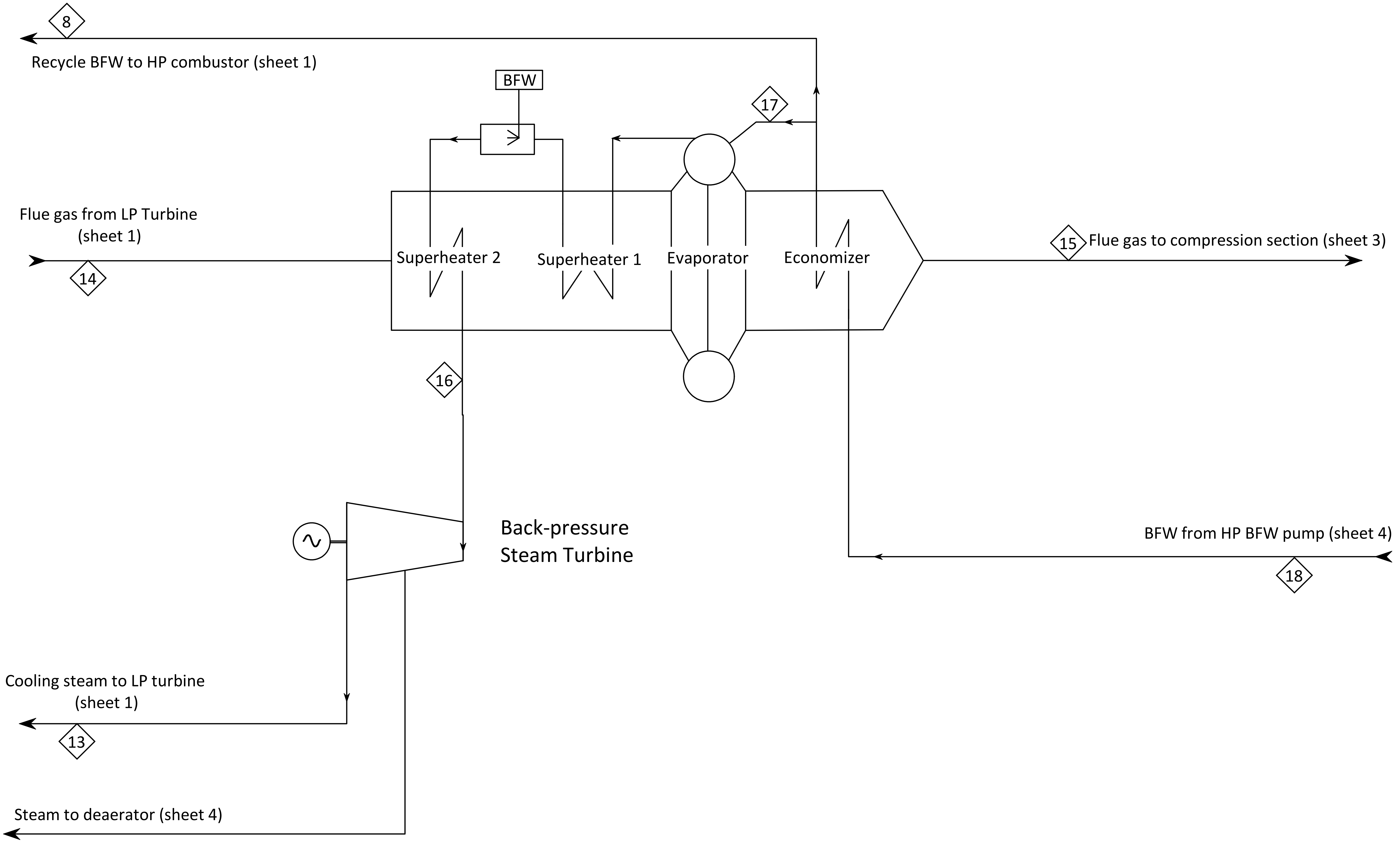
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

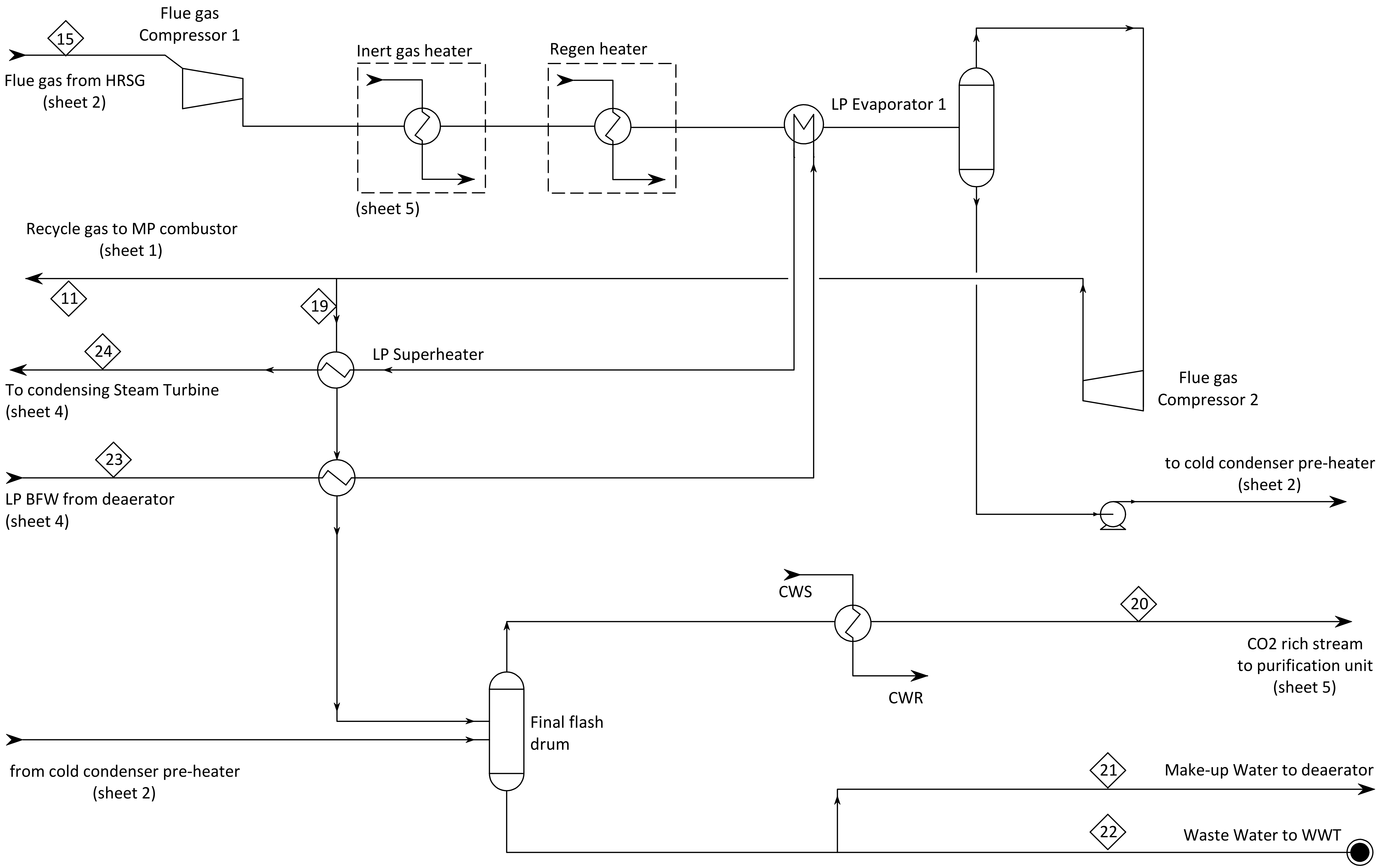
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



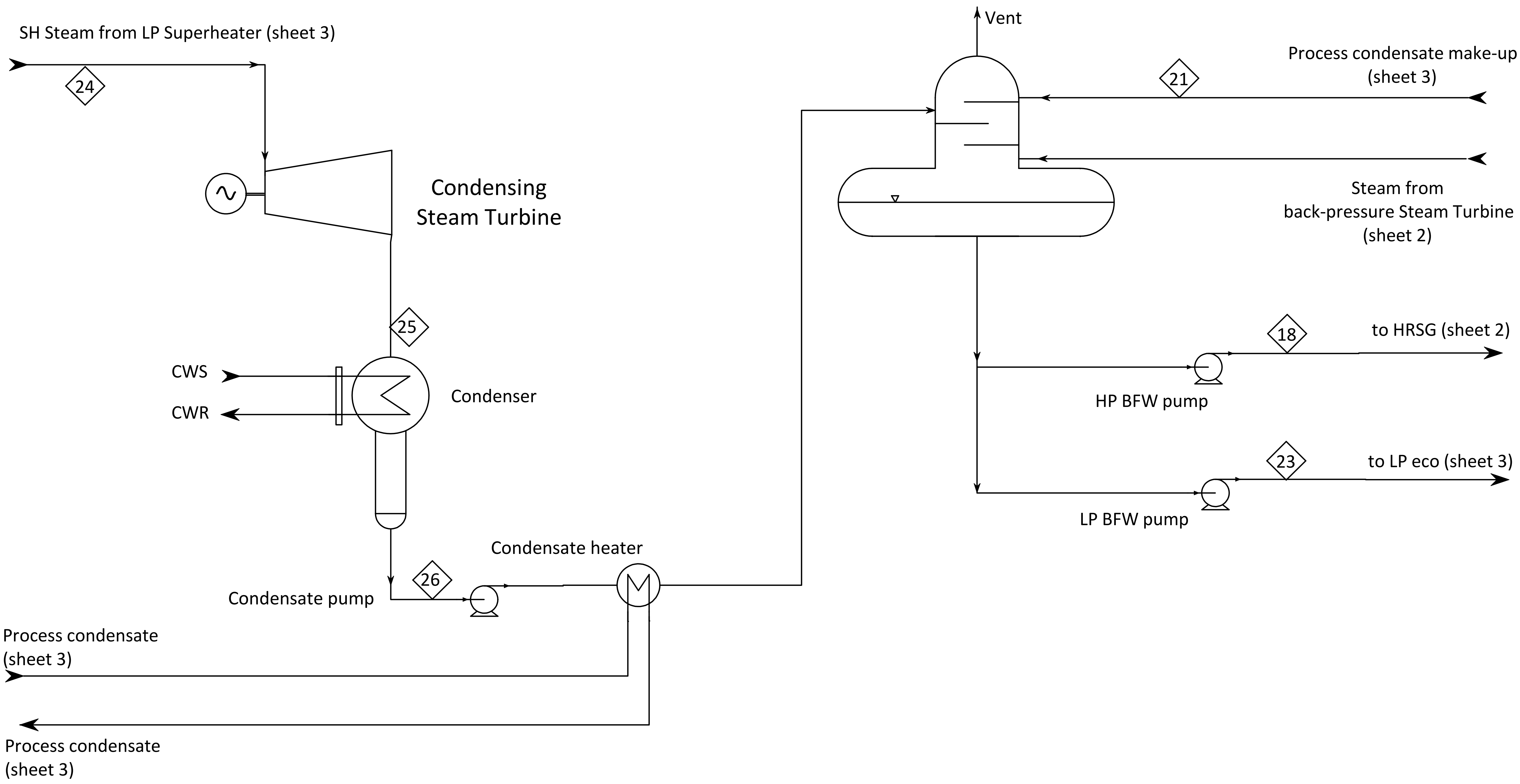
0	Jun 2014	Draft report issue	NF	LM	**UNIT:** Oxy-turbine	
Rev.	Date	Comment	By	Appr	**CASE:** 4b - Revised CES CYCLE	Sheet 01 of 05



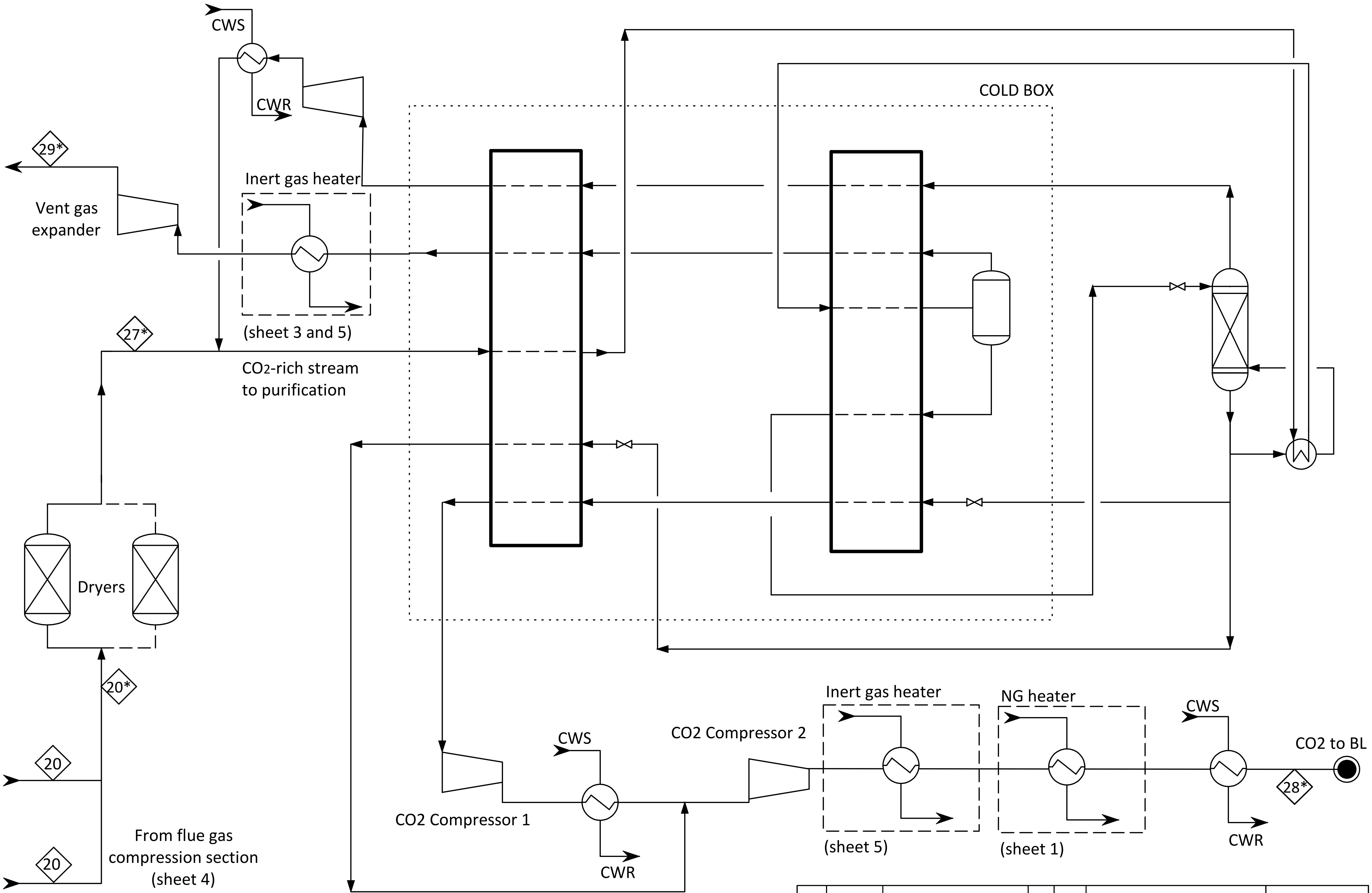
0	Jun 2014	Draft report issue	NF	LM	**UNIT:** HRSG and back pressure steam turbine	
Rev.	Date	Comment	By	Appr	**CASE:** 4b - Revised CES CYCLE	Sheet 02 of 05



0	Jun 2014	Draft report issue	NF	LM	**UNIT:** Flue gas compression and heat recovery	
Rev.	Date	Comment	By	Appr	**CASE:** 4b - Revised CES CYCLE	Sheet 03 of 05



0	Jun 2014	Draft report issue	NF	LM	**UNIT:** LP steam cycle	
Rev.	Date	Comment	By	Appr	**CASE:** 4b - Revised CES CYCLE	Sheet 03 of 05



0	Jun 2014	Draft report issue	NF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** 4b - Revised CES CYCLE	Sheet 05 of 05

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

	Case 4b - Revised CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0	1	
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	FF	NF	
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	NF	LM	
	PROJECT NO:	1-BD-0764 A	APPROVED	LM	LM	
	LOCATION:	The Netherlands	DATE	July 2014	February 2015	

HEAT AND MATERIAL BALANCE

UNIT 3000 - OXY TURBINE

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	NG to HP combustor	NG to MP combustor	Air to ASU	Oxygen from ASU	Oxygen to HP combustor	Oxygen to MP combustor	Recycle BFW to HP combustor	Flue gas to HP turbine	HP turbine exhaust gas
Temperature (°C)	15	65	115	9	15	151	15	290	900	672
Pressure (bar)	70	125	37	amb	40	125	37	125	120	36
TOTAL FLOW										
Mass flow (kg/h)	59,470	14,870	44,600	1,003,895	229,865	58,590	171,275	198,300	271,760	271,760
Molar flow (kmol/h)	3,300	825	2,475	34,785	7,160	1,825	5,335	11,010	13,695	13,695
LIQUID PHASE										
Mass flow (kg/h)								198,300		
GASEOUS PHASE										
Mass flow (kg/h)	59,470	14,870	44,600	1,003,895	229,865	58,590	171,275		271,760	271,760
Molar flow (kmol/h)	3,300	825	2,475	34,785	7,160	1,825	5,335		13,695	13,695
Molecular Weight (kg/kmol)	18.0	18.0	18.0	28.9	32.1	32.1	32.1		19.8	19.8
Composition (%mol)	as assigned	as assigned	as assigned							
Ar				0.92%	2.00%	2.00%	2.00%	-	0.27%	0.27%
CO_2				0.04%	0.00%	0.00%	0.00%	-	6.53%	6.53%
H_2O				0.97%	0.00%	0.00%	0.00%	100.00%	92.64%	92.64%
N_2				77.32%	1.00%	1.00%	1.00%	-	0.19%	0.19%
O_2				20.75%	97.00%	97.00%	97.00%	-	0.37%	0.37%
Total				100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE

1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

FOSTER WHEELER	**Case 4b - Revised CES Cycle - HEAT AND MATERIAL BALANCE**		**REVISION**	0	1
	CLIENT :	IEAGHG	**PREP.**	FF	NF
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	NF	LM
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM	LM
	LOCATION:	The Netherlands	**DATE**	July 2014	February 2015

HEAT AND MATERIAL BALANCE
UNIT 3000 - OXY TURBINE

STREAM	11	12	13	14	15	16	17	18	19	20
	Recycle flue gas from compressor	Flue gas to MP turbine	MP turbine cooling steam	MP turbine exhaust gas	Flue gas from HRSG	HP steam to back-pressure steam turbine	BFW to HP steam generator	BFW from HP BFW pump	Flue gas to final flash drum	Flue gas to purification unit
Temperature (°C)	350	1533	330	595	94	570	290	90	350	30
Pressure (bar)	36	34.5	35	1.07	1.05	170	175	190	36	34
TOTAL FLOW										
Mass flow (kg/h)	493,000	980,635	284,000	1,264,635	1,264,635	284,020	284,020	482,320	258,540	165,355
Molar flow (kmol/h)	17,000	38,620	15,765	54,385	54,385	15,765	15,765	26,775	8,915	3,830
LIQUID PHASE										
Mass flow (kg/h)								482,320		
GASEOUS PHASE										
Mass flow (kg/h)	493,000	980,635	284,000	1,264,635	1,264,635	284,020	284,020		258,540	165,355
Molar flow (kmol/h)	17,000	38,620	15,765	54,385	54,385	15,765	15,765		8,915	3,830
Molecular Weight (kg/kmol)	29.0	25.4	18.0	23.3	23.3	18.0	18.0		29.0	43.2
Composition (%mol)										
Ar	1.61%	1.08%	-	0.77%	0.77%	-	-	-	1.61%	3.74%
CO_2	56.44%	26.90%	-	19.10%	19.10%	-	-	-	56.44%	91.63%
H_2O	40.05%	70.74%	100.00%	79.22%	79.22%	100.00%	100.00%	100.00%	40.05%	0.20%
N_2	1.13%	0.76%	-	0.54%	0.54%	-	-	-	1.13%	2.63%
O_2	0.78%	0.52%	-	0.37%	0.37%	-	-	-	0.78%	1.80%
Total	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 4b - Revised CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0	1	
	CLIENT :	IEAGHG	**PREP.**	FF	NF	
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	NF	LM	
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM	LM	
	LOCATION:	The Netherlands	**DATE**	July 2014	February 2015	

HEAT AND MATERIAL BALANCE

UNIT 3000 - OXY TURBINE and UNIT 4000 - CPU

STREAM	21	22	23	24	25	26	27*	28*	29*	
	Make-up to deaerator	Waste water to WWT	BFW from LP BFW pump	SH LP steam to condensing steam turbine	Exhaust steam to condenser	Water from condeser	Dried CO2 stream to purification	CO2 to BL	Inert gas stream	
Temperature (°C)	81	81	88	250						
Pressure (bar)	34	34	4.5	2.00						
TOTAL FLOW										
Mass flow (kg/h)	482,300	119,630	661,640	661,640	661,640	661,640	330,450	278,810	51,640	
Molar flow (kmol/h)	26,770	6,640	36,725	36,725	36,725	36,725	7,650	6,340	1,310	
LIQUID PHASE										
Mass flow (kg/h)	482,300	119,630	661,640		40,885	661,640				
GASEOUS PHASE										
Mass flow (kg/h)				661,640	620,755		330,450	278,810	51,640	
Molar flow (kmol/h)				36,725	34,455		7,650	6,340	1,310	
Molecular Weight (kg/kmol)				18.0	18.0		43.2	44.0	39.4	
Composition (%mol)										
Ar	0.00%	0.00%	-	-	-	-	3.74%	0.25%	20.66%	
CO_2	0.00%	0.00%	-	-	-	-	91.81%	99.74%	53.44%	
H_2O	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	0.00%	0.00%	0.00%	
N_2	0.00%	0.00%	-	-	-	-	2.64%	0.00%	15.40%	
O_2	0.00%	0.00%	-	-	-	-	1.81%	0.01%	10.50%	
Total	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	

NOTE

1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals Consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 4b – Water consumption summary

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	Jul-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 4b - CES cycle - Revised configuration

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water 2° syst. [DT = 11°C] [t/h]
	OXY-TURBINE CYCLE			
	Condensate and recycle water system			
	Condenser		10	65,430
	Turbine and generator Auxiliaries			3,480
	AIR SEPARATION UNIT			
	MAC intercoolers			9,180
	BAC intercoolers			970
	Oxygen compressor intercoolers			
4000	**CO_2 PURIFICATION UNIT**			
	CO2 purification unit			1,900
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	1,455		
	Demineralized water unit	15	-10	
	Waste Water Treatment and Condensate Recovery	-230		
	Balance of plant			
	BALANCE	1,240	0	80,960

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 4b – Electrical consumption summary

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Jul-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 4b - CES cycle - Revised configuration

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**OXY-TURBINE CYCLE**	
	Natural gas compressor	**720**
	Condensate and recycle water system	**7,660**
	Turbine Auxiliaries + generator losses	**4,940**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**127,100**
	Booster air compressor and miscellanea	**12,300**
	HP Oxygen compressor	**4,200**
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	**0**
	Autorefrigerated inerts removal unit compression consumption	**13,760**
	Autorefrigerated inerts removal unit expander production	**-2,690**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**10,130**
	Balance of plant	**1,460**
	BALANCE	**179,580**

6. Overall Performance

The following table shows the overall performance of Case 4b, including CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	Jul-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 4b - CES cycle - Revised configuration

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
HP turbine power output	MWe	70.5
MP turbine power output	MWe	855.3
Condensing Steam turbine power output	MWe	208.8
Backpressure Steam turbine power output	MWe	62.6
Recycle gas compressor	MWe	-357.5
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**839.8**
Oxy-turbine cycle (including NG compressor)	MWe	13.3
Air separation unit + Oxygen compressor	MWe	143.6
CO2 purification and compression unit	MWe	11.1
Utility & Offsite Units	MWe	11.6
ELECTRIC POWER CONSUMPTION	**MWe**	**179.6**
NET ELECTRIC POWER OUTPUT	**MWe**	**660.2**
(Step Up transformer efficiency = 0.997%) (B)	MWe	658.2
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**54.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**42.8%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**49.4%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**38.8%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6458
CO_2 removal efficiency	**%**	**90.2**
Fuel Consumption per net power production	**MWth/MWe**	**2.33**
CO_2 emission per net power production	**kg/MWh**	**47.0**

6.1. Comparison with literature performance data

Scope of this section is to compare the performance data of the case study with the performance data reported in the public domain, in particular in the publication from CO_2-Global[1] for the plant having similar configuration as the present case.

With respect to the configuration proposed in the above mentioned publication and reported in chapter D section 2.1.4, the following design modifications have been introduced:

- Two independent cycles have been considered for HP steam and LP steam generations. HP BFW from a common deaerator is sent to the HRSG for hot BFW production and steam generation, respectively for HP combustor temperature control and for the turbine blade cooling purpose. LP BFW has been sent to the compressor intercoolers and aftercoolers for LP steam generation to be expanded in a condensing steam turbine
- Two different steam turbines have been included in the design. A back pressure steam turbine that expands the steam generated in the HRSG down to the pressure level required for turbine blade cooling. All the steam generated in the HRSG is used for cooling purpose. The condensing steam turbine expands only the LP steam generated in the heat recovery section. In the scheme proposed by CO_2-Global the excess steam not required for cooling purpose is expanded to low pressure, re-heated, mixed with the additional LP steam generated and re-injected in the LP section of the condensing steam turbine. The main driver of this design changes is that, based on the gas turbine modelling results, all the HP steam generated in the HRSG is required for the gas turbine blades cooling.

The following Table 4 summarises the cycle performance, in terms of electrical efficiency for:

- This study case, considering the configuration and the design basis, in terms of combustion outlet temperature, turbine inlet temperature, natural gas composition, 97%mol oxygen purity and especially turbine cooling flow requirements, as described in previous sections.
- Literature data, published by the CO_2-Global.

[1] C. Hustad, CEO, CO_2-Global, *CO_2 Compression for Advanced Oxy-Fuel Cycles*, At Workshop on Future Large CO_2 Compression Systems, DOE Office of Clean Energy Systems, EPRI, and NIST, Gaithersfield, MD - March 30th - 31st , 2009

Table 4. Case 4b – Performance comparison

	Case 4b	**Revised CES cycle** (as per CO_2-Global publication, [1])
Plant net electrical efficiency	42.8%	47%
Cooling flow (%wt of the inlet flow)	29.0%	4.5%

As already noticed for the base CES case 4a, it is clear that the electrical efficiency drops while increasing the cooling requirements for the gas turbine blades. Deeper analysis is not possible as design criteria considered in the publication, e.g. TIT and COT, are not available; furthermore, H&MB data are indicative only.

7. Environmental impact

The oxy-combustion gas turbine plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 5 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power island and in the CO_2 purification unit.

Table 5. Case 4b – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	51,640
Flow, Nm^3/h	29,365
Composition	(%mol)
Ar	20.66%
N_2	15.40%
O_2	10.50%
CO_2	53.44%
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from the flue gas compressor intercoolers and CO_2 purification unit) are treated to recover water, so the main liquid effluent is the cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 350 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
LOCATION: The Netherlands	DATE	Jul-14	Feb-15		
PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
CASE.: 4b - Revised CES cycle	APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	GAS TURBINE PACKAGE							
PK- 3101-1/2	**Gas Turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
T- 3101	HP expander		35 MWe					*Including:*
			Pin: 120 bar; Pout 35					*Lube oil system*
								Cooling system
T- 3102	MP expander		430 MWe					*Hydraulic control system*
			Pin: 34.5 bar; Pout: 1.05 bar					*Seals system*
								Drainage system
G- 3101	Oxy turbine generator							*Including relevant auxiliaries*
F- 3101	HP Combustor		*190 MWt*					*One per train, two in total*
F- 3102	MP Combustor		*580 MWt*					*One per train, two in total*
K- 3101	Recycle gas compressor							*Integrated with LP steam generation*
	1st stage		125 MWe					*(PK-3401)*
	2nd stage		55 MWe					
K- 3102-1/2	**NG compressor**		Flowrate: 18,500 Nm3/h	*400 kWe*				*One per train, two in total*
			Pin: 70 bar; Pout: 130 bar					
			Compression ratio: 1.85					
	HEAT RECOVERY SECTION and BFW SYSTEM							
PK- 3201-1/2	**Heat recovery steam generator**	Horizontal, Natural Circulated, 1 Pressure Level						***2 x 50% HRSG package***
D- 3201	HP steam drum							
E- 3201	HP Economizer 1st section							
E- 3202	HP Economizer 2nd section							
E- 3203	HP Evaporator							
E- 3204	HP superheater							

		CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4b - Revised CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
			DATE	Jul-14	Feb-15		
			ISSUED BY	NF	NF		
			CHECKED BY	LM	LM		
			APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	HP BFW pump	Centrifugal	550 m3/h x 1900 m	3400 kW				*One operating one spare, per each train*
P- 3202 A/B	LP BFW pump	Centrifugal	700 m3/h x 30 m	100 kW				*One operating one spare, per each train*
	DRUM							
D- 3201	Deaerator							
	Steam generator blowdown drum (continuous)							
	Intermittent blowdown drum							
	EXCHANGER							
	Blowdown cooler							
	PACKAGES (Common to both train)							
PK- 3202	**Fluid Sampling Package**							
PK- 3203	**Phosphate Injection Package**							
	Phosphate storage tank							
	Phosphate dosage pumps							*One operating one spare*
PK- 3204	**Oxygen scavenger Injection Package**							
	Oxygen scavenger storage tank							
	Oxygen scavenger dosage pumps							*One operating one spare*
PK- 3204	**Amine Injection Package**							
	Amine storage tank							
	Amine dosage pumps							*One operating one spare*
	BACK-PRESSURE STEAM TURBINE							
PK- 3301-1/2	**Back pressure steam turbine and Generator Package**							***2 x 50% package***
	Including:							
ST- 3301	Back-pressure steam turbine		32 MWe HP steam inlet: 170 bar Exhaust pressure: 36 bar					*Including:* *Lube oil system* *Cooling* *Idraulic control system* *Seals system (including gland condenser and vacuum system)* *Drainage system*
G- 3301	Steam turbine generator		40 MVA					*Including relevant auxiliaries*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4b - Revised CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jul-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	LP STEAM CYCLE							
PK- 3401-1/2	**Heat recovery steam generation**							***2 x 50% package***
	Each including:							
E- 3401	Condensate preheater							
E- 3402	BFW economisers							
E- 3403	Steam generator							
E- 3404	Steam superheater							
	LP STEAM CYCLE							
PK- 3402-1/2	**Steam turbine and Generator Package**							***2 x 50% package***
	Including:							
ST- 3401	Steam turbine		105 MWe					*Including:* *Lube oil system* *Cooling* *system* *Idraulic control system* *Seals system* *Drainage system*
G- 3401	Steam turbine generator		130 MVA					*Including relevant auxiliaries*
	Inlet/after condenser							
	Gland Condenser							
PK- 3403-1/2	**Steam Condenser Package**							***2 x 50% condenser package***
	Each including:							
	Steam condenser		420 MWth					*Including:* *Condenser hotwell* *Ejector* *Start-up Ejector*
PK- 3404-1/2	**Steam Turbine by-pass system**							***2 x 50% package***
	PUMPS		**Q [m3/h] x H [m]**					
P- 3401 A/B	Condensate pumps	Centrifugal	860 m3/h x 55 m	132 kW				*One operating one spare*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4b - Revised CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2		
		DATE	Jul-14	Feb-15				
		ISSUED BY	NF	NF				
		CHECKED BY	LM	LM				
		APPROVED BY	LM	LM				

EQUIPMENT LIST

Unit 4000 - CO_2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	PACKAGES							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO_2 liquid separator							
	- CO_2 distillation column							
	- CO_2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MWe*				*2x50%*
	- CO_2 compressor 2	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		2900 kW					*1x100%*
	- Overhead recycle compressors							
	- Intercoolers							
	Inert gas heater							
	NG heater							
	Cooling water intercoolers							

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4b - Revised CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jul-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 5000 - Air Separation Unit (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5520 t/d					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 35.0 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		7.0 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
	- Oxygen Compressors	Centrifugal		400 kWe				
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4b - Revised CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jul-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower** including: Cooling water basin	Natural draft	1050 MWth Diameter: 130 m, Height: 180 m, Water inlet: 17 m				concrete	
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/…/F	**Cooling Water Pumps**	Centrifugal	14,500 m3/h x 40 m	1850 kW				*Six in operation, one spare*
P- 6003 A/B	**Cooling tower make up pumps**	Centrifugal	1,550 m3/h x 30 m	160 kW				
	PACKAGES							
	Cooling Water Filtration Package Cooling Water Sidestream Filters		8100 m3/h					
	Sodium Hypochlorite Dosing Package Sodium Hypochlorite storage tank Sodium Hypochlorite dosage pumps							
	Antiscalant Package Dispersant storage tank Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
PK- 6002	Let down station							
	RAW WATER SYSTEM							
PK- 6003	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Portable water pumps	centrifugal						*One operating, one spare*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4b - Revised CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Jul-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	DEMINERALIZED WATER SYSTEM							
PK- 6004	Demin Water Package, including: - Multimedia filter - Reverse Osmosis (RO) Cartidge filter - Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : CASE 4C: SUPERCRITICAL CES CYCLE
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to Case 4c of the study, which is the CES cycle based power plant, with cryogenic purification and separation of the carbon dioxide. The plant is designed to fire natural gas, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The selected CES plant configuration is based on two parallel trains, each including one F-class equivalent oxy-fired gas turbine, composed of three combustors and three expansion sections with intermediate heat recovery section and final condenser. As anticipated in chapter D, while the configurations analysed 4a and 4b described respectively in chapter D.5 and D.6 are based on the "short-term" configuration proposed by CES, this cycle is the "long-term" solution based on CES last development. Main differences are the listed below:

- Combustor coolant stream is superheated steam at supercritical conditions (350 bar and 650°C);
- High pressure stage inlet at high pressure and temperature (300 bar and 1150°C), with blade cooling;
- Three combustors (or two reheating stages) configuration to allow a more efficient fuel split among the different combustor with respect to the two combustor solution.

The description of the main process units is covered in chapter D of this report and only features that are unique to this case are discussed in the following sections, together with the main modelling results.

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1.

Table 1. Case 4c – Unit arrangement

Unit	Description	Trains
3000	Power Island	
	NG compressor	2 x 50%
	Gas Turbine	2 x 50%
	Flue gas condenser	2 x 50%
	Wet flue gas compression	2 x 50%
4000	CO_2 purification and compression	
	Raw gas compression	2 x 50%
	Auto-refrigerated inert removal section	1 x 100%

Unit	Description	Trains
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	2 x 50%
6000	Utility and Offsite	N/A

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Unit 3000 – Power Island

The unit is mainly composed of two trains, each including:

- One F-class equivalent oxy-fired gas turbine, composed of:
  - HP turbine (HPT): combustor and expansion section
  - MP turbine (MPT): combustor and expansion section
  - LP turbine (LPT): combustor and expansion section.
- One flue gas condenser.
- Wet flue gas compressor.
- BFW/Steam system for combustor and HPT cooling.

Technical information relevant to this unit is reported in chapter D, section 2.1.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.2.1. Gas Turbine expander design features

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the stream at gas turbine boundary are summarised in the following Figure 1, Figure 2 and Figure 3.

The natural gas from the metering station is diverted into three streams. Around 23% of the total overall flowrate is compressed to 310 bar, and fed to the high pressure combustor of the gas turbine at around 150°C. Another portion (33%) is fed to the second combustor after being preheated at around 140°C, while remaining fraction is first let down from the grid pressure to the required pressure level of the third combustor through the let-down station and then injected in the combustor at around 125°C. Both the natural gas streams fed to the second and third combustor are pre-heated against the CO_2-rich feed stream to the CPU in the compressor intercoolers.

Oxygen is delivered from the ASU at the pressure level required by the second combustor (around 60 bar). The flowrate required for the combustion in the first reactor is compressed to 310 bar in a dedicated compressor and injected in the combustor at around 215°C. Both the oxygen streams to the second and third combustor are heated up against compressed CO_2 in the CPU at around 120°C before being injected to the combustor.

The fuel and oxidant stream to the first combustor are combined with high pressure superheated steam generated in the downstream heat recovery section. The steam generated in the power plant is regulated in order to control the combustion outlet temperature at 1150°C.

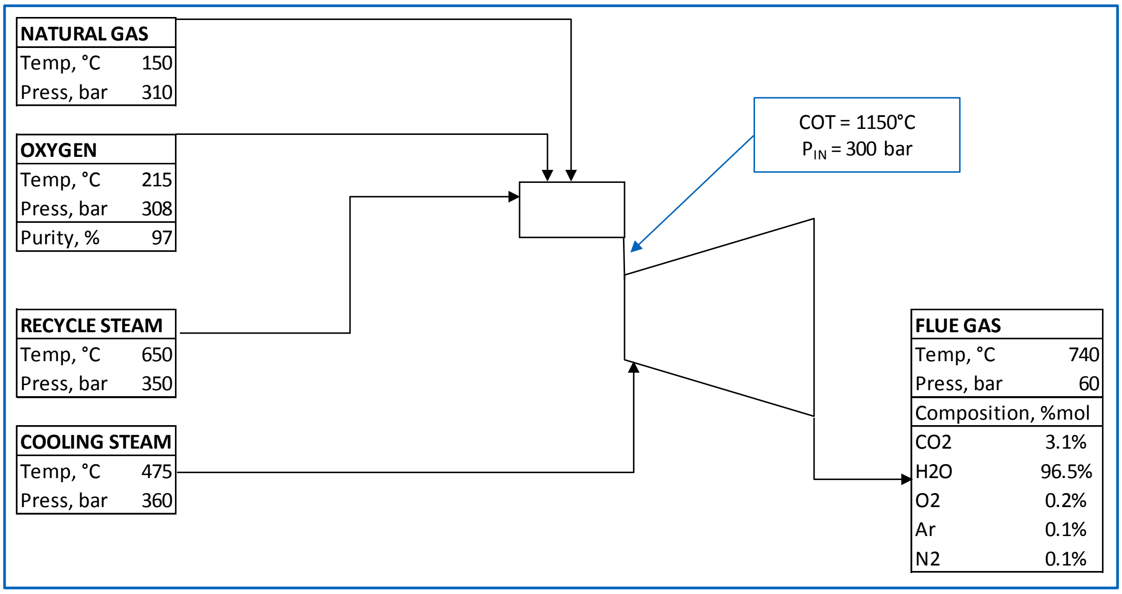


Figure 1. CES gas turbine – HPT section

The HPT inlet conditions are set at 300 bar and 1150°C, thus implying that the coolant stream is required for the HPT blades. Superheated steam upstream the final superheating stage at around 475°C is used as cooling medium for the HP turbine blade in order to maintain the blade metal temperature lower than 860°C.

Heat available from the flue gas exhaust exiting the HP section at 60 bar and 740°C is recovered superheating the HP steam superheating to be used for combustor temperature control.

Part of the cooled exhaust flue gas is fed to the second combustor, while the remaining fraction bypasses the combustor to be used for gas turbine cooling. The cooling flowrate is set to control the gas turbine blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st stator	860°C
1st rotor	830°C
from 2nd stator	830°C
from 2nd rotor	800°C

The MPT inlet conditions are set at 58.5 bar and 1533°C. Combustion outlet temperature is controlled acting on the natural gas flowrate to the second combustor.

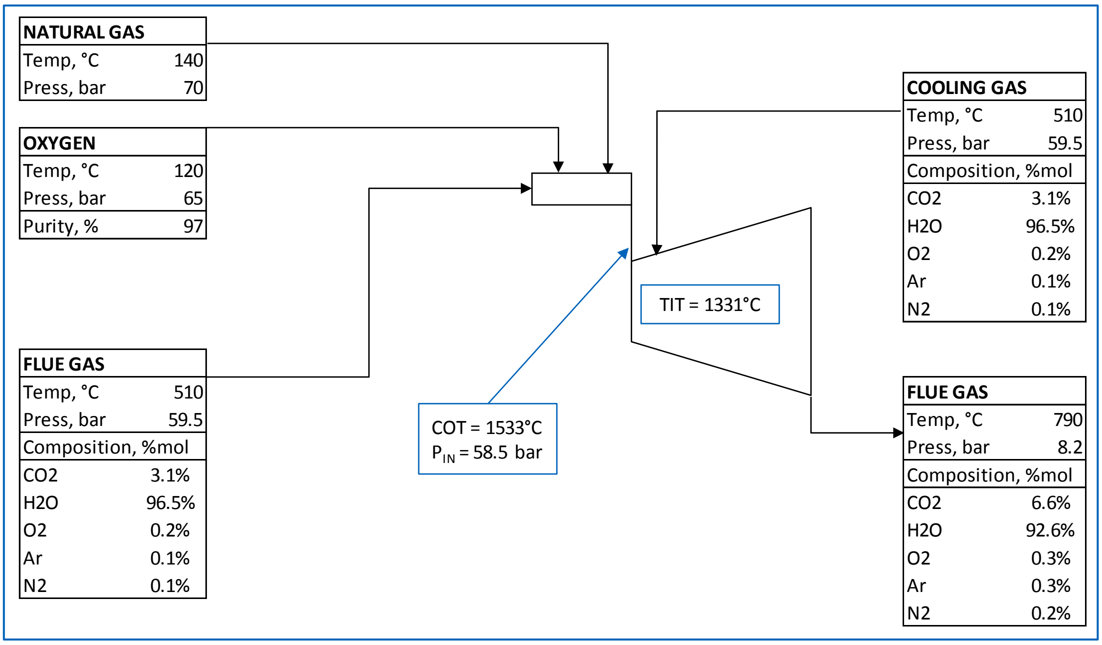


Figure 2. CES gas turbine – MPT section

The MP turbine has 4 cooled stages, so as to have an acceptable Mach number in the gas stream with a peripheral velocity similar to that of the reference gas turbine. The rotational speed of 6700 RPM has been selected to allow an adequate design of the stages given the limited volume flow rate.

Heat available from the flue gas exhaust exiting the MP section at 8.2 bar and 790°C is recovered by superheating the HP steam to be used for the HP combustor temperature control. Most of the steam downstream the heat exchanger is sent to the heat recovery section downstream the HP turbine for final heating before being injected in the combustor, while a fraction is used as cooling medium for the HP turbine blade.

Part of the cooled exhaust flue gas downstream the heat recovery section is fed to the third combustor, while the remaining fraction bypasses the combustor to be used for the LP turbine cooling. The cooling flowrate is set to control the gas turbine blade metal temperature as detailed in the above table for the MP turbine.

Pressure of the coolant stream at turbine inlet has to be high enough to allow the coolant can be injected in the main stream at the end of the cooling circuit.

In the CES configuration, as the cooling stream for the MP turbine and the LP turbine is part of the exhaust gas from the upstream sections of the gas turbine (respectively the HP turbine and the MP turbine) the pressure of the coolant streams are addressed to the turbines at a pressure slightly higher than the main flow given the pressure drop across respectively the HP and the MP combustors.

On the other hand, it is not convenient to increase the pressure drop available to the coolant stream by throttling the main flow because this would mean a loss of expansion work and consequently of cycle efficiency. As a consequence, the pressure losses have to be minimised and hence be the same of the combustor pressure drop (plus the pressure drop through the heat recovery heat exchanger upstream the combustor).

This requirement imposes a limitation on the minimum pressure of the LP combustor of around 8 bar (thus implying the mentioned MP turbine outlet pressure and LP turbine inlet pressure).

In fact, as the coolant stream inlet pressure reduces, the pressure losses in the cooling circuit increases because the flow density reduces and therefore the cooling stream velocity increases. And eventually there is no pressure margin left to inject the coolant in the main flowing stream.

According to the prediction of the simulation model, 7.6 bar is the minimum inlet pressure that allows a feasible cooling circuit with a combustor outlet temperature of 1533°C. Combustion outlet temperature is controlled acting on the natural gas flowrate to the third combustor.

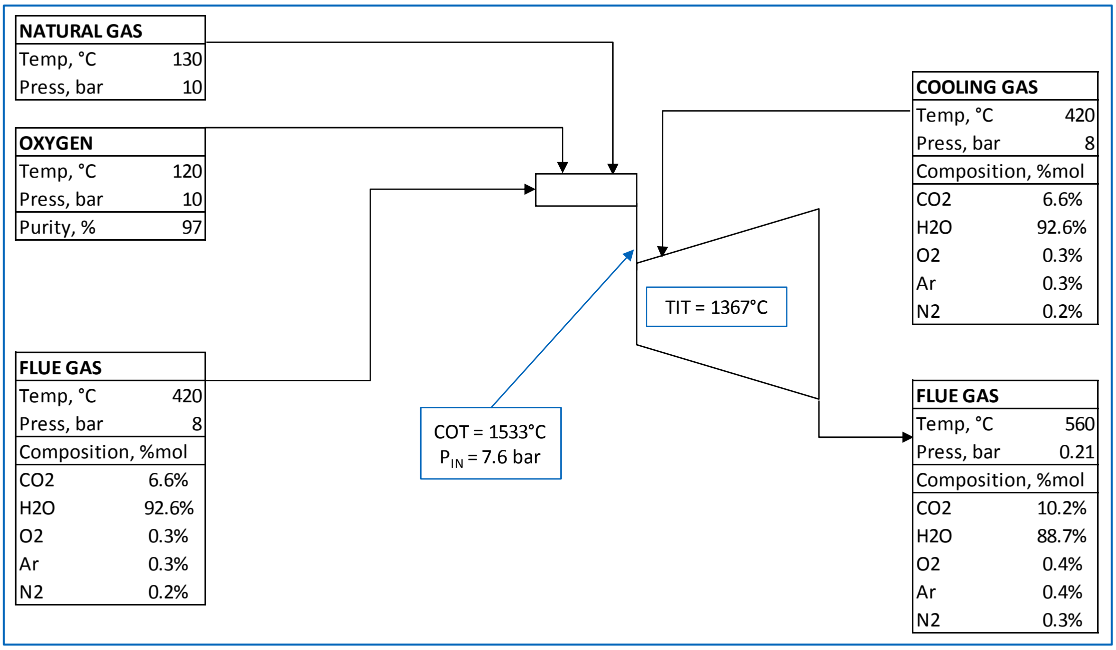


Figure 3. CES gas turbine – LPT section

The LP turbine has 3 cooled stages followed by an uncooled section. The LPT expands the flue gas down to condenser pressure, which is set to 0.21 bar, as the minimum pressure that allows heat recovery for HP steam generation downstream the expander. A rotational speed of 3000 RPM allows having an acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine at the given exhaust pressure.

The heat available from the flue gas exhaust from the LP section is recovered generating HP steam required for HP combustor temperature control from the boiler water.

2.2.2. *Heat recovery section*

The exhaust gases from the HP turbine enter the heat recovery section at 740°C. Heat is recovered to produce high-pressure superheated steam to be injected in the HP combustor, at 350 bar and 650°C. The HP steam is generated recovering heat from the flue gas downstream each section of the gas turbine. Final temperature of 650°C is the maximum temperature level considering reasonable approach in the three heat exchangers.

It has to be noted that, with respect to the first configuration proposed by CES and published by DOE and reproduced in this report as case 4a, the steam is generated at pressure higher than the critical point. As a consequence is it possible to enhance the heat recovery from the heat exchanger downstream the LP turbine, cooling down the exhaust gas to around 60°C for HP steam generation, as at this condition there is no evaporation plateau.

Inert gas and regenerator preheating in the CPU is also achieved recovering heat from the flue gas downstream the HP turbine.

2.2.3. *Flue gas condenser and compression section*

Cooled flue gas from the heat recovery section is cooled and partially condensed in a flue gas condenser.

The wet flue gas from the condenser is compressed up to atmospheric pressure and sent to the CPU. Due to the temperature profile through the whole expansion section, there is no need of boiler water pre-heating before sending the boiler water to the heat recovery section for steam generation. As for that, number of compression stage is optimised in order to reduce the power demand of the compressor. In addition, as cooling water is used in the compressor intercooler, the outlet temperature from each compressor stage is kept below 80-90°C.

Condensed water from the flue gas is recycled back to the condenser hotwell. Most of the flow is pumped to a condensate polishing section, in order to be treated and recycled back to the oxy-cycle to be used for coolant steam generation. The net water production from the combustion process is sent to the waste water treatment.

2.2.4. *Heat integration*

The oxy-fuel cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. In particular heat available at high temperature level from the oxy-turbine cycle is used to provide heat required in the CPU mainly for TSA regenerator heating and inert gas heating before expansion, while heat available at low temperature level in the compressor intercoolers is used for oxygen and natural gas pre-heating in order to enhance gas turbine efficiency.

The following interfaces have been considered:

- Oxygen to the second and third combustors is heated against compressed CO_2 from the final compression before being sent to plant B.L.
- Heat available from the flue gas exhaust from the HP turbine is used as heating medium in the TSA regenerator and inerts gas heaters of the CPU.
- Natural gas to the third combustor is heated against the CO_2-rich stream downstream the first compression stage in the raw gas compression section of the CPU.
- Natural gas to the second combustor is heated against the CO_2-rich stream downstream the second compression stage in the raw gas compression section of the CPU.

2.3. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 65 bar. Oxygen pressure is set by the requirement of the second gas turbine combustor.

A dedicated compressor is installed to compress the oxygen up to the pressure and the temperature level required by the first gas turbine combustor.

The oxygen flowrate is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.4. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- Raw flue gas compression (1 - 34 bar);
- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;

- The remaining part of the compression system up to 110 bar.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.5. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on one natural draft cooling tower, with the following main characteristics:

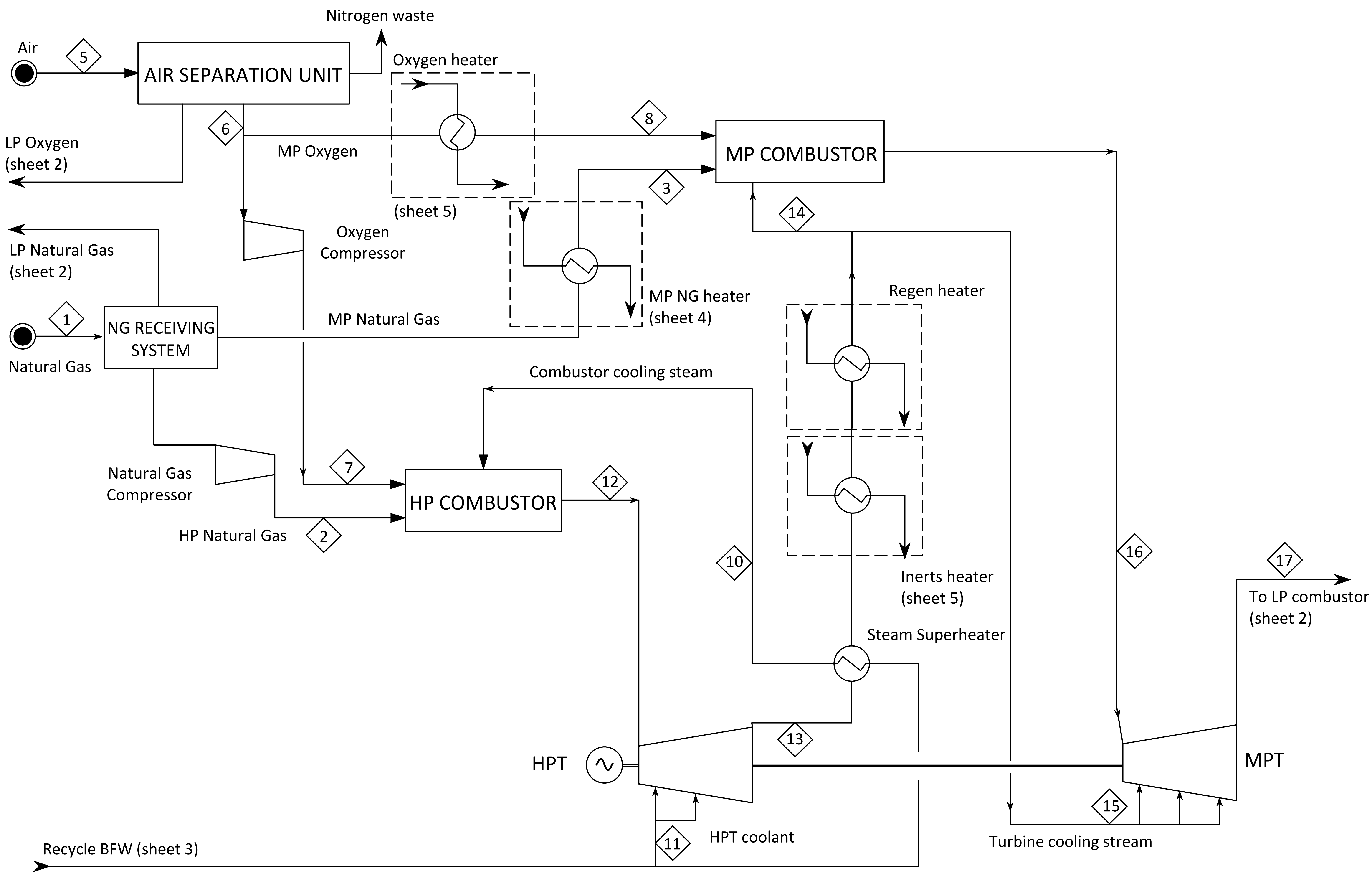
Basin diameter	120 m
Cooling tower height	180 m
Water inlet height	17 m

- Natural gas receiving station;
- Raw water system;
- Demineralised water plant;
- Waste Water Treatment
- Fire fighting system;
- Instrument and Plant air.

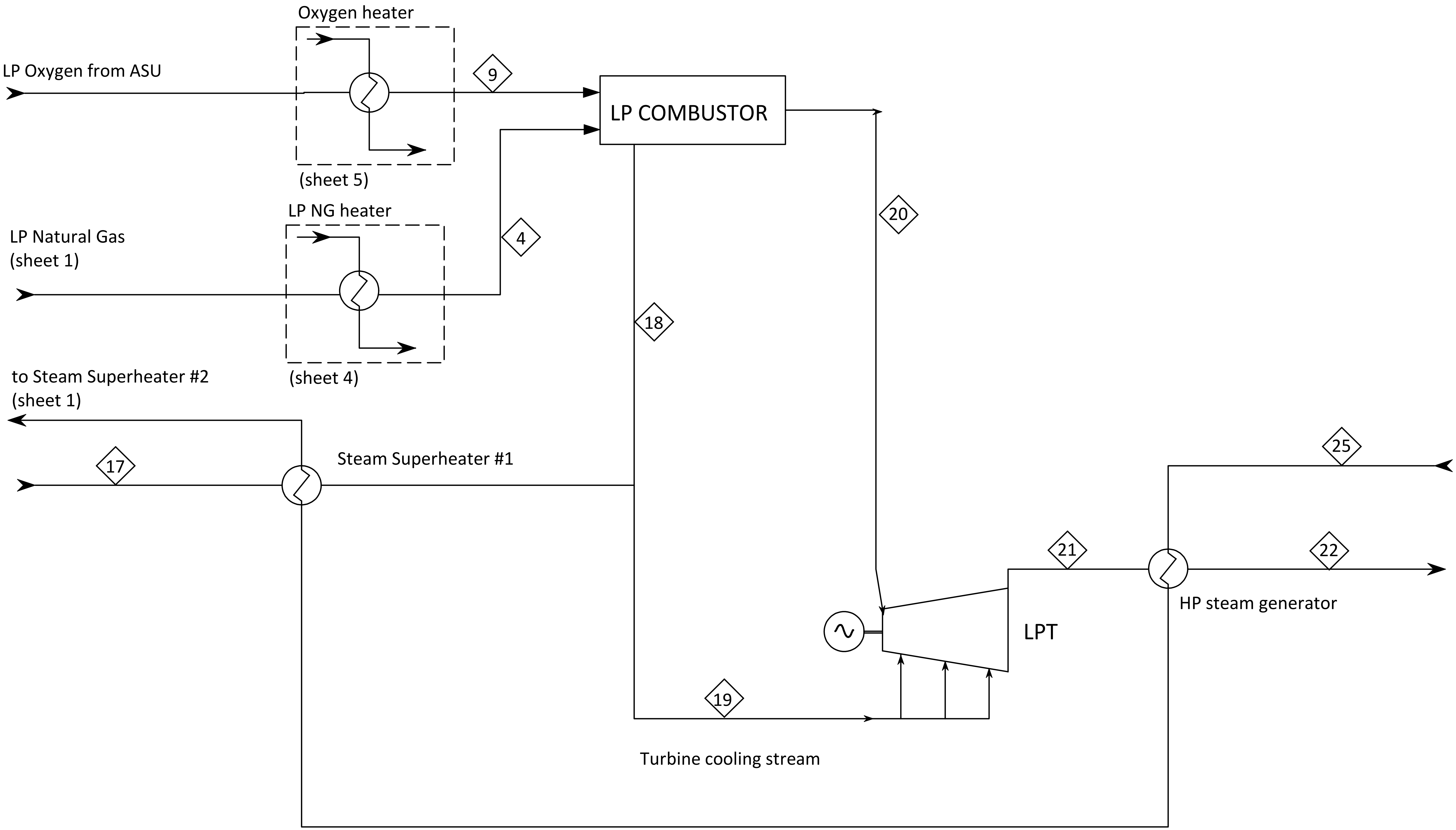
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

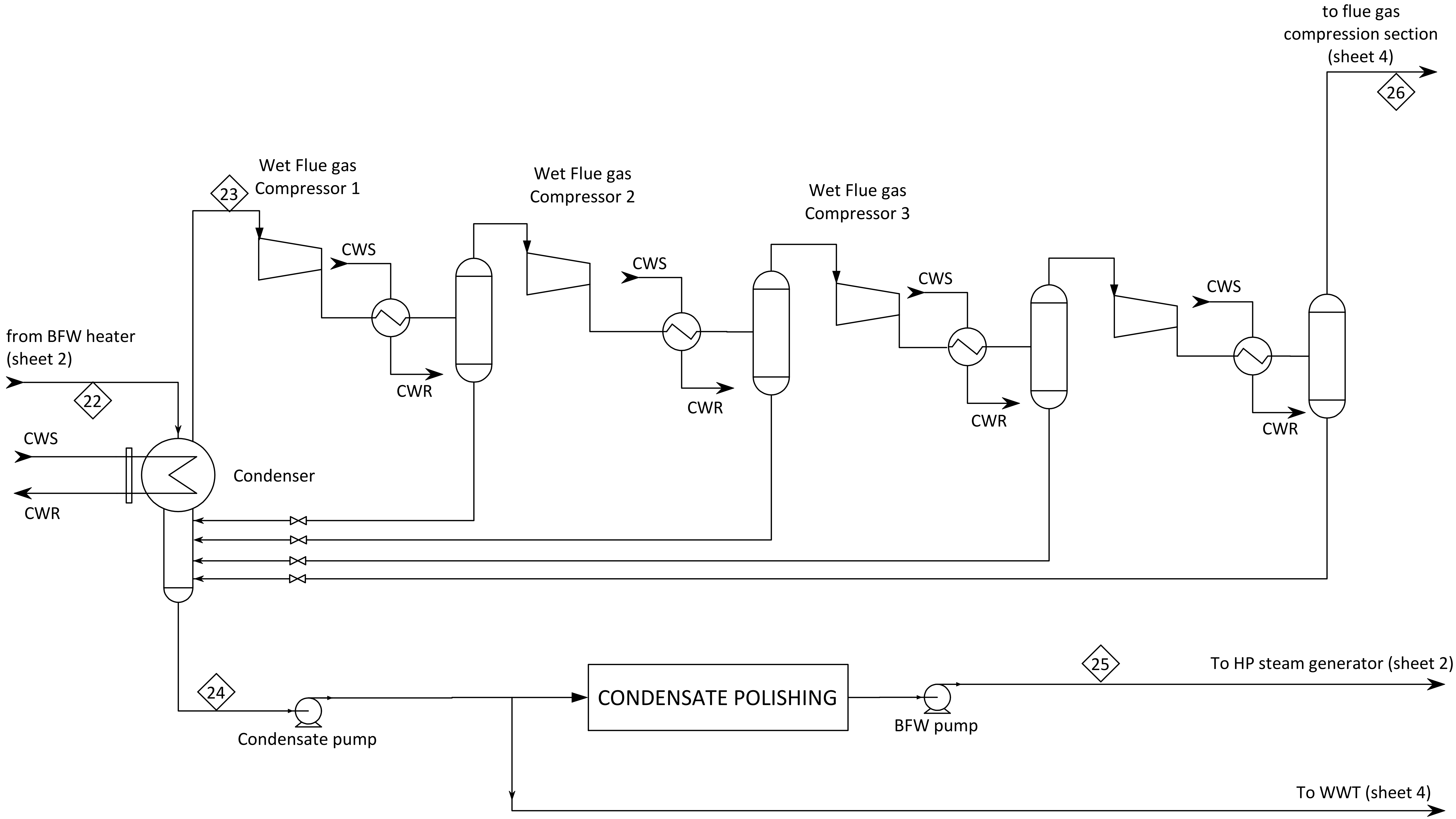
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



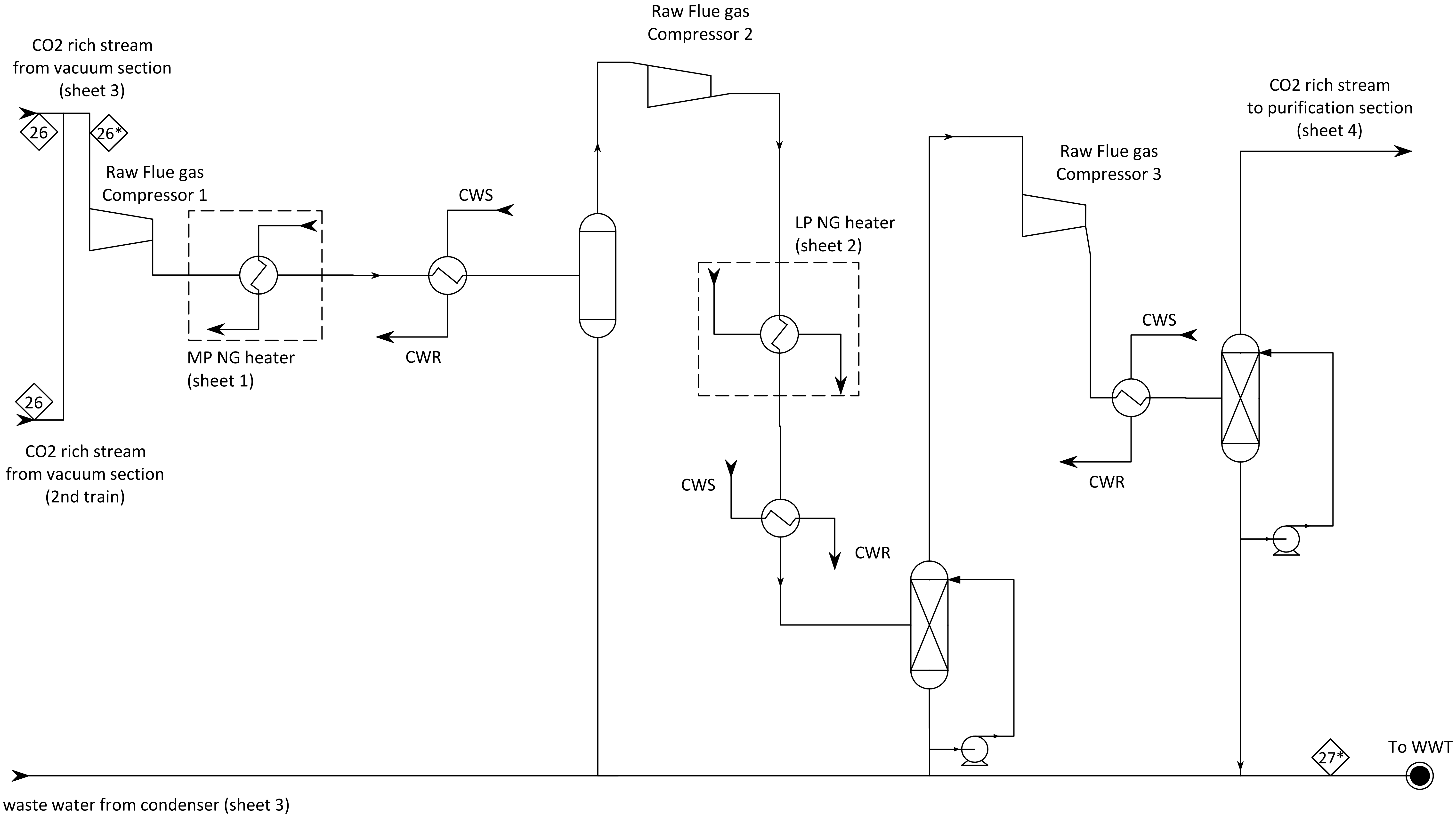
0	mar-15	Final report issue	NF	LM	**UNIT:** Oxy cycle: HPT and MPT	
Rev.	Date	Comment	By	Appr	**CASE:** 4c - CES CYCLE	Sheet 01 of 05



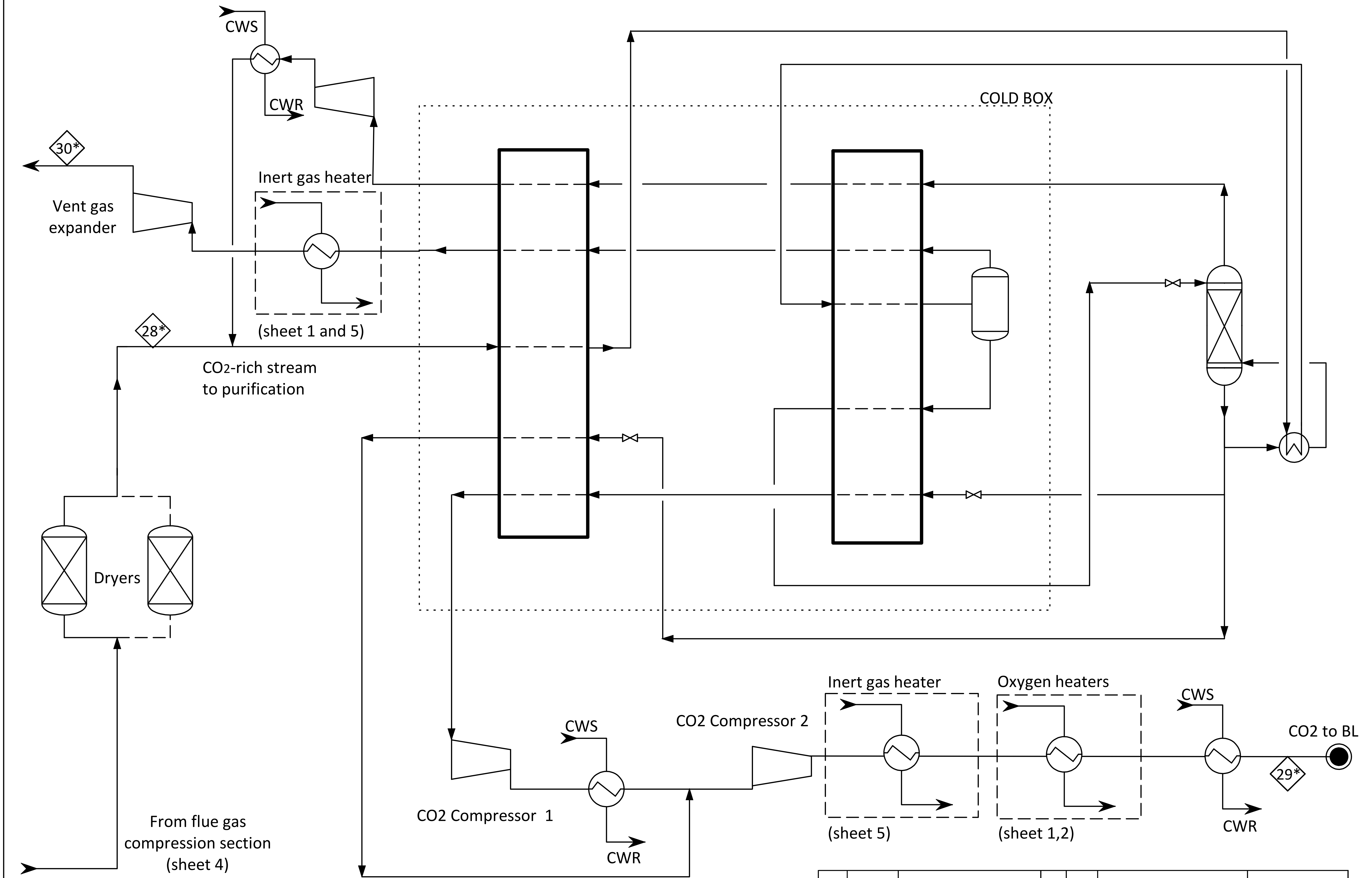
0	mar-15	Final report issue	NF	LM	**UNIT:** Oxy cycle: LPT	
Rev.	Date	Comment	By	Appr	**CASE:** 4c - CES CYCLE	Sheet 02 of 05



0	mar-15	Final report issue	NF	LM	**UNIT:** Oxy cycle: Condenser and wet gas compressor	
Rev.	Date	Comment	By	Appr	**CASE:** 4c - CES CYCLE	Sheet 03 of 05



0	mar-15	Final report issue	NF	LM	**UNIT:** CO2 purification - compression section	
Rev.	Date	Comment	By	Appr	**CASE:** 4c - CES CYCLE	Sheet 01 of 05



0	mar-15	Final report issue	NF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** 4c - CES CYCLE	Sheet 01 of 05

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

FOSTER WHEELER	Case 4c - Supercritical CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0	1	
	CLIENT :	IEAGHG	PREP.	NF		
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM		
	PROJECT NO:	1-BD-0764 A	APPROVED	LM		
	LOCATION:	The Netherlands	DATE	May 2015		

HEAT AND MATERIAL BALANCE

STREAM	1	2	3	4	5	6	7	8	9	10
	Natural gas from BL	NG to HP combustor	NG to MP combustor	NG to LP combustor	Air to ASU	Oxygen from ASU	Oxygen to HP combustor	Oxygen to MP combustor	Oxygen to LP combustor	steam to HP combustor
Temperature (°C)	15	150	140	130	9	15	215	120	120	650
Pressure (bar)	70	310	69	10	amb	30	310	65	10	340
TOTAL FLOW										
Mass flow (kg/h)	59,470	13,875	19,370	26,225	1,010,840	231,720	54,675	75,485	101,560	380,120
Molar flow (kmol/h)	3,300	770	1,075	1,455	35,050	7,214	1,702	2,350	3,162	21,100
LIQUID PHASE										
Mass flow (kg/h)									278700	
GASEOUS PHASE										
Mass flow (kg/h)	59,470	13,875	19,370	26,225	1,010,840	231,720	54,675	75,485	101,560	380,120
Molar flow (kmol/h)	3,300	770	1,075	1,455	35,050	7,214	1,702	2,350	3,162	21,100
Molecular Weight (kg/kmol)	18.0	18.0	18.0	18.0	28.8	32.1	32.1	32.1	32.1	18.0
Composition (%mol)	as assigned	as assigned	as assigned	as assigned						
Ar					0.92%	2.00%	2.00%	2.00%	2.00%	-
CO_2					0.04%	0.00%	0.00%	0.00%	0.00%	-
H_2O					0.97%	0.00%	0.00%	0.0%	0.0%	100.00%
N_2					77.32%	1.00%	1.00%	1.00%	1.00%	-
O_2					20.75%	97.00%	97.00%	97.00%	97.00%	-
Total					100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 4c - Supercritical CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0		
	CLIENT :	IEAGHG	**PREP.**	NF		
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	LM		
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM		
	LOCATION:	The Netherlands	**DATE**	May 2015		

HEAT AND MATERIAL BALANCE

STREAM	11	12	13	14	15	16	17	18	19	20
	HPT coolant stream	Flue gas to HPT	HPT exhaust flow	HPT exhaust flow to MP combustor	MPT coolant stream	Flue gas to MPT	MPT exhaust flow	MPT exhaust flow to MP combustor	LPT coolant stream	Flue gas to LPT
Temperature (°C)	475	1150	740	510	510	1533	790	420	420	1533
Pressure (bar)	340	300	60	59.5	59.5	58.5	8.2	8.0	8	7.6
TOTAL FLOW										
Mass flow (kg/h)	58,330	448,670	507,000	268,335	238,665	363,190	601,855	362,600	239,255	490,385
Molar flow (kmol/h)	3,240	23,610	26,850	14,210	12,640	17,685	30,325	18,270	12,055	22,955
LIQUID PHASE										
Mass flow (kg/h)										
GASEOUS PHASE										
Mass flow (kg/h)	58,330	448,670	507,000	268,335	238,665	363,190	601,855	362,600	239,255	490,385
Molar flow (kmol/h)	3,240	23,610	26,850	14,210	12,640	17,685	30,325	18,270	12,055	22,955
Molecular Weight (kg/kmol)	18.0	19.0	18.9	18.9	18.9	20.5	19.8	19.8	19.8	21.4
Composition (%mol)										
Ar	-	0.14%	0.13%	0.13%	0.13%	0.37%	0.27%	0.27%	0.27%	0.49%
CO_2	-	3.54%	3.11%	3.11%	3.11%	9.09%	6.60%	6.60%	6.60%	12.13%
H_2O	100.00%	96.01%	96.50%	96.50%	96.50%	89.91%	92.65%	92.65%	92.65%	86.65%
N_2	-	0.10%	0.09%	0.09%	0.09%	0.26%	0.19%	0.19%	0.19%	0.34%
O_2	-	0.20%	0.18%	0.18%	0.18%	0.38%	0.29%	0.29%	0.29%	0.39%
Total	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

	Case 4c - Supercritical CES Cycle - HEAT AND MATERIAL BALANCE		REVISION	0		
	CLIENT :	IEAGHG	**PREP.**	NF		
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	LM		
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM		
	LOCATION:	The Netherlands	**DATE**	May 2015		

HEAT AND MATERIAL BALANCE

STREAM	21	22	23	24	25	26*	27*	28*	29*	30*
	LPT exhaust flow	Exhaust flow to condeser	Wet flue gas from condenser	Condensate from condenser	BFW recycle to coolant stream generation	Flue gas to CPU	Waste water to WWT	CO2-rich stream to purification	CO2 to BL	Inert gas stream
Temperature (°C)	560	58	29	29	32	26	30	28	30	80
Pressure (bar)	0.21	0.21	0.21	0.21	350	1.05	2.5	33	110	1.1
TOTAL FLOW										
Mass flow (kg/h)	729,640	729,640	187,435	557,060	438,450	172,576	242,075	340,290	284,970	55,320
Molar flow (kmol/h)	35,010	35,010	4,910	30,920	24,340	8,170	13,440	7,900	6,476	1,424
LIQUID PHASE										
Mass flow (kg/h)		32,590		557,060	438,450		242,075			
GASEOUS PHASE										
Mass flow (kg/h)	729,640	697,050	187,435			172,576		340,290	284,970	55,320
Molar flow (kmol/h)	35,010	33,200	4,910			8,170		7,900	6,476	1,424
Molecular Weight (kg/kmol)	20.8	21.0	38.2			21.1		43.1	44.0	38.8
Composition (%mol)										
Ar	0.41%	0.43%	2.94%	-	-	0.52%	-	3.65%	0.16%	19.47%
CO_2	10.23%	10.78%	72.90%	-	-	12.96%	-	90.64%	99.83%	48.89%
H_2O	88.72%	88.10%	19.56%	100.0%	100.0%	85.73%	100.0%	-	-	-
N_2	0.29%	0.31%	2.07%	-	-	0.37%	-	2.57%	0.00%	14.26%
O_2	0.36%	0.37%	2.53%	-	-	0.42%	-	3.14%	0.01%	17.39%
Total	100.00%	100.00%	100.00%	100.0%	100.0%	100.00%	100.0%	100.00%	100.00%	100.00%

NOTE
1. Streams marked up with * correspond to the total flow of two trains. The remaining figures are referred to single train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Water consumption summary, reported in Table 2;
- Electrical consumption summary, shown in Table 3.

Table 2. Case 4c – Water consumption summary

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power pl	DATE	Apr-15
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Case 4 - CES cycle

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water [DT = 11°C] [t/h]
	OXY-TURBINE CYCLE			
	Condensate and recycle water system			
	Condenser		5	55,250
	Turbine and generator Auxiliaries			6,000
	Condenser vent compressor after cooler			2,700
	AIR SEPARATION UNIT			
	MAC intercoolers			9,530
	BAC intercoolers			1,370
	Oxygen compressor intercoolers			0
4000	**CO_2 PURIFICATION UNIT**			
	CO2 purification unit			3,470
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	1,400		
	Demineralized water unit	8	-5	
	Waste Water Treatment and Condensate Recovery	-230		
	Balance of plant			
	BALANCE	1,178	0	78,320

Note: (1) Minus prior to figure means figure is generated

Table 3. Case 4c – Electrical consumption summary

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Apr-15
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 4 - CES cycle

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
3000	**OXY-TURBINE CYCLE**	
	Natural gas compressor	**2,080**
	Condensate and recycle water system	**11,580**
	Flue gas compression	**14,000**
	Turbine Auxiliaries + generator losses	**5,470**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**131,580**
	Booster air compressor and miscellanea	**20,100**
	HP Oxygen compressor	**5,780**
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	**26,600**
	Autorefrigerated inerts removal unit compression consumption	**14,010**
	Autorefrigerated inerts removal unit expander production	**-2,910**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**9,800**
	Balance of plant	**1,460**
	BALANCE	**239,550**

6. Overall performance

The following table shows the overall performance of Case 4c, including CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Apr-15
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Case 4 - CES cycle

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	118.9
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**1701**
HP turbine power output (@ gen terminals)	MWe	194.1
MP turbine power output (@ gen terminals)	MWe	264.3
LP turbine power output (@ gen terminals)	MWe	534.1
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**992.5**
Oxy-turbine cycle (including NG compressor)	MWe	33.1
Air separation unit + Oxygen compressor	MWe	157.5
CO2 purification and compression unit	MWe	37.7
Utility & Offsite Units	MWe	11.3
ELECTRIC POWER CONSUMPTION	**MWe**	**239.6**
NET ELECTRIC POWER OUTPUT	**MWe**	**752.9**
(Step Up transformer efficiency = 0.997%) (B)	MWe	750.7
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**64.6%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**48.9%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**58.4%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**44.3%**
Equivalent CO_2 flow in fuel	kmol/h	7159
Captured CO_2	kmol/h	6464
CO_2 removal efficiency	**%**	**90.3**
Fuel Consumption per net power production	**MWth/MWe**	**2.05**
CO_2 emission per net power production	**kg/MWh**	**40.8**

6.1. Comparison with CES performance data

Scope of this section is to compare the results of the case study with those claimed by CES for the power plant having same configuration as the present study case.

The following Table 4 summarises the cycle performance, in terms of electrical efficiency, and the main design features potentially affecting the performance that differs from the CES simulation and the simulation based on the GS code results for the gas turbine.

The main design modifications implemented in the study case 4c are described below, for each cycle main component/parameter.

HPT

The HP combustor outlet temperature of 1150°C requires cooling of the gas turbine blade, as the metal temperature is kept below 860°C, while the CES model do not consider any cooling requirement. As described above, part of the steam from the 2nd heat recovery exchanger is diverted and used for this purpose; the cooling medium required is around 13% of the combustor outlet stream (weight basis).

In addition to the higher steam generation required, using a cooling stream for the HPT blades implies a lower turbine outlet temperature, mainly affecting the final steam generation temperature.

MPT and LPT cooling requirements

The cooling requirements for both the MP and LP turbine blades as estimated by the GS code are higher than the ones the model from CES.

As the available cooling medium is part of the cooled flue gas exhaust from the upstream expansion stage, the increased cooling requirement corresponds to a higher portion of flue gas that bypasses the MP and LP combustors. Therefore, the amount of combustion gas to be heated-up in the combustors is lower, thus lowering also the amount of fuel fed to the second and third combustors. The resulting natural gas to the first combustor increases from around 17% to around 23% of the total amount of fuel.

LP combustor pressure

The LP combustor pressure is increased in the study case with respect to the CES simulation.

The following has been considered in the selection of the combustor pressure and consequently of the MPT outlet and LPT inlet pressure.

Considering the assigned combustor exit temperature of 1530°C, the minimum pressure required for coolant injection is around 5 bar, while 8 bar is the minimum pressure in line with the pressure profile of the system. In fact, at lower pressure, the pressure drop through the injection channel is higher than the pressure drop across the LP combustor, implying a loss of expansion work (see also section 2.2.1).

LPT discharge pressure

As a consequence of the higher cooling requirement and also of the increased LP combustor pressure that implies a higher expansion ratio through the LPT, the turbine outlet temperature is lower at constant discharge pressure in the study case simulation with respect to the CES simulation. As for that, in order to have a proper temperature profile for steam generation in the downstream heat exchanger, the LPT outlet pressure is increased up to 21 kPa.

This implies a lower LPT power production but also a lower consumption of the downstream exhaust gas compressor.

Steam generation temperature profile

The increased cooling requirement as estimated by the GS code implies a reduced outlet temperature from each gas turbine section. Consequently, the temperature profile of the steam generated in the three heat recovery sections is lower, the maximum achievable temperature being around 650°C.

On the other hand, the heat available at lower temperature allows heating the BFW without any upstream heat recovery. As for that, the exhaust gas recompression configuration (mainly the number of stages) downstream the condenser is selected in order to minimise power consumptions as no heat recovery is required.

In addition, heat recovery in the CPU is exploited for natural gas and oxygen pre-heating before being injected in the combustion, enhancing plant efficiency.

Oxygen delivery pressure

The study is based on pumped oxygen ASU capable to deliver oxygen at 65 bar as required by the second combustor. In fact, the power consumptions and cost saving related to the oxygen compression from 11 bar to 65 bar for both the amount of oxygen fed to the first and second combustor, more than offset the additional power demand of the air compressors, related to the higher oxygen delivery pressure.

Table 4. Case 4c – Performance comparison

	Case 4c	CES cycle (as provided by CES)
Performance		
Plant net electrical efficiency	48.9%	53.1%
Design parameters		
Cooling stream (fraction of flue gas from combustor, %wt)		
First combustor	13%	NONE
Second combustor	65%	28%
Third combustor	49%	28%
NG to 1st reactor	23%	17%
LPT inlet pressure	7.6 bar	2.95 bar
HPT outlet temperature	740°C	808°C
MPT outlet temperature (*)	790°C	718°C
LPT outlet temperature (*)	560°C	666°C
Condensation pressure	21 kPa	14 kPa

(*) Affected by the different outlet pressure

Based on the description and the parameters summarized above, it can be drawn that the main reason of the lower electrical efficiency is the increased cooling requirements for the gas turbine blades. In fact, this implies a lower temperature profile mainly affecting the steam generation temperature and an increased natural gas fraction to the first combustor, which lower the overall cycle efficiency. It has to be noted that adopting a lower coolant temperature would be likely more advantageous and is currently being pursued by CES as part of their on-going cycle optimization work.

7. Environmental impact

The oxy-combustion gas turbine plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 5 summarizes the expected flow rate and composition of the inerts vent. Minor and fugitive emissions are related to seal losses in the power island and in the CO_2 purification unit.

Table 5. Case 4a – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	55,320
Flow, Nm^3/h	31,920
Composition	(%mol)
Ar	19.47%
N_2	14.26%
O_2	17.39%
CO_2	48.89%
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The process units do not produce significant liquid waste as the blow-downs (e.g. from the flue gas condenser / compressor intercoolers and CO_2 purification unit) are treated to recover water, so the main liquid effluent is the cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids.

Cooling Tower blowdown

Flowrate : 335 m^3/h

7.3. Solid effluents

The plant does not produce significant solid waste.

8. Equipment list

The list of main equipment and process packages is included in this section.

CLIENT: IEAGHG
LOCATION: The Netherlands
PROJ. NAME: Oxy-turbine power plants
CONTRACT N.: 1-BD-0764 A
CASE.: 4c - Supercritical CES cycle

REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
DATE	May-15			
ISSUED BY	NF			
CHECKED BY	LM			
APPROVED BY	LM			

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	GAS TURBINE PACKAGE							
PK- 3101-1/2	**Gas Turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
T- 3101	HP expander		100 MWe Pin: 300 bar; Pout 60					*Including:* *Lube oil system*
T- 3102	MP expander		135 MWe Pin: 58.5 bar; Pout: 8.2 bar					*Cooling system* *Idraulic control system* *Seals system*
T- 3102	LP expander		270 MWe Pin: 7.6 bar; Pout: 0.21 bar					*Drainage system*
G- 3101	Oxy turbine generator							*Including relevant auxiliaries*
F- 3101	HP Combustor		*180 MWt*					*One per train, two in total*
F- 3102	MP Combustor		*250 MWt*					*One per train, two in total*
F- 3103	LP Combustor		*340 MWt*					
K- 3102-1/2	**NG compressor**		Flowrate: 17,500 Nm3/h Pin: 70 bar; Pout: 310 bar Compression ratio: 4.4	*1250 kWe*				*One per train, two in total*
	HEAT RECOVERY SECTION and BFW SYSTEM							
PK- 3201-1/2	**Heat recovery section**							***2 x 50% package***
	Each including:							
E- 3201 A	BFW economisers							
E- 3202 A/B	Steam superheater							
E- 3203	Inert gas heater							
E- 3204	Regenerator heater							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	BFW pumps	Centrifugal	250 m3/h x 3550 m each	3250 kW each				*Two operating one spare, per each train*
	DRUM							
D- 3201	Deaerator							

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4c - Supercritical CES cycle		

REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
DATE	May-15			
ISSUED BY	NF			
CHECKED BY	LM			
APPROVED BY	LM			

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
FLUE GAS CONDENSER PACKAGE and COMPRESSION PACKAGE								
PK- 3301-1/2	**Condenser Package**							***2 x 50% condenser package***
	Each including:							
	Flue gas condenser		350 MWth					*Including condenser hotwell*
	PUMPS		**Q [m3/h] x H [m]**					
P- 3301 A/B	Flue gas condensate pump	Centrifugal	600 m3/h x 50 m	110 kW				*One operating one spare, per each train*
PK- 3303-1/2	**Wet flue gas compressors**							***2 x 50% compression train***
	- Wet flue gas compressor #1	axial	Flowrate: 2 x 55,000 Nm3/h Pin: 21 kPa; Pout : 31.5 kPa Compression ratio: 1.5	*2 x 1,000kW*				*Two operating per each train*
	- Wet flue gas compressor #2	axial	Flowrate: 2 x 50,000 Nm3/h Pin: 29.5 kPa; Pout : 46 kPa Compression ratio: 1.56	*2 x 1,000kW*				*Two operating per each train*
	- Wet flue gas compressor #3	axial	Flowrate: 2 x 48,500 Nm3/h Pin: 42 kPa; Pout : 69 kPa Compression ratio: 1.64	*2 x 1,100kW*				*Two operating per each train*
	- Wet flue gas compressor #3	axial	Flowrate: 2 x 47,000 Nm3/h Pin: 65 kPa; Pout : 109 kPa Compression ratio: 1.68	*2 x 1,100kW*				*Two operating per each train*
	Condesate separators							
	Intercoolers							
	CW cooler #1							
	CW cooler #2							
	CW cooler #3							
	CW cooler #4							

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4c - Supercritical CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
FOSTER WHEELER		DATE	May-15			
		ISSUED BY	NF			
		CHECKED BY	LM			
		APPROVED BY	LM			

EQUIPMENT LIST

Unit 4000 - CO2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	PACKAGES							
PK - 4001	**CO2-rich gas compression**							
	Including:							
	Raw flue gas compressors							
	- Raw flue gas compressor #1	axial	Flowrate: 2 x 92,000 Nm3/h Pin: 1.05 bar; Pout : 4.2 bar Compression ratio: 4.0	*2 x 5.75 MWe*				*2x50%*
	- Raw flue gas compressor #2	axial	Flowrate: 2 x 92,500 Nm3/h Pin: 3.8 bar; Pout : 15.2 bar Compression ratio: 4.0	*2 x 5.75 MWe*				*2x50%*
	- Raw flue gas compressor #3	axial	Flowrate: 2 x 100,000 Nm3/h Pin: 14.4 bar; Pout : 35 bar Compression ratio: 2.44	*2 x 3.5 MWe*				*2x50%*
	Condesate separators							
	Intercoolers							
	Natural gas heater #1/#2							
	CW cooler #1 / #2							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO2 liquid separator							
	- CO2 distillation column							
	- CO2 compressor 1	centrifugal	Flowrate: 2 x 36,500 Nm3/h	*2 x 1.5 MWe*				*2x50%*
	- CO2 compressor 2	centrifugal	Flowrate: 2 x 73,000 Nm3/h	*2 x 6 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		3100 kW					*1x100%*
	- Overhead recycle compressors (optional)							
	- Intercoolers							
	Inert gas heater							
	Oxygen heaters							
	Cooling water intercoolers							

CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
LOCATION: The Netherlands	DATE	May-15			
PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF			
CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM			
CASE.: 4c - Supercritical CES cycle	APPROVED BY	LM			

EQUIPMENT LIST

Unit 5000 - Air Separation Unit (2 x 50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		2 x 5560 t/d					*2x50% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 36.5 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		10 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
	- Oxygen Compressors	Centrifugal		3.5 MWe				
TK - 5001	LOX storage tank		700 t					*Common to both trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	May-15			
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF			
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM			
	CASE.: 4c - Supercritical CES cycle	APPROVED BY	LM			

EQUIPMENT LIST

Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower** including: Cooling water basin	Natural draft	990 MWth Diameter: 120 m, Height: 180 m, Water inlet: 17 m				concrete	
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/…/F	**Cooling Water Pumps**	Centrifugal	14,000 m3/h x 40 m	1780 kW				*Six in operation, one spare*
P- 6003 A/B	**Cooling tower make up pumps**	Centrifugal	1,500 m3/h x 30 m	160 kW				
	PACKAGES **Cooling Water Filtration Package** Cooling Water Sidestream Filters		7830 m3/h					
	Sodium Hypochlorite Dosing Package Sodium Hypochlorite storage tank Sodium Hypochlorite dosage pumps							
	Antiscalant Package Dispersant storage tank Dispersant dosage pumps							
	NATURAL GAS RECEIVING SYSTEM							
PK- 6001	Metering station							
PK- 6002	Let down station							
	RAW WATER SYSTEM							
PK- 6003	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Portable water pumps	centrifugal						*One operating, one spare*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: 4c - Supercritical CES cycle	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	May-15			
		ISSUED BY	NF			
		CHECKED BY	LM			
		APPROVED BY	LM			

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	DEMINERALIZED WATER SYSTEM							
PK- 6004	Demin Water Package, including: - Multimedia filter - Reverse Osmosis (RO) Cartidge filter - Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Plant and instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	ECONOMICS
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	F. FURESI
CHECKED BY	:	N. FERRARI
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENTS

1. Introduction

The purpose of this chapter is to present the results of the economic analysis, carried out to evaluate the Levelized Cost of Electricity (LCOE) and the CO_2 Avoidance Cost (CAC) of the study cases.

Capital cost and operating & maintenance (O&M) costs for the different cases have been evaluated and are presented in this chapter, along with the results of the financial model.

All economical inputs used to perform this analysis are set in accordance with the economic bases reported in chapter B.

Due to the possible floating of some economic input data, an exhaustive sensitivity analysis is also performed and presented in this chapter on key parameters such as:

- Cost of novel equipment,
- Natural Gas cost,
- Plant life (project duration),
- Discount rate,
- Costs related to CO_2 emission or transport & storage,
- Operating capacity factor of the plant.

A full economical assessment is made for all the main study cases listed in the Table 1, including the conventional air-blown combined cycle without CCS, used as reference case, and the four oxy-turbine power plants described in the previous chapters of the report.

Table 1 - Study cases

Case	Description
Case 0 (reference)	Conventional air-blown combined cycle
Case 1	Semi-closed oxy-combustion combined cycle (SCOC-CC)
Case 2	NET Power cycle – single combustor
Case 3	Modified S-Graz cycle
Case 4b	Revised CES cycle

For the S-GRAZ cycle and the CES cycle, different alternative configurations have been technically assessed, while the economic and financial analyses are performed for the most promising configuration only.

Based on the technical assessment of the S-Graz cycle and the Modified S-Graz cycle (Chapter D.3 and D.4 of the report), the latter shows a slightly lower electrical efficiency; however, the investment cost required by the S-Graz cycle is expected to be higher, due to the last stages of the gas turbine that expands the flue gas down to vacuum conditions and the very large volume and exchanger surface of the flue gas vacuum condenser. These equipment are not included in the modified S-Graz cycle power plant, which uses a more standard condensing steam turbine. The lower investment cost, along with the lower need for technology development, are the main drivers for the selection of the Modified S-Graz cycle alternative (identified as case 3) for the financial analysis.

For the CES cycle, the main financial analysis is based on the supercritical CES cycle (Chapter D.7), as the most efficient configuration. This case (identified as case 4 in this chapter E) corresponds to the long-term solution proposed by CES for their oxy-turbine cycle, which requires almost the same technology development as other cycle configurations included in the financial analysis, e.g. NET Power scheme.

The improvement in the economics resulting from the long-term development of the supercritical CES cycle, with respect to the short-term solution described in the technical chapters of the report, is shown as sensitivity case. In particular, comparison is shown with the revised CES cycle configuration (Chapter D.6) as it is based on a more conventional gas turbine design (and consequently lower investment cost at constant efficiency), expanding the flue gas down to atmospheric pressure and not to vacuum condition as in the originally proposed configuration.

All the technical features of the oxy-turbine cycle cases are given in the previous chapters of the report, while for the conventional combined cycle reference shall be made to the IEAGHG report 2012/08 '*CO_2 capture at gas fired power plants*'. The following sections provide the results of the economical modelling only.

2. Capital cost

2.1. Definitions

The main cost estimating bases are described in chapter B. This section provides details on the Total Capital Requirement (TCR), also named as Total Investment Cost (TIC), of the various study cases.

TCR is defined in general accordance with the White Paper "*Toward a common method of cost estimation for CO_2 capture and storage at fossil fuel power plants*", (March 2013), produced collaboratively by authors from EPRI, IEAGHG, Carnegie Mellon University, MIT, IEA, GCCSI and Vattenfall.

The **Total Capital Requirement** (**TCR**) is defined as the sum of:

- Total Plant Cost (**TPC**)
- Interest during construction
- Spare parts cost
- Working capital
- Start-up costs
- Owner's costs.

The Total Plant Cost (TPC) is the installed cost of the plant, including contingencies.

The TPC of the different study cases is presented in the following sections, broken down into the following main process units:

- Reference case:
  - Combined cycle
  - Utility units
- Oxy-turbine cases:
  - Power island
  - CO_2 purification unit (CPU)
  - Air separation unit (ASU)
  - Utility units

Moreover, for each process unit, the TPC is split into the following items, as further discussed in the next sections:

- Direct materials
- Construction and other costs
- EPC services
- Contingency.

2.2. Estimating methodology

The estimate is an AACE Class 4 estimate (accuracy range ±35%), based on 2Q2014 price level, in euro (€).

2.2.1. Total Plant Cost

The estimating methodology used by FW for the evaluation of the Total Plant Cost (TPC) items of the process units is described in the following sections.

Direct materials

The oxy-turbine power plants include novel equipment that are either under development or at conceptual stage only, so for estimating purpose they should be considered as first of a kind (FOAK) plants. The study, however, has investigated the potential capabilities of the oxy-turbine power plants with respect to benchmark technologies for capture of the CO_2, these latter generally assumed as ready for commercial application. Therefore, the study has treated the oxy-turbine cycles as N^{th}-of-a-kind (NOAK) plants for estimating purposes and has evaluated the cost of novel equipment as already developed and suitable for large-scale commercial application. Nevertheless, sensitivity to the cost of this equipment has been performed to take into account, to a certain extent, the intrinsic uncertainty of such estimates.

For the other equipment and for each different process unit, direct materials are estimated using company in-house database or conceptual estimating models. Where a detailed and sized equipment list has been developed, a K-base (commercially available software) run has been made for the equipment estimate. For units having capacity only, cost is based on previous estimates done for similar units, by scaling up or down (as applicable) the cost on capacity ratio.

Construction and EPC services

For each unit or block of units, construction and EPC services are factored on the direct materials costs; factor multipliers are based on FW in-house data from cost estimates made in the past for other power plants.

Other costs

Other costs mainly include:

- Temporary facilities;
- Freight, taxes and insurance;
- License fees.

Temporary facilities, freight, taxes, insurance and license fees are estimated as a percentage of the construction cost, in accordance with Foster Wheeler experience and in-house data bank.

Contingency

A project contingency is added to the capital cost to give a 50% probability of a cost over-run or under-run. For the accuracy considered in this study, FW's view is that contingency should be in the range of 10-15% of the total installed cost. 10% is assumed for this study for all the different units of the plant, for consistency with the other IEAGHG studies.

A process contingency is not added to the plant cost because, as written before, plants are considered as N^{th}-of-a-kind (NOAK) plants.

2.2.2. *Total Capital Requirement*

Total Capital Requirement (TCR) is the sum of the TPC and following items:

- Interest during construction, assumed to be the same as the discount rate (8%).
- Spare parts cost, assumed as 0.5% of TPC.
- Working capital, including 30 days inventories of chemicals.
- Start-up costs, assumed as 2% of TPC, plus 25% of fuel cost for one month, plus 3 months O&M costs and 1 month of catalyst, chemicals etc.
- Owner's costs, assumed as 7% of TPC.

Further details on the above cost items are shown in chapter B of the report.

2.3. Total Plant Cost summary

Table 3 to Table 7 show the TPC of the different study cases listed in Table 1. Each table is followed by the related pie chart of the total plant cost to show the percentage weight of each unit on the overall capital cost of the plant.

Total Plant Cost and Total Capital Requirement figures for the different cases are also reported in the below Table 2 for summary purpose.

For the power production cases, the specific costs, defined as the ratio between either the TPC or the TCR and the net power output, are also reported.

Table 2. TPC and TCR of study cases

Case	**Total Plant Cost (TPC)** (M€)	**Total Capital Requirement (TCR)** (M€)	**Specific cost** [TPC/Net Power] (€/kW)	**Specific cost** [TCR/Net Power] (€/kW)
Case 0 (CCGT) (reference)	592	773	655	855
Case 1 (SCOC-CC)	1,111	1,441	1,470	1,905
Case 2 (Net Power)	1,118	1,451	1,320	1,715
Case 3 (Mod. S Graz)	1,136	1,474	1,500	1,950
Case 4 (Sup.crit CES)	1,160	1,505	1,540	2,000

Table 3. Case 0 – Total Plant Cost

FOSTER WHEELER	OXY-TURBINE POWER PLANT CASE 0 - CONVENTIONAL AIR-BLOWN COMBINED CYCLE				CONTRACT: 1-BD-0764A CLIENT: IEAGHG LOCATION: THE NETHERLANDS DATE: JUNE 2014 REV.: 0
POS.	**DESCRIPTION**	**UNIT 3000** **COMBINED CYCLE**	**UNIT 6000** **UTILITY UNITS**	**TOTAL COST** **EURO**	**NOTES / REMARKS**
1	DIRECT MATERIAL	216,300,000	77,000,000	293,300,000	1) Gross power output: MW 928 Specific cost €/KW : 638
2	CONSTRUCTION and OTHER COSTS	159,300,000	38,400,000	197,700,000	2) Total Net Power : MW 904
3	EPC SERVICES	31,800,000	15,400,000	47,200,000	Average Cost €/KW : 655
4	**TOTAL INSTALLED COST**	**407,400,000**	**130,800,000**	**538,200,000**	
5	PROJECT CONTINGENCY	40,700,000	13,100,000	53,800,000	
6	PROCESS CONTINGENCY	-	-	-	**BUSINESS CONFIDENTIAL**
7	**TOTAL PLANT COST (TPC)**	**448,100,000**	**143,900,000**	**592,000,000**	

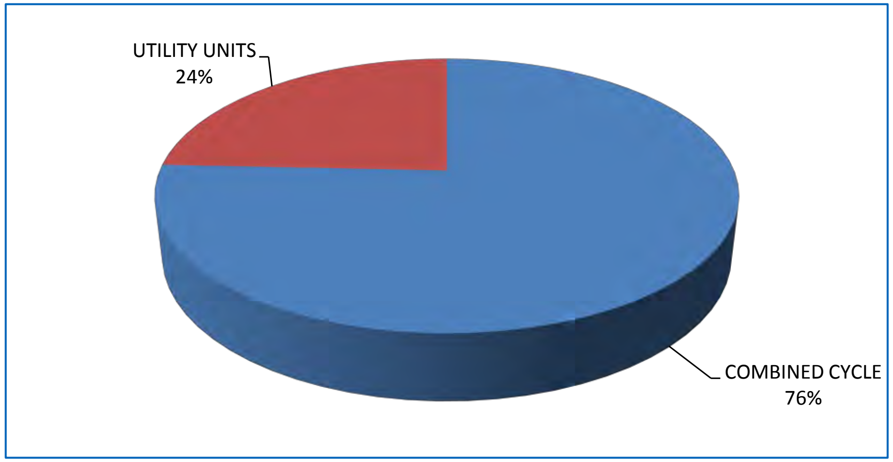


Figure 1. Case 0 – Unit percentage weight on TPC

Table 4. Case 1 – Total Plant Cost

OXY-TURBINE POWER PLANT
CASE 1 - SCOC-COMBINED CYCLE

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 COMBINED CYCLE	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	232,900,000	64,200,000	169,700,000	109,000,000	575,800,000	1) Gross power output: MW 968.0 Specific cost €/kW : 1,150
2	CONSTRUCTION and OTHER COSTS	171,600,000	20,800,000	94,800,000	54,400,000	341,600,000	2) Total Net Power : MW 756.8
3	EPC SERVICES	34,200,000	11,600,000	25,000,000	21,600,000	92,400,000	Specific cost €/kW : 1,470
4	**TOTAL INSTALLED COST**	**438,700,000**	**96,600,000**	**289,500,000**	**185,000,000**	**1,009,800,000**	
5	PROJECT CONTINGENCY	43,900,000	9,700,000	29,000,000	18,500,000	101,100,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**482,600,000**	**106,300,000**	**318,500,000**	**203,500,000**	**1,110,900,000**	

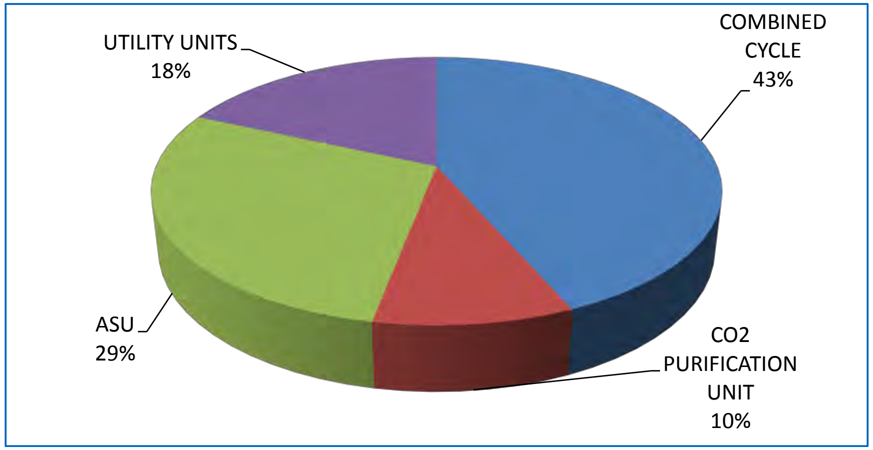


Figure 2. Case 1 – Unit percentage weight on TPC

Table 5. Case 2 – Total Plant Cost

OXY-TURBINE POWER PLANT
Case 2 - NET POWER

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 POWER and CO2 CYCLE	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	264,200,000	35,200,000	181,500,000	94,600,000	575,500,000	1) Gross power output: MW 1056.0
							Specific cost €/kW : 1,060
2	CONSTRUCTION and OTHER COSTS	194,600,000	11,400,000	98,500,000	47,200,000	351,700,000	
							2) Total Net Power : MW 845.9
3	EPC SERVICES	38,800,000	6,400,000	25,000,000	18,800,000	89,000,000	Specific cost €/kW : 1,320
4	**TOTAL INSTALLED COST**	**497,600,000**	**53,000,000**	**305,000,000**	**160,600,000**	**1,016,200,000**	
5	PROJECT CONTINGENCY	49,800,000	5,300,000	30,500,000	16,100,000	101,700,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**547,400,000**	**58,300,000**	**335,500,000**	**176,700,000**	**1,117,900,000**	

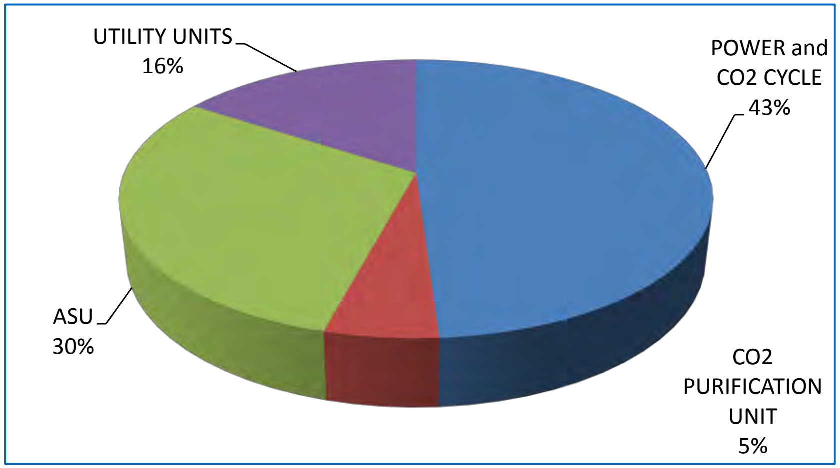


Figure 3. Case 2 – Unit percentage weight on TPC

Table 6. Case 3 – Total Plant Cost

FOSTER WHEELER	OXY-TURBINE POWER PLANT CASE 3 - MODIFIED S-GRAZ CYCLE					CONTRACT: 1-BD-0764A CLIENT: IEAGHG LOCATION: THE NETHERLANDS DATE: FEBRUARY 2015 REV.: 0
POS.	**DESCRIPTION**	**UNIT 3000** **POWER ISLAND**	**UNIT 4000** **CO2 PURIFICATION UNIT**	**UNIT 5000** **ASU**	**UNIT 6000** **UTILITY UNITS**	**TOTAL COST EURO**	**NOTES / REMARKS**
1	DIRECT MATERIAL	248,800,000	59,300,000	170,800,000	108,000,000	586,900,000	1) Gross power output: MW 995.2 Specific cost €/kW : 1,140
2	CONSTRUCTION and OTHER COSTS	183,300,000	19,300,000	95,900,000	53,900,000	352,400,000	2) Total Net Power : MW 755.5
3	EPC SERVICES	36,600,000	10,800,000	25,000,000	21,500,000	93,900,000	Specific cost €/kW : 1,500
4	**TOTAL INSTALLED COST**	**468,700,000**	**89,400,000**	**291,700,000**	**183,400,000**	**1,033,200,000**	
5	PROJECT CONTINGENCY	46,900,000	8,900,000	29,200,000	18,300,000	103,300,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**515,600,000**	**98,300,000**	**320,900,000**	**201,700,000**	**1,136,500,000**	

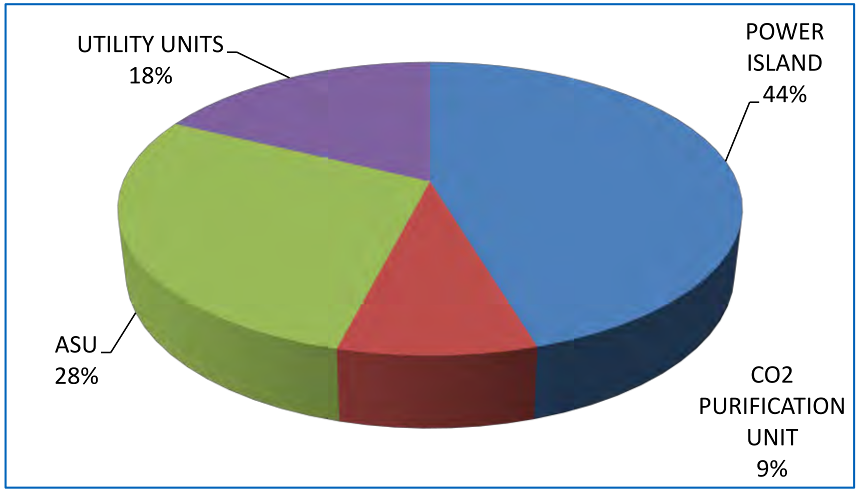


Figure 4. Case 3 – Unit percentage weight on TPC

Table 7. Case 4 – Total Plant Cost

	OXY-TURBINE POWER PLANT CASE 4 - SUPERCRITICAL CES CYCLE						CONTRACT: 1-BD-0764A CLIENT: IEAGHG LOCATION: THE NETHERLANDS DATE: APRIL 2015 REV.: 0
POS.	DESCRIPTION	UNIT 3000 POWER ISLAND	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	228,200,000	65,300,000	194,100,000	111,500,000	599,100,000	1) Gross power output: MW 992.5 Specific cost €/kW : 1,170
2	CONSTRUCTION and OTHER COSTS	168,100,000	21,100,000	115,800,000	55,700,000	360,700,000	2) Total Net Power : MW 750.7
3	EPC SERVICES	33,500,000	11,800,000	25,000,000	22,100,000	92,400,000	Specific cost €/kW : 1,540
4	**TOTAL INSTALLED COST**	**429,800,000**	**98,200,000**	**334,900,000**	**189,300,000**	**1,052,200,000**	
5	PROJECT CONTINGENCY	43,000,000	9,800,000	35,700,000	18,900,000	107,400,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**472,800,000**	**108,000,000**	**370,600,000**	**208,200,000**	**1,159,600,000**	

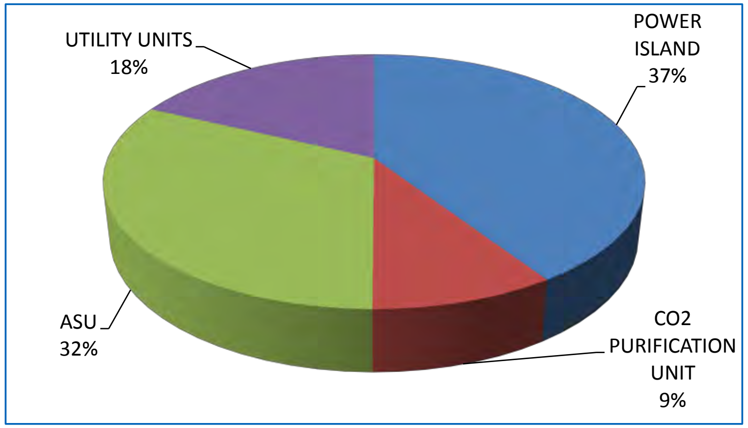


Figure 5. Case 4 – Unit percentage weight on TPC

3. Operating and Maintenance costs

The definition of the Operating and Maintenance (O&M) costs is given in chapter B of the report. Following sections provide estimated operating and maintenance costs for the different cases, which are generally allocated as:

- Variable costs;
- Fixed costs.

However, accurately distinguishing the variable and fixed costs is not always feasible. Certain cost items may have both variable and fixed components; for instance, the planned maintenance and inspection of the oxy-turbine, whose occurrence is based on number of running hours, should be allocated as variable component of maintenance cost, but for simplicity in the analysis all maintenance costs have been assumed to be fixed.

3.1. Variable costs

The following tables show the variable costs for the study cases listed in Table 1, including the following main cost items:

- Feedstock
- Raw water make-up
- Chemicals.

The consumption of the various items and the corresponding costs are yearly, based on the expected equivalent availability of the plant (90%). Reference values for natural gas and main consumables prices are summarized in the table below.

Item	Unit	Cost
Natural Gas	€/GJ (LHV)	8
Raw water	€/m^3	0.2
CO_2 transport and storage	€/t CO_2 stored	10
CO_2 emission cost	€/t CO_2 emitted	0

Table 8. Breakdown of variable costs

		Yearly Variable Costs														
											Revision:	0				
											Date:	April 2015				
											Issued by:	NF				
											Approved by:	LM				
Yearly Operating hours = 7884		Case 0 Base Case			Case 1 SCOC - CC			Case 2 NET Power			Case 3 Modified S-Graz			Case 4 Revised CES		
Consumables	**Unit Cost**	**Consumption**		**Operating Costs**	**Consumption**		**Operating Costs**	**Consumption**		**Operating Costs**	**Consumption**		**Operating Costs**	**Consumption**		**Operating Costs**
		Hourly	Yearly		Hourly	Yearly		Hourly	Yearly		Hourly	Yearly		Hourly	Yearly	
	€/t	kg/h	t/y	€/y	kg/h	t/y	€/y	kg/h	t/y	€/y	kg/h	t/y	€/y	kg/h	t/y	€/y
Feedstock																
Natural Gas	372.0	118,900	937,408	348,730,600	118,900	937,408	348,730,600	118,900	937,408	348,730,600	118,900	937,408	348,730,600	118,900	937,408	348,730,600
Auxiliary feedstock																
Make-up water	0.20	690,000	5,439,960	1,088,000	1,120,000	8,830,080	1,766,000	1,050,000	8,278,200	1,655,600	1,088,000	8,577,792	1,715,600	1,240,000	9,776,160	1,955,200
Chemicals	-	-	-	201,600	-	-	207,900	-	-	90,000	-	-	161,100	-	-	171,900
TOTAL YEARLY OPERATING COSTS	**Euro/year**			**350,020,200**			**350,704,500**			**350,476,200**			**350,607,300**			**350,857,700**

3.2. Fixed costs

Fixed costs include:

- Operating Labour Costs
- Overhead Charges
- Maintenance Costs.

3.2.1. *Operating Labour costs*

The plants of the different study cases can be virtually divided into the following main areas of operation:

- Power island
- Process unit (ASU and CPU) and utilities.

The same division is reflected in the design of the centralized control room, which has the same number of main DCS control groups, each one equipped with a number of control stations, from where the operation of the units of each area is controlled.

The area responsible and his assistant supervise each area of operation; both are daily positions. The shift superintendent and the electrical assistant are common for the different areas; both are shift positions. The rest of the operation staff is structured around the standard positions: shift supervisors, control room operators and field operators.

The maintenance personnel are based on large use of external subcontractors for all medium-major type of maintenance work. Maintenance costs take into account the service outsourcing. Plant maintenance personnel like the instrument specialists perform routine maintenance and resolve emergency problems.

The yearly cost of the direct labour is calculated assuming for each individual an average cost equal to 60,000 Euro/year, referred to year 2013.

The following tables report the labour force for the different configurations, along with the direct labour cost.

Table 9 – Case 0 - Operating Labor costs

Conventional CC			
	Combined cycle + utilities	TOTAL	Notes
OPERATION			
Area Responsible	1	1	daily position
Assistant Area Responsible	1	1	daily position
Shift Supervisor	5	5	2 positions per shift
Control Room Operator	10	10	2 positions per shift
Field Operator	10	10	2 positions per shift
Subtotal		27	
MAINTENANCE			
Mechanical group	4	4	daily position
Instrument group	4	4	daily position
Electrical group	3	3	daily position
Subtotal		11	
LABORATORY			
Superintendent+Analysts	2	2	daily position
Subtotal		2	
TOTAL		40	

Cost for personnel
Yearly individual average cost = 60,000 Euro/year
Total cost = 2,400,000 Euro/year

Table 10 – Case 1, 2, 3, 4 – Operating Labor costs

Oxy-turbine cycle				
	Oxy-turbine cycle	ASU+CPU Utilities	TOTAL	Notes
OPERATION				
Area Responsible	1	1	2	daily position
Assistant Area Responsible	1	1	2	daily position
Shift Superintendent	5		5	1 position per shift
Electrical Assistant	5		5	1 position per shift
Shift Supervisor	5	5	10	2 positions per shift
Control Room Operator	5	15	20	4 positions per shift
Field Operator	5	15	20	4 positions per shift
Subtotal			64	
MAINTENANCE				
Mechanical group	5		5	daily position
Instrument group	5		5	daily position
Electrical group	5		5	daily position
Subtotal			15	
LABORATORY				
Superintendent+Analysts	3		3	daily position
Subtotal			3	
TOTAL			82	

Cost for personnel
Yearly individual average cost = 60,000 Euro/year
Total cost = 4,920,000 Euro/year

3.2.2. *Overhead charges*

All other company services not directly involved in the operation of the plant fall in this category, such as:

- Management;
- Administration;
- Personnel services;
- Technical services.

These services vary widely from company to company and are also dependent on the type and complexity of the operation. It is assumed that this cost is equal to 30% of the operating labour and maintenance labour cost.

3.2.3. *Maintenance costs*

A precise evaluation of the cost of maintenance would require a breakdown of the costs amongst the numerous components and packages of the plant. Since these costs are all strongly dependent on the type of equipment selected and statistical maintenance data provided by the selected vendors, this type of evaluation of the maintenance cost is premature at study level.

For this reason the annual maintenance cost of the plant is estimated as a percentage of the Total Plant Cost of each case, as shown in the following:

- Novel technologies 2.5%
- Other units (ASU, CPU, steam turbines, utilities…) 1.5%

In general, estimates can be separately expressed as maintenance labour and maintenance materials. A maintenance labour to materials ratio of 40:60 can be statistically considered for this breakdown.

The yearly maintenance cost for all cases of the study is reported in the following Table 11, with reference to year 2014.

Table 11 – Maintenance costs (reference year: 2014)

Case	Total Plant Cost (M€)	Maintenance (M€/year)
Case 0	592	8.88
Case 1	1,111	18.28
Case 2	1,118	20.82
Case 3	1,136	18.97
Case 4	1,160	20.54

3.2.4. *Annual insurance and local taxes costs*

0.5% of the TPC is assumed to cover the insurance cost and to cover the local taxes and fees.

3.3. Summary

The following table reports the summary of the O&M costs for the different cases.

Table 12. O&M costs base cases

		Revision:	0	1	
		Date:	September 2014	April 2015	
		Issued by:	NF	NF	
		Approved by:	LM	LM	
O&M COSTS					
	Case 0 €/year	**Case 1** €/year	**Case 2** €/year	**Case 3** €/year	**Case 4** €/year
Fixed Costs					
Direct labour	2,400,000	4,920,000	4,920,000	4,920,000	4,920,000
Adm./gen overheads	1,785,600	3,670,000	3,974,900	3,752,700	3,940,200
Insurance & Local taxes	2,960,000	5,554,500	5,589,500	5,682,500	5,798,000
Maintenance	8,880,000	18,283,500	20,824,500	18,972,500	20,535,000
Subtotal	**16,025,600**	**32,428,000**	**35,308,900**	**33,327,700**	**35,193,200**
Variable Costs (Availability = 90%)					
Feedstock	348,730,600	348,730,600	348,730,600	348,730,600	348,730,600
Water Makeup	1,088,000	1,766,000	1,655,600	1,715,600	1,955,200
Chemicals	201,600	207,900	90,000	161,100	171,900
Subtotal	**350,020,200**	**350,704,500**	**350,476,200**	**350,607,300**	**350,857,700**
TOTAL O&M COSTS	**366,045,800**	**383,132,500**	**385,785,100**	**383,935,000**	**386,050,900**

4. Financial analysis

4.1. Objective of the economic modelling

The economic modelling is a simplified financial analysis that estimates, for each case, the Levelized Cost of Electricity (LCOE) and the CO_2 Avoidance Cost (CAC), based on specific macroeconomic assumptions.

The LCOE prediction is calculated under the assumption of obtaining a zero Net Present Value (NPV) for the project, corresponding to an Internal Rate of Return (IRR) equal to the Discount Rate (DR). Therefore, the financial analysis is a high-level economical evaluation only, while the rigorous project profitability for the specific case is beyond the scope of the present study.

4.2. Definitions

4.2.1. *Levelized Cost Of Electricity (LCOE)*

The Cost of Electricity (COE) in power production plants is defined as the selling price at which electricity must be generated to reach the break even at the end of the plant lifetime for a targeted rate of return.

However, with the purpose of screening different technology alternatives, the levelized value of the cost of electricity (LCOE) is commonly preferred to the year-by-year data. The LCOE is defined as the uniform annual amount which returns the same net present value as the year-by-year amounts.

In this analysis, long-term inflation assumptions and price/cost variations throughout the project life-time are not considered and, therefore, the COE matches with the LCOE.

4.2.2. *Cost of CO_2 avoidance*

For the power production cases, the CO_2 Avoidance Cost (CAC) is calculated by comparing the costs and specific emissions of a plant with CCS with those of the reference case without CCS. For a power generation plant, it is defined as follows:

$$CO_2 \text{ Avoidance Cost (CAC)} = \frac{LCOE_{CCS} - LCOE_{Reference}}{CO_2Emissions_{\ Reference} - CO_2Emissions_{\ CCS}}$$

where:

Cost of CO_2 avoidance is expressed in Euro per tonne of CO_2
LCOE is expressed in Euro per kWh
CO_2 emissions is expressed in tonnes of CO_2 per kWh.

The selected reference case for the evaluation of the CAC is Case 0, i.e. the conventional combined cycle without capture of the generated carbon dioxide.

4.3. Macroeconomic bases

The economic assumptions and macroeconomic bases are reported in chapter B of the report. These mainly include:

- Reference dates and construction period,
- Financial leverage,
- Discount rate,
- Interests during construction,
- Spare parts cost,
- Working capital,
- Start-up cost,
- Owner's cost,
- Insurance cost,
- Local taxes and fees,
- Decommissioning cost.

The principal financial bases assumed for the financial modelling are reported also hereafter for reader's convenience:

ITEM	DATA
Type of fuel	Natural Gas at 8 €/GJ (LHV)
Discount Rate	8%
Capacity factor	90%
CO_2 transport & storage cost	10 €/t $_{\text{STORED}}$
CO_2 emission cost	0 €/t $_{\text{EMITTED}}$
Inflation Rate	Constant Euro
Currency	Euro reported in 2Q2014

4.4. Financial analysis results

This section summarizes the results of the financial analysis performed for the study cases, based on the input data reported above.

Figure 6 and Figure 7 reports the LCOE and CAC for all study cases. LCOE figures also show the relative weight of:

- Capital investment,
- Fixed O&M,
- Variable O&M,
- Fuel,
- CO_2 transportation & storage.

A summary of the economical modelling results is also reported in the following Table 13.

Table 13. Financial results summary: LCOE and CO_2 avoidance cost

Case	Description	LCOE €/MWh	CAC €/t
Case 0 (reference)	CCGT	62.5	-
Case 1	SCOC-CC	92.8	97.9
Case 2	NET-POWER	83.6	67.6
Case 3	Modified S-Graz	93.7	101.4
Case 4	CES	95.1	106.0

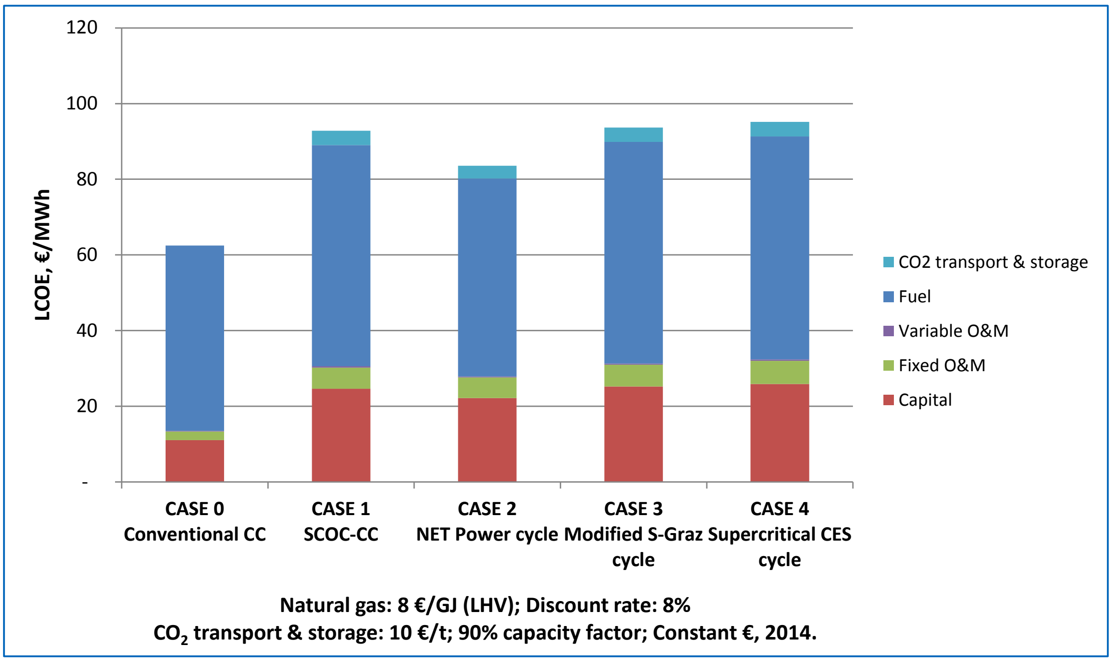


Figure 6. LCOE for base cases

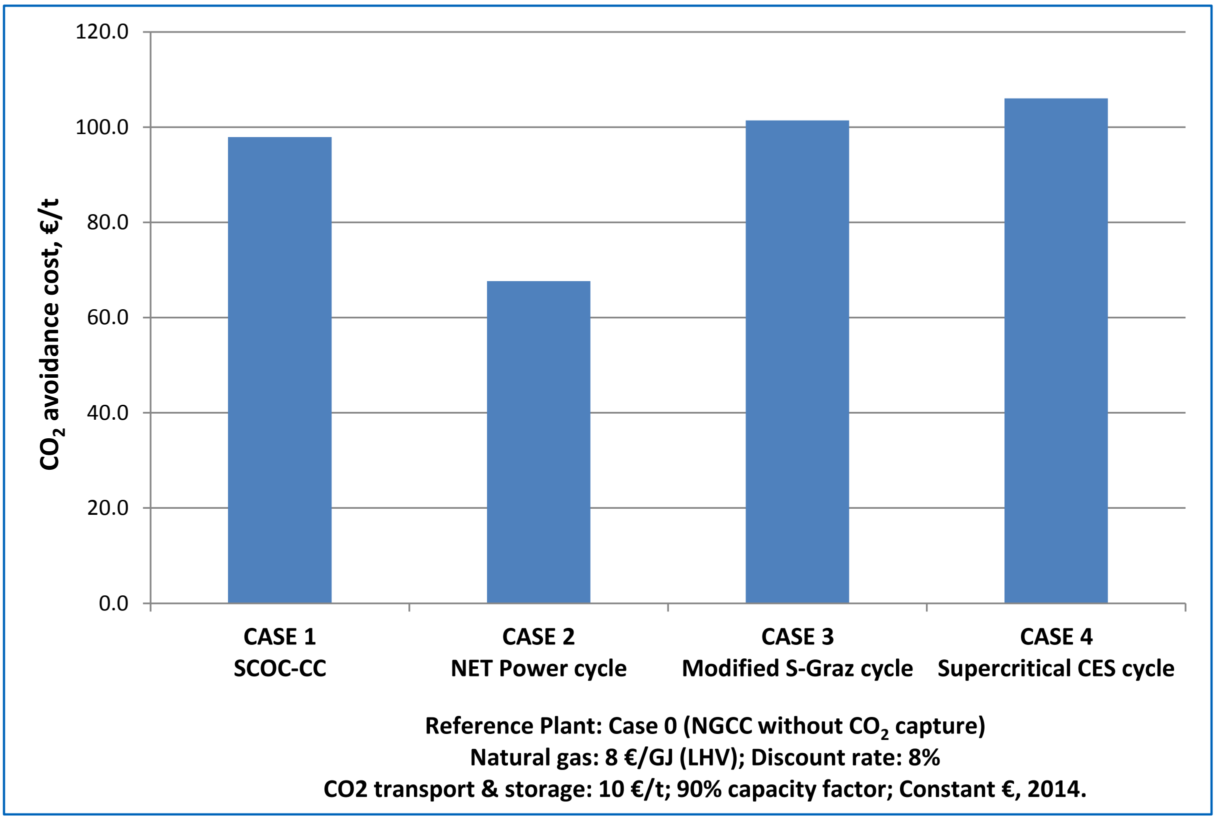


Figure 7. Cost of CO_2 avoidance for base cases

4.5. Sensitivity analysis to the cost of novel equipment

This section summarizes the results of the sensitivity analyses performed to estimate the LCOE and the CO_2 Avoidance Cost of the different study cases, versus the variation of the novel equipment investment costs.

The equipment considered as novel technology are the following:

- Oxy-turbines (including compressor, combustor and expander) for all the cases
- Flue gas cooler (regenerative section) of the NET power cycle.

For each case, the sensitivity range is selected in proportion to the level of component development, as already identified in the oxy-fuel cycle review included in chapter C.1 of the report. A wider range is considered for the cycles characterised by higher development index, as summarised in the following Table 14.

Table 14. Sensitivity range (novel equipment cost)

Case	Description	Novel equipment development index (see below)	Sensitivity range
Case 1	SCOC-CC	Turbine: 2	±20%
Case 2	NET-POWER	Turbine: 3 Combustor: 2 Heat exchanger: 2	±30%
Case 3	Modified S-Graz	Turbine: 2 Compressor: 2	±20%
Case 4	Supercritical CES	Turbine: 3 (*) Combustor: 2	±30%

(*) Turbine development index increased with respect to the figure shown in Chapter C.1, to take into account the increased complexity of the supercritical CES cycle with respect to the configuration currently proposed in the public domain and described in Chapter C.1

1:	Commercial components or technology very close to the state-of-the-art can be used. Modifications to existing components or new design activities are moderate and relatively low-cost.
2:	The component requires dedicated design, but component development seems feasible in short-mid term with the current knowledge. Limited modifications to the process parameters are possible, considering techno-economic limitations on component design.
3:	The component requires dedicated design and operates at conditions far from current commercial components. Significant modifications to the process parameters might be needed on the basis of possible techno-economic limitations on component design.
4:	The component requires a technology breakthrough and the successful techno-economic development is highly uncertain.

LCOE

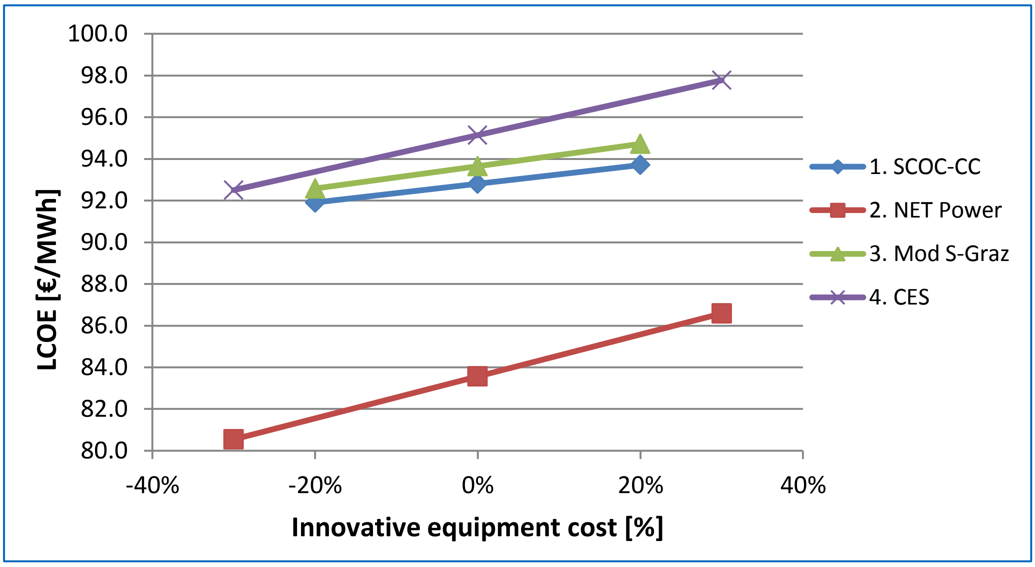


Figure 8. LCOE variation as function of novel equipment investment cost

CO_2 emission avoidance cost

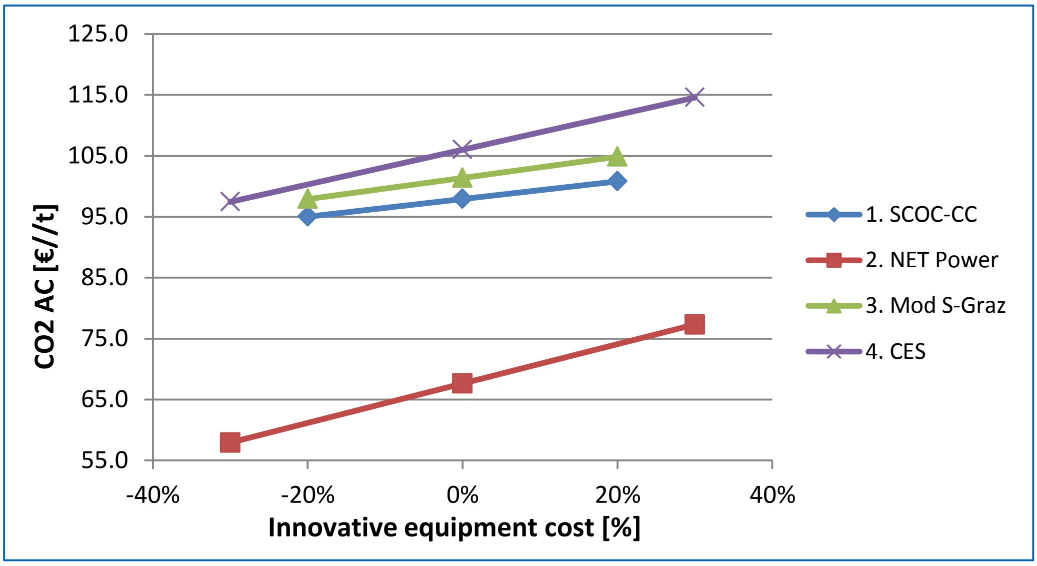


Figure 9. CAC variation as function of novel equipment investment cost

4.6. Sensitivity analysis to technical design parameters

This section summarizes the results of the sensitivity analyses performed to estimate the LCOE and the CO_2 Avoidance Cost of the different study cases, versus the variation of the plant performances (and related investment cost variation if applicable) as analysed in some of the sensitivities to the technical design parameters.

The sensitivity cases considered in this analysis are summarised in the following Table 15.

Table 15. Sensitivity cases (technical design parameter)

Case	Description	Design parameter
Case 1	SCOC-CC	Combustion outlet temperature
Case 2	NET-POWER	Combustion outlet temperature
Case 2	NET-POWER	Cooling water system
Case 2	NET-POWER	CO_2 capture rate & purity
Case 3	Modified S-Graz	Oxygen purity
Case 4	CES	Technology development

4.6.1. *SCOC-CC: Sensitivity to COT*

The impact of varying the combustion outlet temperature on plant performance is technically assessed in the dedicated chapter of the report D.1. Scope of this section is the evaluation of the impact on the total plant cost, LCOE and CAC.

The TPC is mainly affected by the following design changes:

- For the case with lower COT the gas turbine size is almost the same. However a lower investment cost is expected due the lower temperature profile, in particular for the first rotor and stator.
- For the case with higher COT the gas turbine cost is expected to be higher due to both the larger size and the material selection required to resist at higher temperature. Presently it is not possible to have a proper estimate of the cost of future material. For this analysis, a correction factor to the material cost of the equipment manufactured with ‘current’ material is assumed (valid also for following section 4.6.2).
- Steam turbine capacity and cost is lower in both cases.
- For the case with higher COT, a slight increase of the ASU investment cost is expected as delivery pressure is higher.

The following Table 16 summarizes the main technical and economic parameters affected by the COT, as well as main financial results.

Table 16. SCOC-CC: Sensitivity to COT

		Base case COT: 1533°C	COT: 1613°C	COT: 1453°C
Net power output	MWe	756.8	765.4	747.7
Net electrical efficiency	%	49.3%	49.8%	48.7%
TPC	M€	1,111	1,128	1,106
Specific TPC	€/kW	1,470	1,475	1,480
LCOE	€/MWh	92.8	92.2	93.8
CAC	€/t	97.9	95.9	101.3

4.6.2. *NET Power: Sensitivity to COT*

The impact of the COT on plant performance is technically assessed in the dedicated chapter of the report D.2. Scope of this section is the evaluation of the impact on the total plant cost, LCOE and CAC.

The TPC is mainly affected by the following design changes:

- For the case with lower COT and the same metal temperature, a slightly lower investment cost is expected for the gas turbine due the lower operating temperature of the combustor and the lower temperature profile, in particular for the first rotor and stator.
- For the case with higher COT and the same metal temperature, a slightly higher investment cost is expected for the gas turbine due the higher operating temperature of the combustor and the higher temperature profile, in particular for the first rotor and stator.
- For the case with higher COT and higher metal temperature, a more significant cost increase is expected with respect to the previous case, due to the material selection required to resist at higher temperature.

The following Table 17 summarizes the main technical and economic parameters affected by the combustion outlet temperature, as well as main financial results.

Table 17. NET power: Sensitivity to COT

		Base case COT: 1150°C Metal T: 860°C	**COT: 1100°C** Metal T: 860°C	**COT: 1200°C** Metal T: 860°C	**COT: 1200°C Metal T: 950°C**
Net power output	MWe	846	834	847	871
Net electrical efficiency	%	55.1	54.3	55.1	56.7
TPC	M€	1,118	1,104	1,129	1,154
Specific TPC	€/kW	1,320	1,325	1,335	1,325
LCOE	€/MWh	83.6	84.4	83.7	82.0
CAC	€/t	67.6	70.3	68.2	62.5

4.6.3. *NET Power: near-zero emission plants with different CO_2 purity*

The impact on plant performance for the near-zero CO_2 emission cases with different CO_2 puritys is technically assessed in chapter D.2 of the report. Scope of this section is the evaluation of the impact on the total plant cost, the LCOE and the CAC.

The TPC is mainly affected by the following design changes:

- For the low CO_2 purity (around 98%) case, the near-zero emission plant shows an investment cost lower than the base case as the CPU is not necessary. This feature, in combination with the higher power production, leads to a lower LCOE.
- If a higher capture rate were required with the same CO_2 purity of the base case, the investment cost increases due to the increased size of the CPU and the additional equipment required, mainly the membrane and the inert gas compressor. This feature, in combination with a lower power production, leads to a higher LCOE.
- The near-zero emission plants show CAC lower than the base case, due to the increased capture rate.

The following Table 18 summarizes the main technical and economic parameters affected by the various CO_2 capture rates and purities.

Table 18. NET power: near-zero emission plant with different CO_2 purity

		Base case 90% capture 99.8% purity	100% capture ~98% purity	98% capture 99.8% purity
Net power output	MWe	846	850	841
Net electrical efficiency	%	55.1	55.3	54.7
TPC	M€	1,118	1,080	1,134
Specific TPC	€/kW	1,320	1,270	1,340
LCOE	€/MWh	83.6	82.7	84.8
CAC	€/t	67.6	58.0	65.3

4.6.4. *NET Power: alternate cooling water system*

The impact on plant performance in case of mechanical draft cooling tower with extremely aggressive approach of 4°C as opposed to the natural draft system with 7°C approach has been technically assessed in chapter D.2 of the report. Scope of this section is the evaluation of the impact on the total plant cost, the LCOE and the CAC.

The TPC is lower than the based case due to the selection of a less expensive cooling tower type, while the impact on the process unit is negligible. The lower TPC, in combination with the increased efficiency, improves the plant economics, in terms of lower LCOE and CAC.

The following Table 19 summarizes the main technical and economic parameters affected by the different cooling water system designs.

Table 19. NET power: Different cooling water system

		Base case (natural draft CT, 7°C approach)	**Mech. draft CT, 4°C approach)**
Net power output	MWe	846	852
Net electrical efficiency	%	55.1	55.4
TPC	M€	1,118	1,061
Specific TPC	€/kW	1,320	1,245
LCOE	€/MWh	83.6	81.7
CAC	€/t	67.6	61.5

It has to be noted that similar considerations could be deemed valid also for the oxyfuel cycles other than the NET Power one, as the benefit in terms of performance and costs are related to the lower cooling water temperature level and not to a specific integration between the NET power cycle and the cooling water system based on mechanical draft cooling tower.

In fact, all the study cases will benefit of a lower cooling temperature due to the reduction of some auxiliary consumptions. In particular:

- ASU air compressor. The power saving relevant to the MAC is higher for the other oxy-fuel cycle cases as the compressor is cooling water intercooled.
- Raw gas compressor in the CPU. The compression of the net CO_2 product from atmospheric pressure to the cold box pressure is included in most of the other cycles, while it is not included in the NET power configuration.
- Flue gas compressors. As the intercooling of the recycle gas compressor train is entirely made with cooling water, the saving relevant to the exhaust gas compression is more significant for the NET Power cycle, than for the other oxy-fuel cycles considered in this study. However, even if the relevant power saving is lower, also the other cycles benefit of a reduced power consumption for the exhaust gas compressors, related to the lower cooling water temperature.
- As the condensation pressure selected for the steam cycle included in some of the oxy-cycle configuration considered in the study (e.g. SCOC-CC, Modified S-Graz and short-term CES configuration) is already quite low for commercially available steam turbine (i.e. 4 kPa), an increased production for the lower cooling water temperature conditions in the steam condenser has not beeen considered. However, some oxy-turbine cycles do not include a steam condenser but a flue gas condenser whose condensation pressure can be lowered with a lower CW temperature (e.g. S-Graz cycle and the CES cycle, original and supercritical configuration). In particular, the S-Graz cycle optimum condenser pressure decreased with the cooling water temperature level, with a corresponding increased efficiency ([1]).

In addition to the above listed technical considerations, the saving in the investment cost is similar for all the study cases.

[1] W.Sanz, H. Jericha, F. Luckel, E. Göttlich, F. Heitmeir, *A further step towards a Graz cycle power plant for CO_2 capture*, Proceeding of ASME Turbo Expo 2005, June 6-9, 2005, Nevada (US)

4.6.5. *Modified S-GRAZ: Sensitivity to Oxygen purity*

The impact of the selection of oxygen purity on plant performance is technically assessed in the dedicated chapter of the report D.5. Scope of this section is the evaluation of the impact on the total plant cost, LCOE and CAC.

The TPC is mainly affected by the following design changes:

- ASU design changes due to the different oxygen purity required.
- CPU design changes as the inert content and consequently the CPU design capacity and configuration changes with the purity of the supply oxygen.

The following Table 20 summarizes the main technical and economic parameters affected by oxygen purity, as well as main financial results.

Table 20. Modified S-GRAZ: Sensitivity to Oxygen purity

		Base case 97% O_2 purity	**95% O_2 purity**	**99.5% O_2 purity**
Net power output	MWe	755.5	755	750
Net electrical efficiency	%	49.2%	49.1%	48.8%
TPC	M€	1,136	1,135	1,154
Specific TPC	€/kW	1,500	1,500	1,540
LCOE	€/MWh	93.7	93.7	94.8
CAC	€/t	101.4	101.4	105.2

4.6.6. *CES: improvement relevant to the technology development*

Based on the configuration available from literature and on the scheme provide by CES relevant to their last cycle development, three different configurations of the CES cycle have been technically assessed in the dedicated chapters of the report as summarised below.

The ‘original’ and the ‘revised’ configuration for the CES cycle, analysed respectively in the chapter D.5 and D.6 of the report, are based on the short-term solution proposed by the CES cycle as the gas turbine included in the cycle is supposed to be based on the technology currently available for conventional air-fired gas turbine, requiring only small modifications to be suited to oxygen combustion. Both cycles achieve a net electrical efficiency slightly below 43% (LHV basis).

The supercritical CES cycle, analysed in the chapter D.7 of the report represent the long-term solution proposed by CES as further technology improvement is needed due to the high pressure and temperature conditions at the gas turbine inlet.

Scope of this section is the evaluation of the impact on the plant economics, mainly total plant cost, LCOE and CAC of the designing the CES plant with a 'future' gas turbine with respect to a gas turbine based on current technology, i.e. the supercritical CES cycle vs the revised CES cycle.

Between the two short-term solutions, the revised CES cycle has been selected for this comparison, as it is based on a more conventional gas turbine design. In fact, the flue gas is expanded down to atmospheric pressure and not to vacuum condition as in the 'original' configuration (and in the supercritical one).

The following Table 21 summarizes the main technical and economic parameters affected by oxygen purity, as well as main financial results.

Table 21. CES: Different technology development required

		Supercritical CES cycle	**Revised CES cycle**
Net power output	MWe	750.7	658.2
Net electrical efficiency	%	48.9	42.8
TPC	M€	1,160	1,094
Specific TPC	€/kW	1,540	1,660
LCOE	€/MWh	95.1	106.3
CAC	€/t	106.0	145.3

4.7. Sensitivity analysis to economic parameters

This section summarizes the results of the sensitivity analyses performed to estimate the LCOE and the CO_2 Avoidance Cost of the different study cases, versus the variation of the following main economical parameters:

- Natural Gas cost,
- Plant life (project duration),
- Discount rate,
- Costs related to CO_2 emission or transport & storage,
- Operating capacity factor of the plant.

The sensitivity range has been selected in accordance with the study bases, of which the following Table 22 represents a summary.

Table 22. Sensitivity cases (economic parameter)

Economic parameter	Unit	Base Case	Sensitivity Range
NG price	€/GJ	8	4 - 12
Discount rate	%	8	5 - 10
Plant life	years	25	25 - 40
CO_2 transport & storage	€/t stored	10	0 - 20
CO_2 emission costs	€/t emitted	0	0 - 100
Capacity factor	%	90	50 - 90

4.7.1. NG cost sensitivity

LCOE

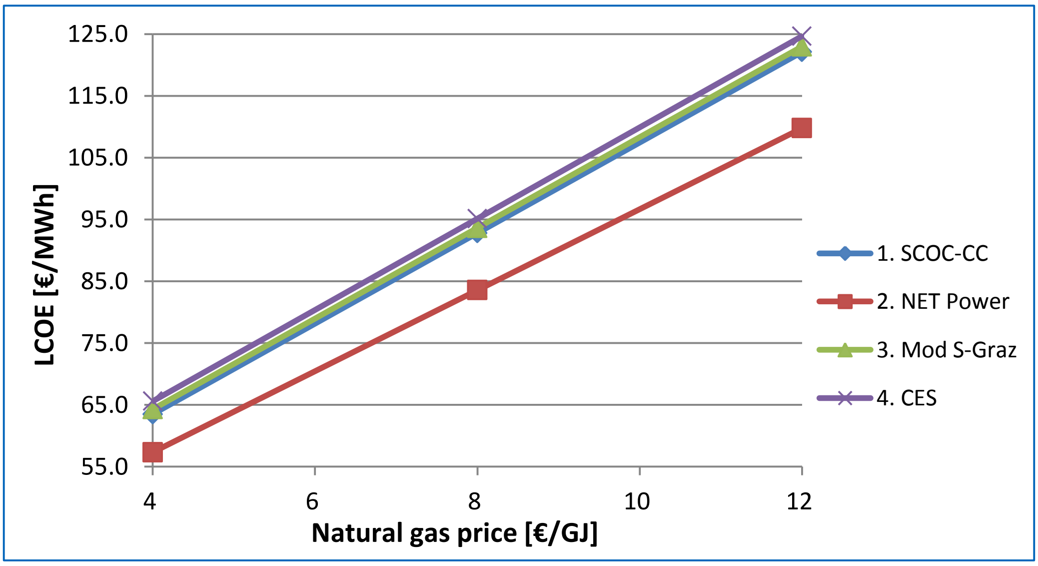


Figure 10. LCOE variation as function of NG price

CO_2 emission avoidance cost

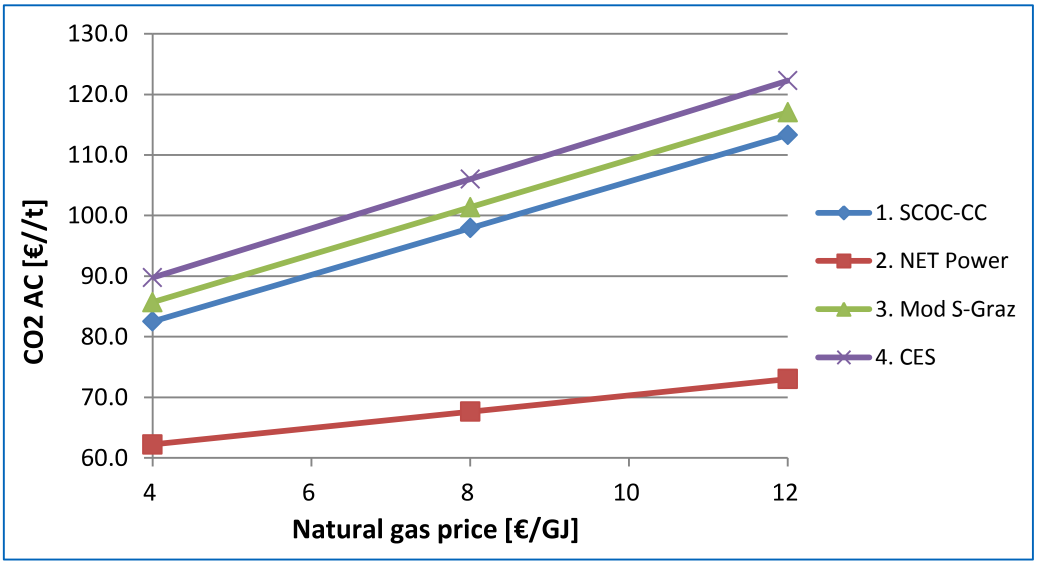


Figure 11. CAC variation as function of natural gas price

4.7.2. Discount rate variation

LCOE

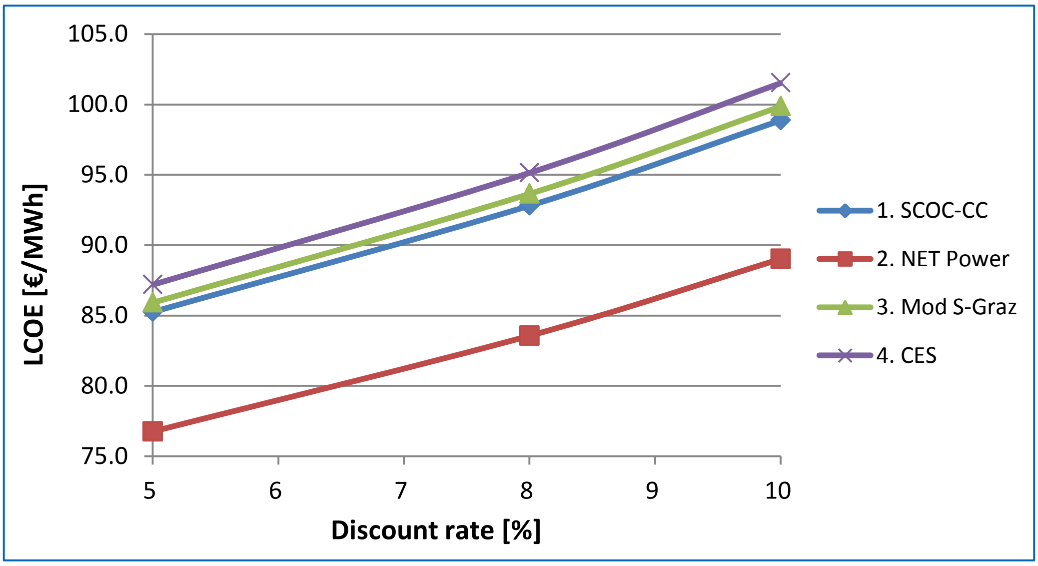


Figure 12: LCOE variation as function of discount rate

CO_2 emission avoidance cost

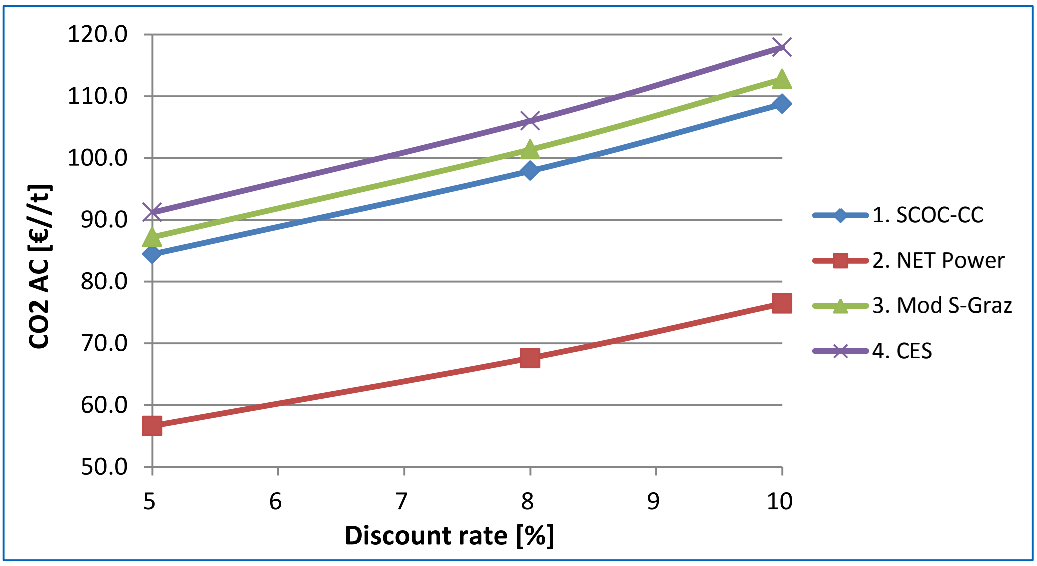


Figure 13. CAC variation as function of discount rate

4.7.3. Plant life

LCOE

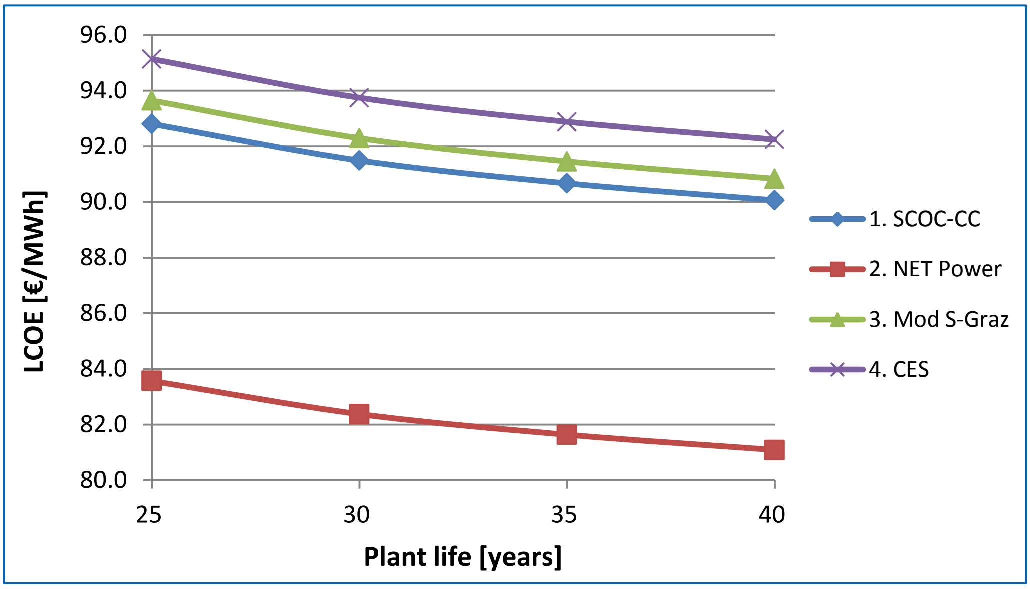


Figure 14. LCOE variation as function of plant life (project duration)

CO_2 emission avoidance cost

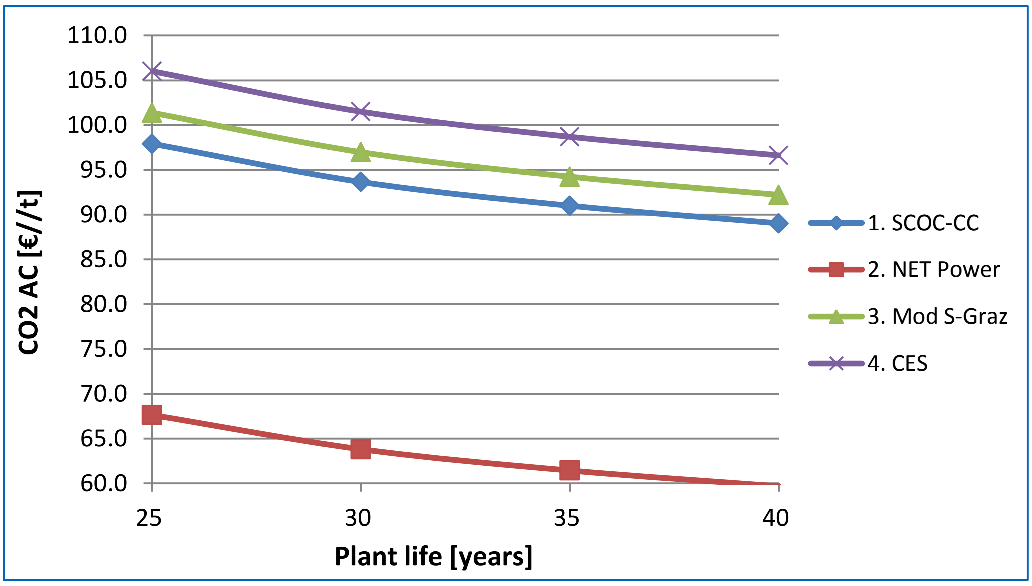


Figure 15: CO_2 emission avoidance cost variation as function of plant life (project duration)

4.7.4. *CO_2 transport & storage cost*

LCOE

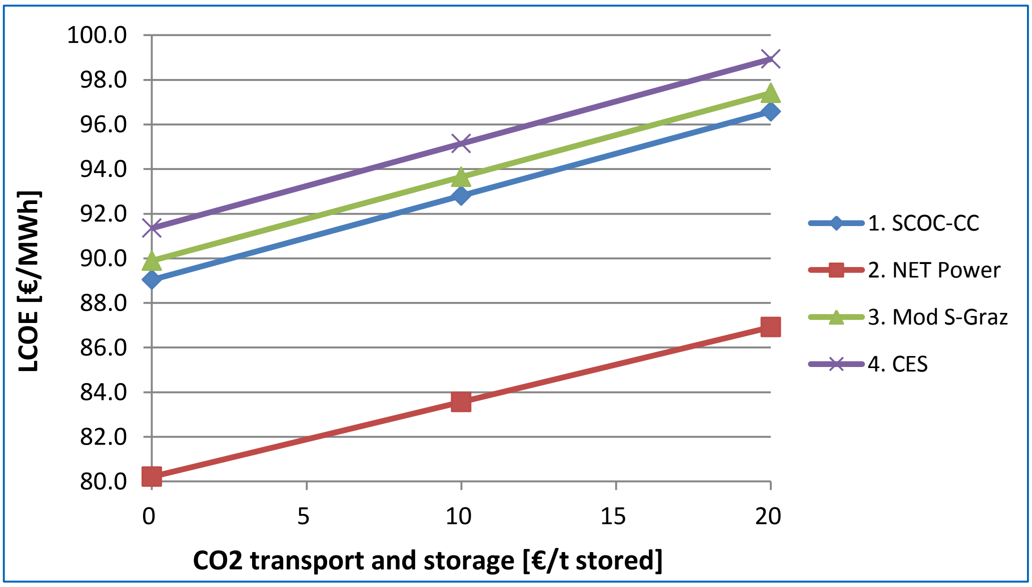


Figure 16. LCOE variation as function of CO_2 transport & storage cost

CO_2 emission avoidance cost

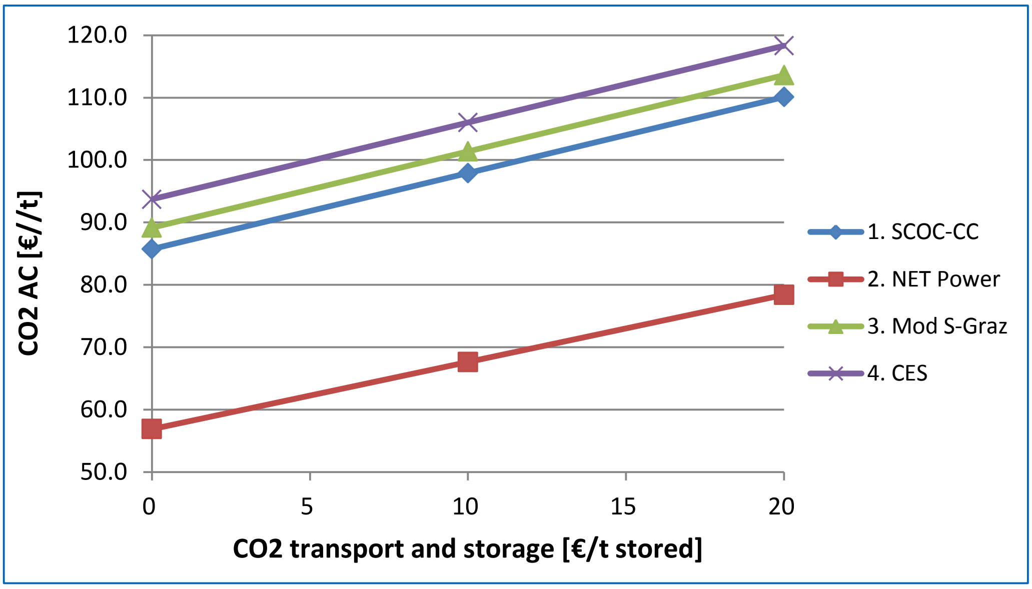


Figure 17. CAC variation as function of CO_2 transport & storage cost

4.7.5. CO_2 emission cost

LCOE

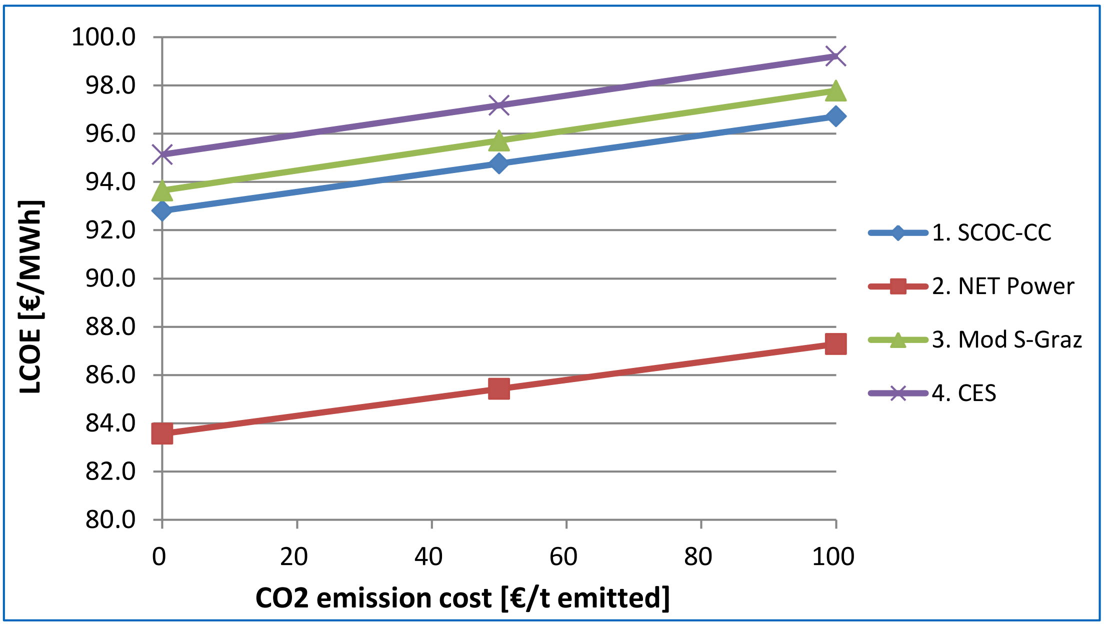


Figure 18. LCOE variation as function of CO_2 emission cost

CO_2 emission avoidance cost

The CO_2 emission avoidance cost in neutral to the variation of CO_2 emission cost.

4.7.6. Load factor

LCOE

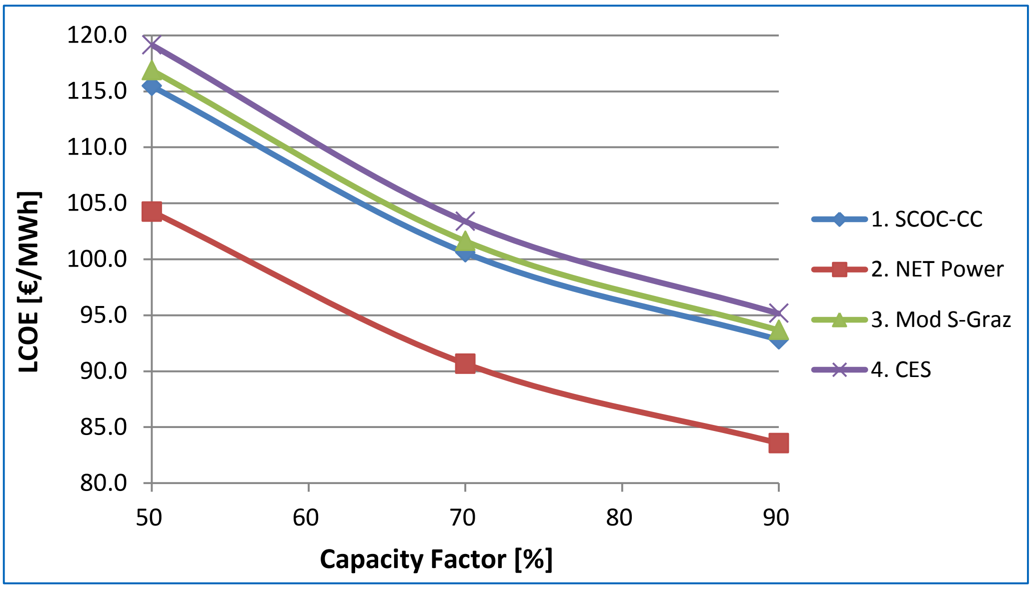


Figure 19. LCOE variation as function of plant load factor

CO_2 emission avoidance cost

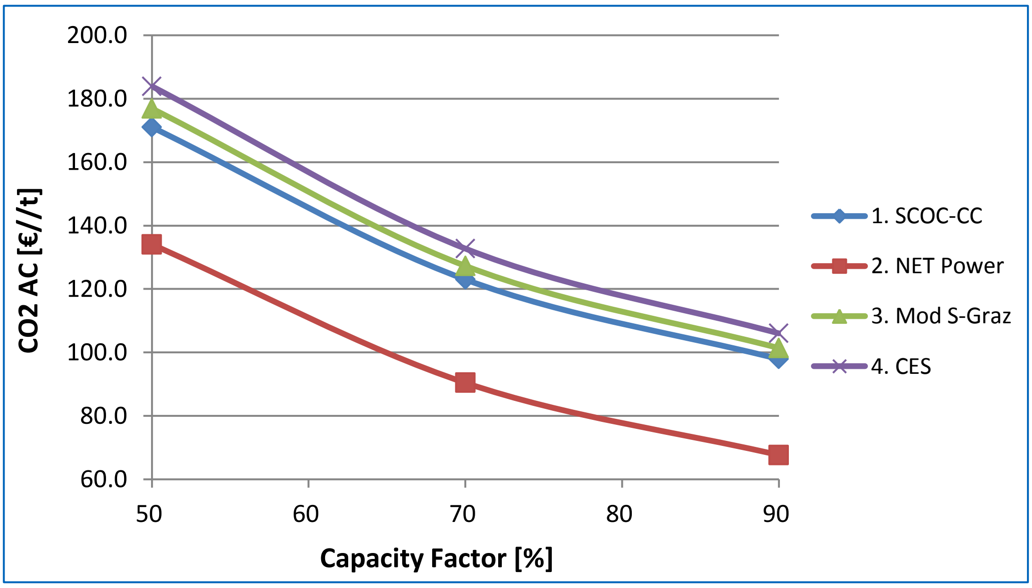


Figure 20. CAC variation as function of plant load factor

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	FUTURE IMPROVEMENT OF OXY-FUEL CYCLES
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	N. FERRARI
CHECKED BY	:	L. MANCUSO
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

The scope of this chapter is to investigate the potential for future development or improvement of the oxy-fuel gas turbine cycle technologies.

The investigation is mainly focused on the possible advantages for the oxy-turbine cycles related to the development and improvements of high temperature materials not only for the turbine, but also for the downstream exchangers. Potential efficiency improvements related to the introduction of the membrane technology for oxygen separation are also assessed.

It has to be noted that the technology of the oxy-fuel cycle is actually a novel technology itself. A few cycles only have undertaken already some tests or are currently developing key equipment, like the turbine in cooperation with gas turbine supplier, while most of the cycles are only at study level, i.e. at thermodynamic definition only.

It is expected that, even if the use of high temperature materials could add significant benefits to the oxy-turbine power plants, the following shall be considered of priority importance for the development at commercial level and at utility scale of the oxy-turbine technology:

- *Gas turbine design for CO_2 rich stream*
  The CO_2 turbine shall be re-designed to adjust the blade geometry and the cooling channels to the properties of the CO_2-rich working fluid. As already mentioned in the previous sections of the report, NET Power only is developing the gas turbine in cooperation with Toshiba to deeply assess this topic.
- *High pressure not cooled gas turbine design*
  Some cycles as the Matiant and the CES are based on high pressure gas turbine (inlet pressure in the range of 120-300 bar) with no blade cooling (selecting a suitable material). Even if it is reasonable that gas turbine design will take advantage of the well- known steam turbine design, the working fluid is different and proper modification of the expander geometry should be carefully investigated and developed.
- *Oxy-combustor design*
- *Regenerative heat exchanger design*
  Multi-current, plate fin exchangers, similar to the technology developed for cryogenic applications, are proposed for the regenerative section of the oxy-fuel cycle. However, suitability and design of these heat exchangers at the pressure and temperature level of the oxy-fuel cycle is still to be proven.

- *Power plant scale-up.*

  For the technical assessment included in the report, a F-class equivalent gas turbine is considered commercially available, in order to design a full-scale power plant in line, in terms of power production and specific capital cost, with the conventional air-fired gas turbine.

2. High temperature materials development

2.1 High temperature materials for conventional gas turbines

Advancements in gas turbine materials have always played a primary role for the performance of power plants, as the equipment heat recovery or the power generation increase with the capability of the materials to withstand elevated temperature services.

A wide spectrum of high performance materials - special steels, titanium alloys and superalloys - is used for construction of gas turbines and in particular for those components that are critical towards the equipment performance, as shown in the following:

- **Compressor blades**
  Corrosion of compressor blades in conventional air-fired gas turbine can occur due to air moisture condensation on the blades, containing salts and acid. To prevent the corrosion and also improve the erosion resistance, several martensitic stainless steels with aluminium slurry coatings are used for the compressor blades.

- **Combustor chamber**
  The primary basis for the combustor material is increasing the temperature creep rupture strength without sacrificing the oxidation / corrosion resistance. Cobalt base or nickel based super-alloys have been recently adopted for some combustion system components. In addition to designing with improved materials, combustion systems are further protected with thermal barrier coating, which provides an insulating layer reducing the base metal temperature.

- **Turbine disk**
  The use of nickel-based or nickel-iron based super-alloys provides to the rotors the necessary temperature capability required to also meet the firing temperature requirements in the future.

- **Turbine blades and vanes**
  Turbine blades must withstand severe combination of temperature, stress and environment. In particular the 1^{st} stage bucket is particularly critical, and is generally the limiting component of the gas turbine, as it combines several difficulties (extreme temperatures in the range of 1400-1500°C, high pressure, high rotational speed, vibration, small circular area, etc.). In order to overcome those barriers, gas turbine blades are made using advanced cast nickel-base super-alloys.

The nozzle, even if exposed to high temperature, suffers a lower mechanical stress. Cobalt-base super-alloys are usually selected for the 1st stage nozzles while iron-base or nickel-base super-alloys are applicable to later stages' nozzles.

Directionally Solidified (DS) and single crystal (SC) casting improve the turbine blade resistance to hot corrosion and oxidation and the long term stability and creep strength with respect to the conventionally equi-axed cast super-alloys, as graphically shown in Figure 1.

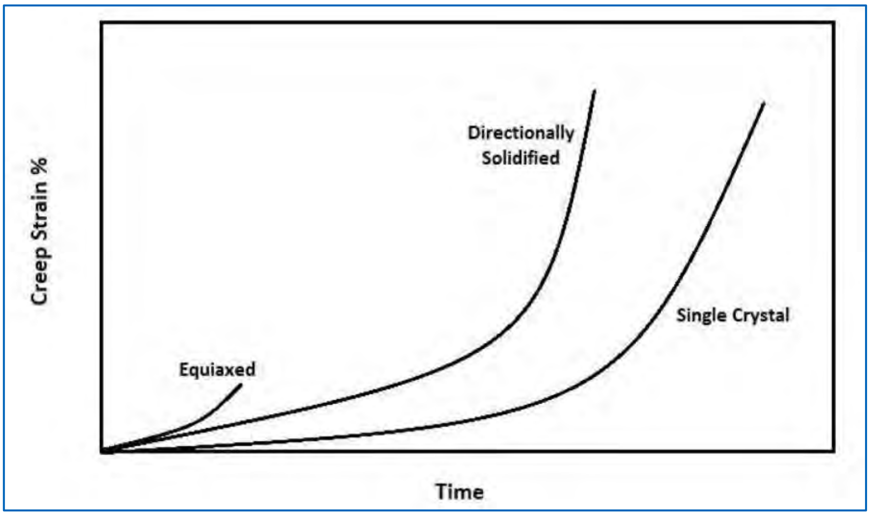


Figure 1. Relative creep deformation of equiaxed, DS and SC super-alloy casting

2.1.1 *Material coating*

In order to design super-alloys which have the necessary creep strength on one side and the required resistance to corrosion / oxidation on the other side, it has become inevitable to provide coatings (diffusion coating, overlay coating or thermal barrier coatings) on to the surface of the blades to enhance corrosion resistance and to increase the temperature difference between the gas stream and the blade metal.

Improvement of thermal barrier coating (TBC) has been a focal theme for the R&D efforts in the recent years, and could be extremely useful also for the oxy-fuel turbine application.

2.1.2 *Advanced materials in R&D*

Future increase in turbine inlet temperatures, beyond what is possible with super-alloys, can be achieved using ceramic materials in place of superalloys.

Ceramic materials are capable to withstand high temperatures, are up to 40% lighter than comparable high temperature alloys and cost around 5% the cost of superalloys.

Ceramic materials based on silicon carbide and silicon nitride could be potential candidates for gas turbine applications, including oxy-fuel gas turbine.

The introduction of Ceramic matrix composites (CMC) as hot gas path components such as the combustion chamber could be a possible way for increasing the combustor operating temperature without increasing the penalty associated with increased cooling air.

Chromium-base and platinum-base alloys have the potential to be used at temperatures up to 1,700°C. Despite their high cost, they are attractive for some gas turbine applications due to their exceptional resistance to oxidation, high melting points, ductility, thermal shock resistance and thermal conductivity.

2.1.3 Cooling techniques

Improvement of cooling techniques has been proving of fundamental importance in increasing the peak temperature of gas turbine cycles. The gain in firing temperature attained during evolution of gas turbines is actually due to the refinement of the cooling techniques rather than to an improvement of materials.

Effectiveness of cooling circuits has been enhanced in various ways including closed-loop operations. This particular technology was investigated by different manufacturers provided that open loop air-cooled gas turbines have a significant temperature drop across the first stage nozzles, which reduces TIT for a given combustion outlet temperature. The closed-loop cooling system allows the turbine to increase the TIT for better performance, yet without increased combustion temperatures.

Closed-loop steam cooling was introduced by General Electric at the end of the 90s and some full scale prototypes have been built and tested. The first unit of this class has been in commercial operation since September 2003 at Baglan Bay (UK) power station and has achieved significant operating experience. Despite of the advantages offered in terms of efficiency and specific power output, this solution was abandoned because it is essentially suitable for base load generation but lacks the operational flexibility required of to current combined cycle units.

Closed-loop steam cooling could be particularly suitable to oxy-turbine cycle. Besides the mentioned benefits, it allows mixing the oxygen for combustion directly at the compressor inlet. This would reduce the energy requirement for oxygen compression and increase the temperature of the O_2 to the combustor leading to a better efficiency. This solution is precluded in open-loop cooling because a huge amount of oxygen bypasses the combustor and remains in the CO_2 stream therefore significantly reducing the CO_2 purity and increasing the energy consumption of the ASU.

2.2 Impact on oxy-turbine cycle performance

The high temperature materials and coating treatments developed for the conventional air-fired gas turbine components are in principle applicable also to the oxy-combustion machines, with the main purpose of increasing the maximum allowed turbine metal temperature, reducing the cooling stream and consequently increasing plant efficiency.

To show the potential improvement of cycle efficiency due to the increased turbine blade metal temperature, two different cases have been assessed with different design metal temperatures: one is the SCOC-CC cycle, representative of a combined cycle, the other is the NET power cycle, representative of a recuperative cycle.

A similar effect is expected also for the other oxy-fuel cycles.

2.2.1 High temperature material: higher COT and metal temperature

As an effect of using materials with an improved resistance at higher temperature, the following design parameters have been modified with respect to the design considered in the technical assessment of the study base cases:

- The combustion outlet temperature is increased as detailed in the below table, thus lowering the required recycle flowrate to the combustion chamber and consequently the recycle compressor consumptions:

Oxy-fuel cycle	Combustion outlet temperature	
	Base case	New materials
SOCC-CC Inlet P: 45 bar	1533°C	1613°C
NET Power Inlet P: 300 bar	1150°C	1200°C

- The maximum allowable blade metal temperature is increased as detailed in the below table, thus reducing the required cooling flow to the gas turbine blades, and consequently lowering the total recycle flowrate and related compressor consumptions, while also improving gas turbine efficiency:

Turbine section	Maximum blade temperature	
	Base case	New materials
1st stator	860°C	950°C
1st rotor	830°C	890°C
from 2nd stator	830°C	890°C
from 2nd rotor	800°C	860°C

The combined effect of the increased COT and maximum allowed metal temperature is a higher temperature of the gas turbine exhaust gas entering the downstream HRSG (for the SCOC-CC) or the regenerative sections (for the NET Power), if the gas turbine compressor ratio is kept constant.

For the combined cycle, two different alternatives have been assessed, the first applying the new material only to the gas turbine section, keeping the same limitation of the base case in the bottoming steam cycle (maximum steam temperature around 600°C), the second considering also a higher temperature limit for the steam temperature (around 640°C) as improved material are applied also to the steam turbine.

In the first alternative, due to the limitation on the maximum steam temperature, a higher flue gas exhaust temperature does not lead to real benefits to the cycle efficiency. Therefore, the gas turbine compressor ratio has been increased to maintain the exhaust gas temperature in the optimum range for steam generation (620-630°C). In the second case, the optimum exhaust gas temperature being increased, the gas turbine compressor ratio is the same as in the base case.

On the other hand, for a regenerative cycle like the NET Power cycle, the increased exhaust temperature leads to higher temperature design requirements for the hot side of the heat exchanging section, also affecting material selection for the equipment downstream the gas turbine.

2.2.2 <u>*SCOC-CC*</u>

The impacts on key design parameters and plant performances with high temperature materials for the gas turbine blades, and consequently higher COT and metal temperature as described in above, are shown in the following Table 1.

Table 1. SCOC-CC: high temperature material applied

	Base case	New materials (only GT)	New materials (GT+ST)
Key design parameters			
Compressor ratio	44.5	55.0	44.5
Turbine inlet temperature	1352°C	1412°C	1449°C
Max steam temperature	600°C	600°C	640°C
Exhaust gas temperature	620°C	625°C	666°C
Performance figure			
Gas turbine efficiency (LHV basis)	39.8%	41.4%	40.3%
Gross electrical efficiency (LHV basis)	63.0%	63.7%	64.1%
Net electrical efficiency (LHV basis)	49.3%	49.8%	50.3%

In the case with the steam cycle based on current material, the potential gas turbine efficiency improvement when applying new generation materials is around 1.6 percentage points. Due to the higher metal temperature allowed, a lower recycle gas is required and consequently also the total exhaust gas flowrate from the gas turbine is lower, leading to a lower steam generation in the downstream HRSG. Therefore, power generation from the steam turbine decreases, lowering the overall plant net efficiency gain in the range of 0.5 percentage points.

In the case with the new generation material applied to the steam cycle, the gas turbine efficiency improvement is lower (0.5 percentage point as the compressor ration is not increased). On the other hand, the efficiency of the steam cycle increases with the higher steam temperature allowed, leading to an overall efficiency improvement of around 1 percentage point with respect of the base case (0.5 percentage points higher than the first case).

2.2.3 *NET power*

The impacts on key design parameters and plant performances when considering high temperature materials for the gas turbine blades, and consequently higher COT and metal temperature as described in above, are shown in the following Table 2.

Table 2. NET Power: high temperature material applied

	Base case	New materials
Key design parameters		
Exhaust gas temperature	740°C	793
Cooling stream (percentage of flue gas)	11.5%	7.5%
Performance figure		
Gas turbine efficiency (LHV basis)	68.7%	70.4%
Gross electrical efficiency (LHV basis)	68.7%	70.4%
Net electrical efficiency (LHV basis)	55.1%	56.7%

The potential gas turbine efficiency improvement when applying new generation materials is around 1.7 percentage points (similarly to the efficiency gain evaluated for the gas turbine included in the combined cycle).

In a regenerative cycle, as the gas turbine is the generators, the gas turbine efficiency corresponds to the plant gross electrical efficiency. As the impact on the auxiliary consumption is negligible, the same improvement is expected for the overall plant net electrical efficiency.

Based on the above results, it is evident that there is the potential for future improvement of plant performance as a consequence of the application of new generation high temperature material for the oxy-turbine power plants, although the benefit for the regenerative cycles is greater as there are not additional constraints for the downstream equipment.

3. Membrane technology for oxygen separation

Nowadays the only technology available at large commercial scale for the oxygen production for utility scale power plants is the cryogenic air separation process.

Cryogenic air separation, although it is a very reliable and mature technology, accounts for a significant part of the plant power demand and total plant cost. Moreover, it is not expected that cryogenic technology will incur significant cost reduction or efficiency improvements in the coming years. Therefore, new oxygen separation processes are under development for use in large scale power and industrial applications.

On this respect, Oxygen Transport Membrane (OTM), also known as ITM standing for Ion Transport Membrane, is one of the most promising technologies for oxygen separation currently under development.

These materials separate oxygen from air at high temperature in an electrochemically driven process. The oxygen in air is ionized on the surface of the support and diffuses through the membrane as oxygen ions, driven by an oxygen activity (partial pressure) gradient to form oxygen molecules on the other side. Impurities such as nitrogen are rejected by the membrane, allowing a selective separation.

The air separation system that results from the use of such oxygen ion transport membranes produces a hot, potentially 100% pure oxygen stream and a hot, pressurized, oxygen-depleted stream from which significant amounts of energy can be still extracted.

Because of the requirement to feed a high-pressure oxygen-containing gas to the membrane separator and the production of the hot high-pressure off-gas stream, ITM Oxygen systems advantageously integrate with turbo-machinery-based power cycles. Compressed air, obtained directly from an air compressor or extracted from the compression side of a gas turbine, is heated to 800-900°C, and supplied to the membrane. Heat for air preheating can be obtained by burning part of the fuel (as in the AZEP cycle) or recovering heat from the exhaust flue gas from other turbo-machinery within the oxy-fuel cycle (as in the ZEITMOP cycle).

A portion of the oxygen is extracted across the membrane and the resulting oxygen-depleted non-permeate off-gas (which can contain sufficient oxygen to support post-combustion, if desired) can be expanded in an expander or combustion turbine to recover useful work.

This separation process looks particularly attractive for the expected low costs and low energy consumption compared to cryogenic plants.

3.1 Technology development

Numerous process and economic studies have been conducted on ITM technology for oxygen separation. ITM Oxygen has consistently shown excellent economic performance across multiple applications where oxygen and power are needed. ITM Oxygen technology integrates well with power generation cycles, and with well-known approaches for CO_2 capture, including pre-combustion capture and oxy-fuel combustion capture.

Detailed engineering studies have been conducted by the U.S. Department of Energy (U.S. DOE), the Electric Power Research Institute (EPRI) and WorleyParsons Group, Inc, in cooperation with technology suppliers as Air Products.

In particular, Air Products has designed and constructed a Sub-scale Engineering Prototype (SEP) in 2006. The ITM Oxygen development program is currently focused on commissioning and start-up of an Intermediate-Scale Test Unit (ISTU) for operation and testing in 2014, with a design capacity of 100 TPD[1].

Within the European FP7 projects, the DEMOYS project, in which Foster Wheeler was involved, was focused on the application of ITM for oxygen production in IGCC or for coal oxy-combustion process. The technical assessment and economic analysis of these plants indicate that ITM may lead to significant cost reduction compared to conventional cryogenic processes, as shown in Figure 2.

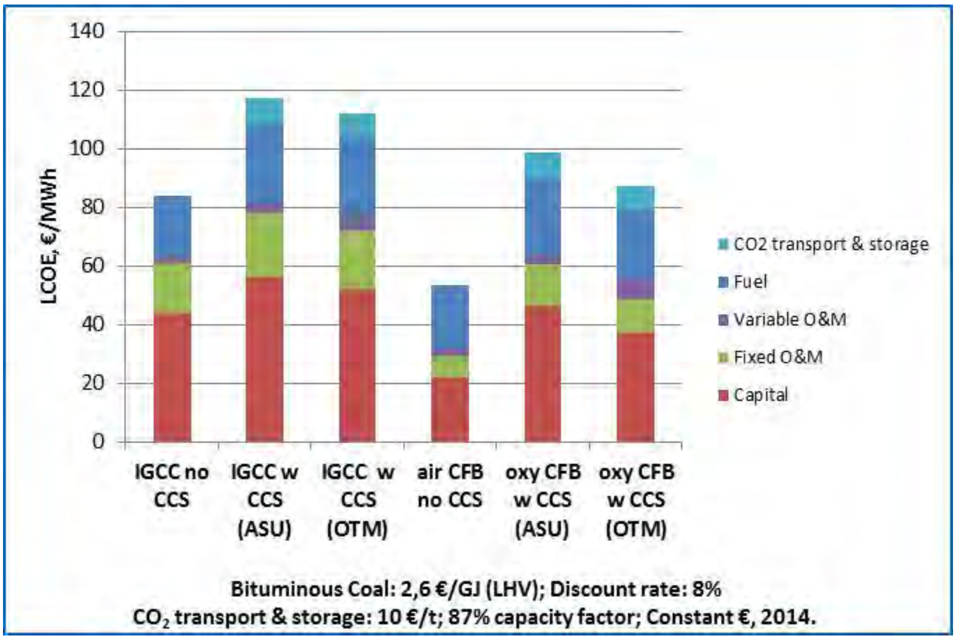


Figure 2. LCOE: OTM vs. cryogenic ASU in IGCC and oxy-combustion CFB plant

[1] V. White at al., *ITM Oxygen Technology for Gasification Applications*, Presentation at IChemE: New Horizons in Gasification Rotterdam, The Netherlands, 10-13 March 2014

The DEMOYS project has demonstrated that with respect to the benchmark technologies the use of membranes improves the overall plant performance, while reducing also the total capital requirement for the investment. In fact, the capital cost of an ITM-based unit is from about 20% to 30% lower than the equivalent cost of standard cryogenic Air Separation Unit, though the operating and maintenance costs of the ITM-based plants are about 10% higher than the benchmark plants, mainly due to the variable cost required for the replacement of the membranes at the end of their operating life.

The economical attractiveness of the membrane integrated plants strongly depends on key factors like permeability, specific cost and lifetime of the membranes. However, various sensitivity analyses have clearly demonstrated that LCOE and the CAC of the OTM-based plants are lower than those of plants using benchmark CO_2 capture technologies.

The potential promising results achieved applying ITM within IGCC and coal oxy-combustion plants suggest that similar benefits could also be obtained introducing membrane technology for oxygen separation in oxy-turbine based power plants.

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : OPERATING FLEXIBILITY OF OXY-TURBINE PLANTS
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

Nowadays, existing and new power plants must face the challenges of the liberalized electricity market and the requirement to cover intermediate and peak load constraints, so to respond to the daily and seasonal variation of the electricity demand. Due to the increasing use of renewable sources, the capability to operate flexibly is expected to be of primary importance also in the future, especially in the hopeful event that carbon capture technologies, including oxy-fuel turbines, will find large commercial applications.

Therefore, this study has preliminarily investigated the potential for flexible operation of oxy-turbine power plants. Unfortunately, due to the early status of technology development, specific information is not yet available in the public domain, so the main purpose of this chapter is to evaluate possible constraints to the capability of these plant-types to operate flexibly, including plant start-up and shut-down, low load operation and ramp rates.

The following key aspects have been considered:

- Flexibility of oxy-turbine cycles with respect to standard air fired natural gas combined cycles.
- Additional constraints related to those process units that are not part of a standard combined cycle design, like the ASU and the CPU, including possible design improvements to overcome them, such as the oxygen storage.

The assessment shown in the next sections is mainly based on the results of the IEAGHG report 2012/6 '*Operating Flexibility of Power Plants with CCS*' developed by Foster Wheeler in 2011-12.

The flexibility features of the oxy-turbine cycles, such as cycling capability, efficient turndown, fast start-up and ramp rates are discussed in section 2, while section 3 focuses on some design modifications or operating modes that may enhance the plant operating flexibility and its capability to follow the variable electricity demand, such as peak load operation and tuning of the auxiliary consumptions.

2. Flexibility features of oxy-turbine cycles

The operational flexibility of the conventional combined cycle plants is generally characterized by:

- Good efficiency at partial load operation.
- High cycling capability (e.g. fast start-up and shut down, fast load change and load ramps, low start-up emissions, high start-up reliability).
- Low operating costs (high start-up efficiency or short start-up time).
- Low turndown, corresponding, for a conventional combined cycle, to a low minimum technical environmental load of the gas turbine.
- Good capability of grid frequency control.

The purpose of the following sections is to provide an overview on how these features potentially apply to the oxy-fuel gas turbine cycles.

2.1 Partial load operation

For the oxy-fuel machines, it is reasonable to expect that their partial load behaviour is similar to that of conventional gas turbines.

For an F-class gas turbine the efficiency penalty corresponding to a load reduction down to 60% is around six (6) percentage points, while the overall combined cycle efficiency is around a few percentage points (2-3) lower than the base load operation.

Whilst similar figures are expected for the gas turbine, a higher penalty on the overall plant net electrical efficiency is however predictable for the oxy-turbine cycles due to the inefficient operation of those process units that are not part of a standard combined cycle design, like the ASU and the CPU (refer to section 0 for additional details).

2.2 Start-up and cycling capability

2.2.1 Start-up sequence

Currently, considering the development stage of the oxy-fuel cycle technology, in particular at commercial scale, two different alternatives are envisaged for the oxy-turbine plant start-up.

- Air-firing mode start-up
- Oxygen-firing mode start-up (using CO_2 to establish gas recirculation)

Start-up in air firing mode

This alternative is based on an approach similar to the oxy-combustion boiler based plant, and it is deemed feasible (w/o significant additional cost) in particular for the 'combined cycle' based configuration as the SCOC-CC and the S-Graz cycle.

In fact, the oxy-fired gas turbine shall be designed such as both the compressor and the expander are able to operate in air mode. A start-up stack shall be also included in the design, in order to vent the flue gas until its composition is adequate to feed the downstream CO_2 purification unit.

Depending on the oxy-fuel cycle, additional equipment shall be included in the plant design, like the start-up air compressor to compress ambient air up to the normal suction pressure of the recycle gas (30-50 bar).

The maximum load level that will be achieved during air firing will depend on the load accepted by the compressors. However, in order to minimise uncontrolled emissions from the plant during the switch over to oxy-fuel, it would be advisable to operate the plant at low load.

While the oxy-turbine cycle is started-up in air firing combustion mode, the ASU is cooled down contemporaneously. When the gas turbine is in operation at minimum stable load and oxygen from the air separation unit is generated at the required purity, the combustion mode can be changed from air to oxygen and simultaneously the flue gas recirculation is started for gas turbine cooling purpose.

At the same time, the CO_2 purification unit can be started-up. When plant load is increased to an acceptable value for the compressors and the composition is adequate for downstream treatment, flue gas can be finally fed to the CO_2 purification and compression section.

Oxygen-firing mode start-up

This alternative is based on the use of a CO_2 stream from the network outside plant battery limits, to establish the exhaust gas recirculation in the gas turbine and in the recycle compressor loop. Oxygen from combustion is supplied from the ASU, if the cycle start-up is performed after the completion of the ASU start-up, or from the oxygen storage, if properly designed to provide oxygen while cooling down the ASU.

The main advantages of this approach are that neither any additional equipment is required for the start-up nor air-mode operation has to be considered for the compressor and turbine design. On the other hand, a large amount of CO_2 is required to provide the proper gas inventory for establishing gas re-circulation in a commercial scale power plant, which is not deemed a particular criticality if, in the future, a CO_2 network is available near the power plant.

Any additional limitation relevant to the ASU start-up time, can be easily overcome with a properly designed LOX storage.

2.2.2 *Start-up time*

Assuming that the oxygen supply to the plant is not made with an over the fence approach, the main constraint for a fast start-up is the time for cooling of the ASU and the time to meet the required oxygen purity, in particular if start-up in air firing mode is not envisaged.

The following Table 1 summarises the typical start-up times of an ASU, including the time necessary to achieve the required oxygen purity.

Table 1. Start-up times for ASU in oxy-fuel plant

Initial condition	Start-up time
After defrost	36 hours
After 24 hours shutdown	6 – 8 hours
After 16 hours shutdown	4 – 6 hours
After 8 hours shutdown	3 – 5 hours
Less than 1 hour shutdown	Less than 1 hour

The ASU start-up can be decouple from the oxy-cycle start-up, providing a dedicated and properly sized liquid oxygen storage, with relevant pumps and vaporiser, supplying oxygen to the gas turbine combustor, during ASU start-up. Designing the oxygen storage for the start-up can be an advantage mainly if the cycle is not designed for air-ring mode, allowing combustion in the gas turbine, while the ASU is not yet in operation.

If start-up is performed in air firing mode, oxygen storage allows establishing gas recirculation, with a nitrogen-rich stream, even if oxygen from the ASU is not available at the required purity.

In addition, if the gas turbine operates also in air-firing mode, there is in principle the possibility to switch to oxygen mode during the ASU start-up transient, even if oxygen purity is lower than the normal operating mode. This could be done by properly adjusting the recycle ratio in order to provide the proper temperature control and the adequate excess oxygen to the combustor. The detrimental effect is the reduction of the CO_2 capture rate, due to the higher inert content in the gases feeding the CO_2 purification system.

Though the start-up time of an oxy-turbine power plant is in shadow of the ASU start-up time, some of the technical advances developed in the last years to improve the flexibility of the combined cycles could be considered also in the design of oxy-turbine cycles. This is particularly valid for the SCOC-CC, which has some of the characteristics that typically restrict the operational flexibility of a combined cycle, like:

- Gas Turbine and Steam Turbine ramp restrictions;
- Heat Recovery Steam Generator ramp restrictions;
- Vacuum system and steam chemistry.

The following key technical features, developed in the last years to optimise the combined cycle start-up process, could be generally considered also for the oxy turbine plants:

- The use of final-stage, high-capacity attemperators in the high pressure and medium pressure reheat steam lines, so to properly control the steam temperature and decouple the gas turbine ramp rate from that of the steam turbine. Because of this decoupling, it is possible to start-up quickly the gas turbine, while the steam turbine is put in operation with its dedicated and slower ramp rate. Moreover, a greater cycling capability is possible for the entire power plant, as this can follow the load variations with the gas turbine first and then with the steam turbine.
- The use of once through steam generators (e.g. Benson design), typically for small-scale power plants, has further reduced the restrictions on the temperature and pressure transients, thus improving the operational flexibility both during start-up and load changes. The Benson design, in fact, eliminates the high pressure thick wall drum and allows an unrestricted gas turbine start-up, including a high number of fast start-up and load changes.
- Faster start-up can be achieved by minimising heat losses in the HRSG and in the steam generators, and maintaining vacuum conditions in the condenser during night shutdown.

2.2.3 *Ramp rates*

As for the start-up time, the main limitation of the oxy-turbine power plants is given by the ramp rate of the ASU.

The maximum ramp rate for an ASU is typically 3% per min, while the gas turbine would typically achieve around 15% per min, while the ramp rate of the whole combined cycle is only slightly lower. Therefore, a plant ramp rate in line with the gas turbine capacity could be achieved by using dedicated and properly designed oxygen storage, which provides the difference between gas turbine oxygen requirements and ASU capabilities, decoupling the oxy-cycle and the ASU ramp rates.

2.3 Minimum turndown

The following technical constraints shall be considered for the evaluation of the minimum plant turndown:

- The minimum load of the gas turbine and, if relevant, of the downstream HRSG and steam cycle is its minimum environmental load (further details in section 2.3.1).
- Minimum turndown of the ASU cold box is typically around 40-50% of the design load, maintaining a constant oxygen recovery and purity.
- Minimum turn down of the auto-refrigerated inerts removal section is typically around 30%.

As for that, it is envisaged that the minimum turndown of an oxy-fuel power plant is typically not lower than 50%.

The following aspect has to be taken into account in order to define the minimum efficient turndown. In fact, the ASU compressors as well as the CPU compressors operate efficiently in the typical range 70-100% of their maximum flow, at constant discharge pressure. This characteristic allows to operate efficiently in the range 70% to 100% with a single train configuration. Considering multiple train configuration, efficient operation would be possible even at lower loads.

If plant operation below 70% were required, then there would be an impact on the machine's efficiency, and it could be necessary to:

- Recycle a portion of the compressed stream (air or CO_2);
- Vent a portion of the produced oxygen in the ASU;
- Produce a certain quantity of liquid oxygen for backup storage, if foreseen.

2.3.1 *Minimum technical environmental load*

The minimum technical environmental load is defined as the minimum condition at which the gas turbine is able to operate still meeting the environmental limits, in particular the NO_X and CO emissions. In a conventional air fired gas turbine, it varies generally from 30% to 50% of the base load power production.

Independently from the cycle type, NOx emission is not generally an issue in oxy-turbine plants. The eventually generated NOx is mostly converted into nitric acid as water vapour in the flue gas is condensed at low temperature. Final conversion takes places at higher pressure in the raw gas compression section of the CPU.

On the other hand, CO formation could be a potential issue in oxy-combustion process as CO is expected to remain in the inert gas stream and its formation could be enhanced with respect to the conventional air fired gas turbine due to the limited excess oxygen required for the combustion. In this case, the use of dedicated catalytic systems, e.g. within the Heat Recovery Steam Generator, could be considered to provide additional flexibility to the operation of the plant.

3. Flexible operation of oxy-turbine cycles

3.1 Peak load market

Participation to the peak load market may increase significantly the economic return of the project. Similarly to the conventional combined cycles, power production in the oxy-turbine power plants can be increased during peak electricity demand hours by:

- Gas Turbine over-firing.
- HRSG post-firing (when applicable).

3.1.1 Gas Turbine over-firing

Over-firing of the gas turbine during peak load events allows a power generation that is a few percentage points higher than the base load. During this operation the metal temperatures of some components increase, so prolonged operation at peak load reduces the operating life of the machine and leads to more frequent maintenance and replacement of hot-gas path components, thus increasing the plant maintenance costs.

If downstream systems are properly designed (e.g. overcapacity of the CPU to handle the higher amount of flue gas/CO_2, ASU design capable to cover a peak demand higher than base load requirement), the over-firing operation of the gas turbine is applicable in principle to all the oxy-fuel cycles.

3.1.2 HRSG post-firing

HRSG post-firing could be considered only for the oxy-turbine cycle including a steam cycle in its design configuration. For the SCOC-CC, the same considerations as for the conventional combined cycle can be made.

In fact, firing additional fuel in the post-firing system of the HRSG increases steam generation and consequently the steam turbine power output, if required during peak load hours. This reduces the overall plant efficiency, but increases the net plant electricity production and, therefore, allows the plant covering the higher production requirements, when needed.

As the post firing system acts on the steam generation and the steam turbine performance characterised by lower ramp rates compared to the gas turbines, the capability to respond quickly to an increased demand is slower using the HRSG post-firing if compared with the gas turbine over-firing mode.

The addition of post firing in HRSG increases the investment cost of the project both for the HRSG and the steam turbine, but also for the ASU and CPU, whose sizes should be increased due to the additional fuel gas fed to the post-firing that corresponds to additional oxygen consumption.

Similar considerations are applicable also to the GRAZ and Revised CES cycles because the post firing in the HRSG would increase the steam production and hence the power generation of the back pressure steam turbine. In addition, higher steam production for turbine blade cooling implies lower recycling gas flowrate, reducing recycle gas compressor consumptions and finally increasing the power generation from the gas turbine.

3.2 Tuning power consumptions

Another way of increasing the plant net power output is to reduce its internal power consumption, when electricity prices are high.

In oxy-turbine power plants, the power consumption related to the CO_2 purification (including compression) and to the cryogenic separation of oxygen in the ASU is significant.

A way to reduce the energy penalty related to the ASU power demand is providing liquid oxygen storage in order to temporarily avoid the base load operation of the ASU, with the plant still operating at full load, increasing the electricity exported to the grid. In fact, by over-sizing the ASU it is possible to produce extra oxygen during periods of low electricity requirements from the market, providing storage of liquid product, while increasing the auxiliary consumptions. When the market requires a higher electricity generation, the ASU can be operated at partial load, while the rest of the plant is running at full load. This reduces the auxiliary consumptions, increasing the net electricity exported to the grid. The increase of the investment cost is related to the extra-capacity required for the ASU and oxygen storage facilities.

Another possibility could be storing the CO_2 rich stream (upstream CO_2 purification) to avoid the energy penalty associated to the CO_2 compression, without increasing CO_2 emissions from the plant.

CLIENT : IEAGHG
PROJECT NAME : OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME : INDUSTRIAL AND NICHE APPLICATIONS OF OXY-TURBINES
FWI CONTRACT : 1-BD-0764 A

ISSUED BY : N. FERRARI
CHECKED BY : L. MANCUSO
APPROVED BY : L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

Oxy-turbines may be particularly well suited to certain industrial applications and niche markets. Scope of this chapter is to provide a high-level technical and economic assessment of the most promising applications of oxy-turbines, as listed below:

- **Use of high-CO_2 natural gas**
  Oxy-turbines could be an attractive option for natural gas with high CO_2 concentrations because the impact on the plant configuration and costs related to the capture for the additional CO_2 contained in natural gas is lower. In addition, as oxy-turbines use recycled CO_2 as the working fluid instead of air, the additional CO_2 content in the natural gas has a limited impact on gas turbine performance and design.
- **Provision of CO_2 for EOR**
  CO_2 from utility scale power plants can be used for EOR, but in some cases it may be preferable to build smaller local power plants near to the EOR fields. Oxy-turbine cycles, 200 MWe size, could be suitable for this application.
- **Energy integration with industrial sites**
  Some oxy-turbine cycles have a requirement for an external source of low to moderate temperature heat and others are a net producer of such heat. Large industrial plants such as oil refineries, petrochemical plants and steel mills are sources of low grade heat, some of which cannot be utilised efficiently, while excess heat from the oxy-turbine cycle can be used for district heating applications.
- **Applications requiring compact plants**
  In general, some oxy-turbine plants are expected to be significantly more compact than conventional power plants with post combustion capture of CO_2. This could be a significant advantage for some energy users, for example industrial sites with limited space, off-shore oil and gas production and some other applications which may require CO_2 emission abatement in the longer term, such as ship propulsion.

2. Use of high-CO_2 natural gas

This section provides a high-level technical and economic assessment of oxy-turbine combined cycles fed by high-CO_2 natural gas.

The following basic design assumptions have been taken:

- The oxy-turbine cycle is based on the Modified S-Graz cycle, same gas turbine thermal duty as the study case 3b.
- Two different natural gas compositions are considered: 1) 10% CO_2 content, 2) 70% CO_2 content. The impact on key design features and plant performances is shown with respect to base study case (2% CO_2 content).

The main characteristics of the natural gas considered for these two cases are summarized in the following Table 1.

Table 1. Natural gas composition

Natural Gas analysis, vol%	Base case	10% CO_2 content	70% CO_2 content
Methane	89.0	81.73	27.24
Ethane	7.0	6.43	2.14
Propane	1.0	0.92	0.31
Butane	0.1	0.09	0.03
Pentane	0.01	0.01	0.00
CO_2	2.0	10.0	70.0
Nitrogen	0.89	0.82	0.27
Total	**100.00**	**100.00**	**100.00**
HHV, MJ/kg	51.473	42.304	7.877
LHV, MJ/kg	46.502	38.215	7.115

2.1. Process description

The description reported in this section focuses only on the design changes from the study case 3b, related to the use of natural gas with higher CO_2 content.

For all the other units, reference shall be made to the base case description, included in chapter D.4.

2.1.1. *Unit 3000 – Power Island*

The same unit configuration as the base case 3b is considered, with two power generation trains, each including:

- One F-class equivalent oxy-fired gas turbine.

- One heat recovery steam generator (HRSG), generating steam at one pressure level.
- One back pressure steam turbine, expanding the steam to the pressure level required for turbine blades cooling.
- Wet flue gas compression and low pressure steam cycle.
- Recycled gas compression train (included in the gas turbine package).

The main consequence of the increased content of the CO_2 in the natural gas is the increased mass flowrate of the fuel for a given thermal input to the gas turbine. The higher inert content within the fuel increases the thermal inertia in the combustion chamber, reducing the amount of recycled stream required to control the combustion outlet temperature at its design value of 1533°C. The higher the CO_2 content in the natural gas fuel, the lower the recycle flowrate and consequently the size and the power consumptions of the recycle gas compressors.

In addition, the wet flue gas flowrate, to be compressed from atmospheric pressure to around 2 bar for water removal upstream of the CPU, increases with the CO_2 content in the natural gas, consequently increasing the size and the consumptions of the relevant compressors.

The following Table 2 summarises the main gas turbine operating parameters, affected by the different CO_2 content in the natural gas fuel.

Table 2. Gas turbine design and operating parameters

		Case 3b (2% CO_2 content)	**10% CO_2 content**	**70% CO_2 content**
Design parameters				
COT	°C	1533	1533	1533
Fuel thermal input (each GT)	MWth	768	768	768
Operating parameters				
NG feed	t/h	59.47	72.37	388.7
Recycle flowrate	t/h	752.5	749.6	633.8
HTT inlet flowrate	t/h	1171.1	1179.6	1341.6
HTT inlet composition				
CO_2	%mol	13.16	14.17	36.46
H_2O	%mol	85.53	84.52	62.31
N_2	%mol	0.37	0.37	0.33
O_2	%mol	0.41	0.41	0.42
Ar	%mol	0.53	0.53	0.48

2.1.2. *Unit 5000 – Air Separation Unit*

As the thermal input to the gas turbine is the same as the base case, the different CO_2 content in the natural gas does not lead to significant differences for the oxygen demand and the Air Separation Unit design configuration.

The oxygen supply from the ASU slightly increases with the increase of the CO_2 content in the natural gas due to the reduction of the oxygen within the recycled gas stream. However, the impact on the ASU power demand is negligible.

2.1.3. *Unit 4000 – CO_2 compression and purification*

The CO_2 fed to the power island with the natural gas is recovered in the net CO_2 product stream exiting the power island and fed to the downstream CPU for treatment and compression. Therefore, the capacity and the configuration of the CPU highly depend on the CO_2 content in the fuel gas.

For all the different alternatives analysed in the study, Table 3 summarises the main design data affecting the capacity of the systems included in the CPU. The raw flue gas compression (1 - 34 bar) and the TSA unit capacity and the auto-refrigerated inerts removal section configuration depends on the flowrate and composition of the flue gas entering the CPU, while the final compression system size is proportional to the carbon content in the natural gas, and consequently only the CO_2 flowrate downstream inerts removal.

Table 3. CPU capacity design parameters

		Case 3b (2% CO_2 content)	10% CO_2 content	70% CO_2 content
Design parameters				
Flue gas to CPU	kmol/h	7,865	8,635	23,340
Net CO_2 product to storage	kmol/h	6,450	6,995	19,960

The inerts content (e.g. oxygen, argon and nitrogen) in the net CO_2-rich stream to the CPU mainly depends on the oxygen flowrate and purity. As a consequence of the increased CO_2 content in the natural gas and hence in the CO_2-rich stream, the mole fraction of the inert components decreases, affecting the configuration of the auto-refrigerated inerts removal section.

Considering only a limited increase of the CO_2 content in the natural gas (case 1 with 10% CO_2 content), the auto-refrigerated inerts removal section can be based on the same configuration of the base case, as described in chapter D, section 2.3, including a flash vessel upstream the distillation column required to meet the maximum oxygen content limit in the CO_2 product.

On the other hand, considering a natural gas with very high CO_2 content (case 2 with 70% CO_2 content), the auto-refrigerated inerts removal section shall be based on the configuration as described in chapter D, section 2.3.1, i.e. including only the distillation column for the CO_2 purification, without the upstream flash vessel.

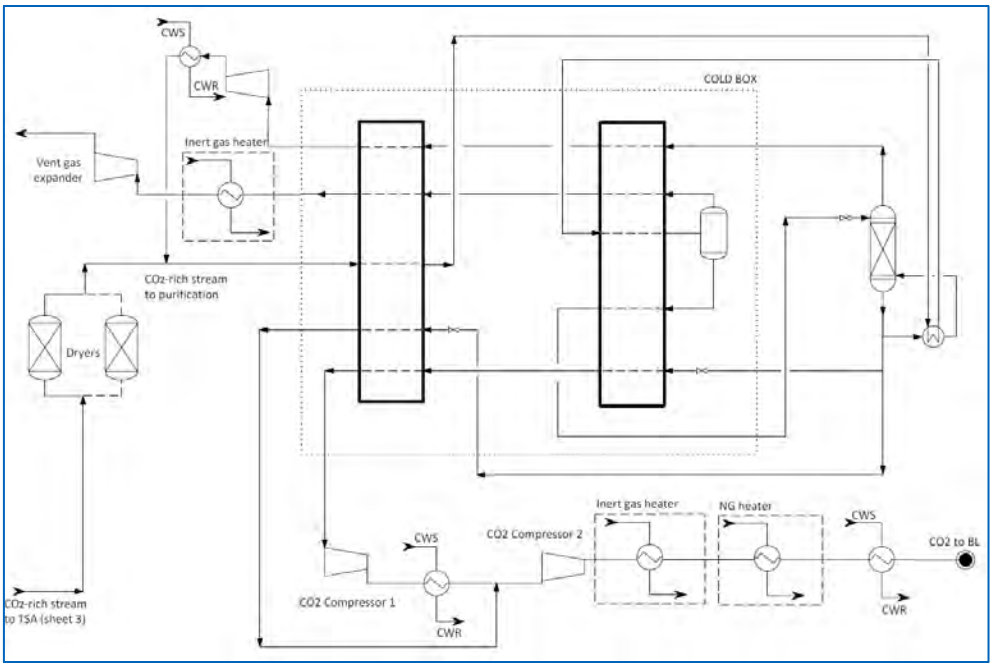


Figure 1. Standard CPU configuration (including flash vessel)

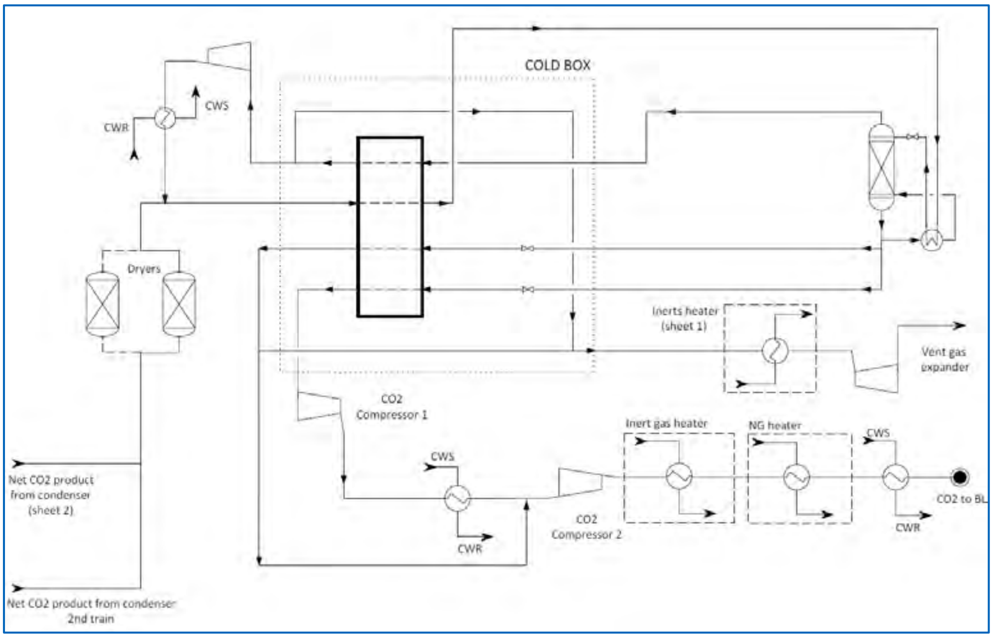


Figure 2. High-purity CPU configuration (no flash vessel)

2.2. Overall performance

The following table shows the overall performances of the cases listed above, including CO_2 balance and removal efficiency, compared with the base study case 3b.

		BASE CASE	10% CO_2 content	70% CO_2 content
PERFORMANCES COMPARISON				
Natural Gas flow rate (A.R.)	t/h	118.9	144.7	777.4
Natural Gas LHV	kJ/kg	46502	38215	7115
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**	**1536**
HTT turbine power output	MWe	1206.1	1202.0	1117.7
HPT turbine power output	MWe	95.5	95.2	91.1
LPT turbine power output	MWe	117.4	116.3	87.7
Turbine recycle gas compressors	MWe	-423.9	-418.1	-292.5
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)	**MWe**	**995.2**	**995.4**	**1004.1**
Oxy-turbine cycle	MWe	46.3	47.6	73.1
Air separation unit	MWe	147.1	147.1	147.4
CO2 purification and compression unit Compressor	MWe	36.4	39.4	107.8
Expander	MWe	-2.9	-3.0	-6.0
Utility & Offsite Units	MWe	10.6	10.6	9.9
ELECTRIC POWER CONSUMPTION	**MWe**	**237.5**	**241.7**	**332.2**
NET ELECTRIC POWER OUTPUT	**MWe**	**757.7**	**753.8**	**671.9**
(Step Up transformer efficiency = 0.997%) (B)	MWe	755.5	751.5	669.9
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**64.8%**	**64.8%**	**65.3%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.2%**	**48.9%**	**43.6%**
Equivalent CO_2 flow in fuel	kmol/h	7159	7745	22122
Captured CO_2	kmol/h	6449	6980	19904
Emitted CO_2	kmol/h	710	765	2217
CO_2 removal efficiency	**%**	**90.1**	**90.1**	**90.0**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**	**2.04**	**2.29**
CO_2 emission per net power production	**kg/MWh**	**41.5**	**44.4**	**144.1**

Based on the above performance data and the design features changes summarised in the previous section, the following considerations can be drawn:

- With a constant thermal input to the gas turbine and with the same combustion outlet temperature, both the turbine expander production and the recycled gas compressor power demand decrease with an increased CO_2 content in the natural gas fuel, resulting in a higher gas turbine net power production.

		Case 3b (2% CO_2 content)	**10% CO_2 content**	**70% CO_2 content**
HTT	MWe	1206.1	1202.0	1117.7
Recycle gas compressors	MWe	423.9	418.1	292.5
Gas Turbine NPO	**MWe**	**782.2**	**783.9**	**825.2**

- Production from the back-pressure steam turbine decreases at higher CO_2 content in the natural gas as heat recovery from recycle gas compressor intercoolers is lower.
- Production from the condensing steam turbine decreases at higher CO_2 content in the natural gas as the amount of water in the wet flue gas, whose condensation in the compressors after-coolers provides the heat required for low pressure steam generation, is lower.
- The main impact on the plant total power consumption is due to the increased capacity and consequently higher power demand of the raw gas compressor and the CPU compressors, as the CO_2 flowrate to be purified and compressed is higher. The increased power production from the inert gas expander (due to the higher CO_2 content in the vent stream) is negligible, compared to the additional power demand.
- Utility and offsites consumption slightly decreases with the increase of the CO_2 content in the natural gas, mainly due to the reduced cooling water requirements for the steam cycle condenser.
- As a result from the above considerations, while the gross electrical efficiency of the cycle increases by around 0.5 percentage point, increasing the CO_2 content in the natural gas from 2% to 70%, the net electrical efficiency drops from around 49% to less than 44%, mainly due to the massive CO_2 flowrate to be treated and compressed in the CPU.

2.3. Investment cost estimate

The overleaf tables show the Total Plant Cost summary of the cases listed above, while Table 4 shows the impact of the increased CO_2 content in the natural gas feed on the TPC with respect to the base study case.

Table 4. TPC comparison

		Case 3b (2% CO_2 content)	10% CO_2 content	70% CO_2 content
Power island	M€	515.6	516.3	520.4
CO_2 purification unit	M€	98.3	103.4	176.3
ASU	M€	320.9	320.9	320.9
Utility unit	M€	201.7	201.7	193.9
Total Plant cost	**M€**	**1,136.5**	**1,142.3**	**1,211.5**
Specific TPC	**€/kW**	**1,500**	**1,520**	**1,810**

The increased TPC is related only to the bigger size of the CPU as keeping constant the capture rate, the amount of CO_2 to be captured increases with CO_2 content in the natural gas. Impact on the cost of the power island, the ASU and on the utilities is negligible.

The detailed financial analysis of these high-CO_2 cases has not been carried out but, while a higher LCOE is expected due to the higher TPC and lower electrical efficiency, the cost per ton of CO_2 avoided is expected to be lower because the quantity of CO_2 captured in the 70% CO_2 case is more than 3 times higher than in the base case.

OXY-TURBINE POWER PLANT
CASE 3 - MODIFIED S-GRAZ CYCLE
(10% CO2 in NG)

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 POWER ISLAND	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	249,200,000	62,400,000	170,800,000	108,000,000	590,400,000	1) Gross power output: MW 995.4 Specific cost €/KW : 1,150
2	CONSTRUCTION and OTHER COSTS	183,600,000	20,300,000	95,900,000	53,900,000	353,700,000	2) Total Net Power : MW 751.5
3	EPC SERVICES	36,600,000	11,300,000	25,000,000	21,500,000	94,400,000	Average Cost €/KW : 1,520
4	**TOTAL INSTALLED COST**	**469,400,000**	**94,000,000**	**291,700,000**	**183,400,000**	**1,038,500,000**	
5	PROJECT CONTINGENCY	46,900,000	9,400,000	29,200,000	18,300,000	103,800,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**516,300,000**	**103,400,000**	**320,900,000**	**201,700,000**	**1,142,300,000**	

OXY-TURBINE POWER PLANT
CASE 3 - MODIFIED S-GRAZ CYCLE
(70% CO2 in NG)

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 POWER ISLAND	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS	
1	DIRECT MATERIAL	251,200,000	106,400,000	170,800,000	103,800,000	632,200,000	1) Gross power output: MW	1004.1
							Specific cost €/KW :	1,210
2	CONSTRUCTION and OTHER COSTS	185,000,000	34,600,000	95,900,000	51,800,000	367,300,000		
							2) Total Net Power : MW	669.9
3	EPC SERVICES	36,900,000	19,300,000	25,000,000	20,700,000	101,900,000	Average Cost €/KW :	1,810
4	**TOTAL INSTALLED COST**	**473,100,000**	**160,300,000**	**291,700,000**	**176,300,000**	**1,101,400,000**		
5	PROJECT CONTINGENCY	47,300,000	16,000,000	29,200,000	17,600,000	110,100,000		
6	PROCESS CONTINGENCY	-	-	-	-	-		
7	**TOTAL PLANT COST (TPC)**	**520,400,000**	**176,300,000**	**320,900,000**	**193,900,000**	**1,211,500,000**		

3. Provision of CO_2 for EOR

This section provides a high-level technical and economic assessment of the use of oxy-fuel gas turbine power plants for EOR applications.

The following basic design assumptions have been taken:

- The power island configuration is based either on the SCOC-CC or on the Modified S-GRAZ cycle.
- In both cases, the plant is based on a single gas turbine plus steam turbine configuration.
- Plant capacity is fixed in order to have around 200 MWe gross power production, i.e. an E-class equivalent gas turbine in terms of power output, volume flowrate and gas turbine geometry.
- Natural gas characteristics are the same as in the study base cases.

3.1. Process description

The description reported in this section focuses only on the design changes from the relevant study base cases.

For the process description of the main plant units, reference shall be made to the base case descriptions, which are case 1 for the SCOC-CC and case 3b for the Modified S-Graz cycle, included respectively in chapter D.1 and chapter D.4.

For both cycles, the main difference is the power island arrangement, which for the EOR application is based on one (1) E-class equivalent oxy-fired gas turbine and one (1) steam turbine. As a consequence of the lower natural gas and oxygen consumptions, also a single ASU train can be considered.

The main design parameters of the gas turbines are given below.

3.1.1. SCOC-CC cycle

The power island is mainly composed of:

- One E-class equivalent oxy-fired gas turbine.
- One heat recovery steam generator (HRSG), generating steam at two levels of pressure, plus a LP integrated deaerator.
- One recycle gas indirect contact cooling system.
- One steam turbine, water-cooled and condensing type, common to the two parallel trains.

The gas turbine expander is designed considering the same design criteria (COT, TIT and pressure ratio) of the F-class equivalent considered in the reference case 1.

A higher rotational speed of 4600 RPM is required with respect to bigger size machines, with the same number of stages (5), so to have acceptable Mach number in the gas stream and peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine. As a consequence, a gearbox is required.

3.1.2. *Modified S-GRAZ cycle*

The power island is mainly composed of:

- One E-class equivalent oxy-fired gas turbine.
- One heat recovery steam generator (HRSG), generating steam at one pressure level.
- One back pressure steam turbine expanding the steam to the pressure level required for turbine blades cooling.
- Wet flue gas compression and low pressure steam cycle.
- Recycled gas compression train (part of the gas turbine package).

The gas turbine expander is designed considering the same design criteria (COT, TIT and pressure ratio) of the F-class equivalent considered in the reference case 3b.

With respect to the double section expander foreseen for the bigger size machine, the most suitable expander for this size and flowrates includes one section having 6 stages at a rotational speed of 6100 RPM, so to have acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine. As a consequence, a gearbox is required.

3.2. Overall performance

The following table shows the overall performance of the two schemes developed for the EOR application, including CO_2 balance and removal efficiency.

The lower efficiency, with respect to the reference cases, is related to the lower efficiency of the smaller gas turbine, with respect to the F-class equivalent considered for the bigger utility plant.

Table 5. SCOC-CC for EOR: performance

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Sep-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

SCOC-CC for EOR

OVERALL PERFORMANCES		
Natural Gas flow rate	t/h	25.0
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**323**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**357**
Gas turbine power output (@ gen terminals)	MWe	126.5
Steam turbine power output (@ gen terminals)	MWe	73.3
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**199.8**
Oxy-turbine cycle	MWe	2.6
Air separation unit	MWe	30.7
CO2 purification and compression unit	MWe	8.5
Utility & Offsite Units	MWe	2.8
ELECTRIC POWER CONSUMPTION	**MWe**	**44.5**
NET ELECTRIC POWER OUTPUT	**MWe**	**155.3**
(Step Up transformer efficiency = 0.997%) (B)	MWe	154.8
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**61.9%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**48.0%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**56.0%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**43.5%**
Equivalent CO_2 flow in fuel	kmol/h	1503
Captured CO_2	kmol/h	1363
CO_2 removal efficiency	**%**	**90.6**
Fuel Consumption per net power production	**MWth/MWe**	**2.08**
CO_2 emission per net power production	**kg/MWh**	**39.7**

Table 6. Modified S-Graz for EOR: performance

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Sep-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Modified S-GRAZ cycle for EOR

OVERALL PERFORMANCES		
Natural Gas flow rate (A.R.)	t/h	25.0
Natural Gas LHV	kJ/kg	46502
Natural Gas HHV	kJ/kg	51473
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**323**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**357**
HTT turbine power output	MWe	260.5
HPT turbine power output	MWe	19.0
LPT turbine power output	MWe	24.7
Turbine recycle gas compressors	MWe	-98.3
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**205.9**
Oxy-turbine cycle	MWe	9.8
Air separation unit	MWe	31.6
CO2 purification and compression unit	MWe	7.0
Utility & Offsite Units	MWe	2.7
ELECTRIC POWER CONSUMPTION	**MWe**	**51.0**
NET ELECTRIC POWER OUTPUT	**MWe**	**154.9**
(Step Up transformer efficiency = 0.997%) (B)	MWe	154.4
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**63.8%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**47.9%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**57.7%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**43.4%**
Equivalent CO_2 flow in fuel	kmol/h	1503
Captured CO_2	kmol/h	1356
CO_2 removal efficiency	**%**	**90.2**
Fuel Consumption per net power production	**MWth/MWe**	**2.09**
CO_2 emission per net power production	**kg/MWh**	**42.9**

3.3. Investment cost estimate

The overleaf tables show the Total Plant Cost summary of the cases listed above, while Table 7 and Table 8 shows the comparison between the capital costs figures of smaller power plants suited for EOR application with respect to the utility scale power plants analysed as study main cases.

Table 7. SCOC-CC for EOR – TPC comparison

		Case 1 (Two eq. F-class GT)	**SCOC-CC for EOR** (One eq. E-class GT)
Power island	M€	482.6	133.0
CO2 purification unit	M€	106.3	35.5
ASU	M€	318.5	86.7
Utility unit	M€	203.5	64.7
Total Plant cost	**M€**	**1,110.9**	**319.9**
Specific TPC	**€/kW**	**1,470**	**2,070**

Table 8. Modified S-Graz cycle for EOR – TPC comparison

		Case 3b (Two eq. F-class GT)	**S-Graz for EOR** (One eq. E-class GT)
Power island	M€	515.6	135.7
CO2 purification unit	M€	98.3	33.1
ASU	M€	320.9	87.6
Utility unit	M€	201.7	62.8
Total Plant cost	**M€**	**1,136.5**	**319.2**
Specific TPC	**€/kW**	**1,500**	**2,070**

OXY-TURBINE POWER PLANT
SCOC-COMBINED CYCLE for EOR

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 COMBINED CYCLE	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	64,200,000	21,500,000	46,100,000	34,600,000	166,400,000	1) Gross power output: MW 199.8 Specific cost €/kW : 1,600
2	CONSTRUCTION and OTHER COSTS	47,300,000	6,900,000	25,800,000	17,300,000	97,300,000	2) Total Net Power : MW 154.8
3	EPC SERVICES	9,400,000	3,900,000	6,900,000	6,900,000	27,100,000	Specific cost €/kW : 2,070
4	**TOTAL INSTALLED COST**	**120,900,000**	**32,300,000**	**78,800,000**	**58,800,000**	**290,800,000**	
5	PROJECT CONTINGENCY	12,100,000	3,200,000	7,900,000	5,900,000	29,100,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**133,000,000**	**35,500,000**	**86,700,000**	**64,700,000**	**319,900,000**	

OXY-TURBINE POWER PLANT
MODIFIED S-GRAZ CYCLE FOR EOR

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 POWER ISLAND	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	65,500,000	20,000,000	46,500,000	33,600,000	165,600,000	1) Gross power output: MW 205.9 Specific cost €/kW : 1,550
2	CONSTRUCTION and OTHER COSTS	48,300,000	6,500,000	26,200,000	16,800,000	97,800,000	2) Total Net Power : MW 154.4 Specific cost €/kW : 2,070
3	EPC SERVICES	9,600,000	3,600,000	6,900,000	6,700,000	26,800,000	
4	**TOTAL INSTALLED COST**	**123,400,000**	**30,100,000**	**79,600,000**	**57,100,000**	**290,200,000**	
5	PROJECT CONTINGENCY	12,300,000	3,000,000	8,000,000	5,700,000	29,000,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**135,700,000**	**33,100,000**	**87,600,000**	**62,800,000**	**319,200,000**	

4. Energy integration with industrial sites or district heating

Some oxy-turbine cycles require an external source of low to moderate temperature heat or, in contrast, can be a net producer of such heat.

This section provides a high-level technical and economic assessment of the possible thermal integration of a oxy-turbine combined cycle with a large heat source or a heat consumer, in order to provide an initial assessment of any benefits of applying oxy-turbines to district heating or at industrial sites having 'free' low grade heat.

The following cases have been assessed.

A. Modified S-Graz cycle for district heating application

   The oxy-turbine cycle is based on the Modified S-Graz cycle, same gas turbine thermal duty of the study main case 3b. The heat available for the low pressure steam cycle in the base cycle is used as heating source in a district heating circuit.

B. Modified S-Graz cycle applied at industrial sites having 'free' low grade heat

   The oxy-turbine cycle is based on the Modified S-Graz cycle, same gas turbine thermal duty of the study main case 3b. The heat available from the hypothetical industrial site is used for steam generation to be expanded in the low pressure steam cycle.

C. NET Power applied at industrial sites having 'free' low grade heat

   The oxy-turbine cycle is based on the NET Power cycle, same gas turbine thermal duty of the study main case 2. The heat available from the hypothetical industrial site is used for fully exploiting the potentiality of the main heat exchanger, in order to increase the temperature of the recycle streams (oxidant and CO_2-rich stream) at the highest possible figure (assumed 5°C approach with respect to the turbine exhaust gas).

4.1. Process description

The description reported in this section focuses only on the design changes from that of the reference study case, i.e. case 3b for the cases A and B described in the above section and case 2b for case C.

For all the other units, reference shall be made to the base case description, included in chapter D.5 (cases A and B) and chapter D.3 (cases C).

4.1.1. *Unit 3000 – Power Island*

Case A

As for the base case, the power island is composed of two power generation trains, each including:

- One F-class equivalent oxy-fired gas turbine.
- One heat recovery steam generator (HRSG) generating steam at one pressure level.
- One back pressure steam turbine expanding the steam to the pressure level required for turbine blades cooling.
- Wet flue gas compression.
- Recycled gas compression train (included in the gas turbine package)

While in the base case the heat available in the inter and after coolers of the wet flue gas compression section is recovered generating steam to be expanded in the low pressure steam cycle, in this case the heat is recovered heating up cold water from a district heating circuit.

Main design parameters of the district heating circuit are summarised in the following Table 9.

Table 9. District heating circuit main parameters

Thermal duty (available from wet flue gas compression section)	MWth	826
Cold water temperature	°C	50
Hot water temperature	°C	85
Water flowrate (each train)	t/h	10,100

Case B

The same unit configuration of the base case 3b is considered, with two power generation trains, each including:

- One F-class equivalent oxy-fired gas turbine.
- One heat recovery steam generator (HRSG) generating steam at one pressure level.
- One back pressure steam turbine expanding the steam to the pressure level required for turbine blades cooling.
- Wet flue gas compression and low pressure steam cycle.
- Recycled gas compression train (included in the gas turbine package)

The low pressure steam from the nearby industrial site is mixed with steam generated in the wet flue gas compressors inter and after coolers, and expanded in the condensing steam turbine, increasing the capacity of both the turbine and the condenser.

The heat source from the industrial site is assumed equal to 826 MWth, comparable with the heat made available to the low pressure steam cycle by the oxy-turbine cycle itself in the wet flue gas compression section. The imported steam flowrate and conditions are summarised in the following Table 10.

Table 10. Low pressure steam from industrial site

Steam flowrate (each train)	t/h	545
Steam temperature	°C	190
Steam pressure	bar	0.75

Case C

The same unit configuration of the base case 2 is considered, with two power generation trains, each including:

- One F-class equivalent oxy-fired gas turbine.
- One main heat exchanger, for recycled gas pre-heating.
- One recycled gas compression train (part of the gas turbine package).

Saturated low pressure steam at the conditions summarised in the following Table 11 is used as heating medium in the main heat exchanger of the NET Power cycle. The following has been considered to define the steam flowrate and conditions:

- The amount of steam and the pressure level is selected in order to provide the required amount of steam to heat the recycle streams up to the maximum temperature, assumed for this case 5°C lower than the exhaust gas temperature from the gas turbine as numerically shown below.

		Base case	With steam from OSBL
Flue gas exhaust temperature at MHE inlet	°C	739	739
Recycle streams temperature	°C	720	734

- Steam pressure (and consequently steam temperature) and water outlet temperature is selected in order to keep the minimum heat exchanger approach at the design value of 4.5°C as selected in the base case.

Table 11. Low pressure steam from industrial site

Steam flowrate (each train)	t/h	44.3
Steam temperature	°C	188
Steam pressure	bar	12
Cold water outlet temperature	°C	60
Duty from OSBL (each train)	MWth	31.1

4.1.2. *Unit 5000 – Air Separation Unit*

For each of the above cases, the thermal input to the gas turbine is the same as the study reference cases. As for that, the oxidant stream required is the same as the reference case and consequently no changes are required in the Air Separation Unit capacity and design configuration.

4.1.3. *Unit 4000 – CO_2 compression and purification*

As for each of the above cases, the natural gas fed to the gas turbine is the same as the study reference cases, the generated CO_2 is same as the relevant base case and consequently no changes are required in the capacity and design configuration of the CO_2 compression and purification unit.

4.2. Overall performance

4.2.1. Modified S-Graz cycle based cases

The following table shows the overall performances of cases A and B, including CO_2 balance and removal efficiency, compared with the base study case 3b.

		BASE CASE	DISTRICT HEATING CASE	HEAT INPUT FROM OUTSIDE B.L.
PERFORMANCES COMPARISON				
Natural Gas flow rate (A.R.)	t/h	118.9	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**	**1536**
ADDITIONAL THERMAL INPUT	**MWth**	**-**	**-**	**826**
HTT turbine power output	MWe	1206.1	1209.6	1206.1
HPT turbine power output	MWe	95.5	98.9	95.5
LPT turbine power output	MWe	117.4	-	238.5
Turbine recycle gas compressors	MWe	-423.9	-440.8	-423.9
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)	**MWe**	**995.2**	**867.8**	**1116.3**
Oxy-turbine cycle	MWe	46.3	33.1	46.9
Air separation unit	MWe	147.1	147.1	147.1
CO2 purification and compression unit	MWe	33.5	33.5	33.5
Utility & Offsite Units	MWe	10.6	3.6	17.6
ELECTRIC POWER CONSUMPTION	**MWe**	**237.5**	**217.3**	**245.0**
NET ELECTRIC POWER OUTPUT	**MWe**	**757.7**	**650.5**	**871.3**
(Step Up transformer efficiency = 0.997%) (B)	MWe	755.5	648.6	868.7
THERMAL DUTY TO DISTRICT HEATING (D)	**MWth**	**-**	**826**	**-**
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**64.8%**	**56.5%**	**72.7%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**49.2%**	**42.2%**	**56.5%**
Thermal efficiency (D/A x 100) (based on LHV)	**%**	**na**	**48.6%**	**na**
CHP efficiency ((B+D)/A x 100) (based on LHV)	**%**	**na**	**90.8%**	**na**
Fuel Consumption per net power production	**MWth/MWe**	**2.03**	**2.37**	**1.77**
CO_2 emission per net power production	**kg/MWh**	**41.5**	**48.3**	**36.1**

Based on the above performance data and the design features changes summarised in the previous section, the following considerations can be drawn for the Modified S-Graz cycle:

- As the ‘high temperature’ heat recovery for steam generation for blade cooling purpose and ‘low temperature’ heat recovery for efficiency improvements are performed in two different sections of the cycle, they can be easily separated and the ‘low temperature’ section substituted by a district heating circuit. The cycle efficiency is around 42% (LHV basis), while the CHP efficiency is higher than 90%.

- On the other hand, as the steam cycle included in the Modified-S-Graz cycle is based on very low pressure steam (i.e. 75 kPa), even 'low grade' heat is suited for generating steam at the proper temperature and pressure level.

4.2.2. *NET Power cycle based case*

The following table shows the overall performances of case C, including CO_2 balance and removal efficiency, compared with the base study case 2.

		BASE CASE	HEAT INPUT FROM OUTSIDE B.L.
DESIGN FEATURES			
Recycle gas flowrate (recycle + oxydant)	t/h	4,493	4,642
Recycle gas final temperature	°C	720	734
Flue gas to gas turbine	t/h	4,553	4,701
PERFORMANCES COMPARISON			
Natural Gas flow rate (A.R.)	t/h	118.9	118.9
Natural Gas LHV	kJ/kg	46502	46502
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**1536**	**1536**
ADDITIONAL THERMAL INPUT (D)	**MWth**	**-**	**62**
HP turbine power output	MWe	1263.9	1304.1
Turbine recycle gas compressors	MWe	-207.9	-214.9
GROSS ELECTRIC POWER OUTPUT (@ gen terminals) (C)	**MWe**	**1056.0**	**1089.2**
Oxy-turbine cycle (including NG compressor)	MWe	13.9	14.0
Air separation unit + Oxygen compressor	MWe	170.9	170.9
CO2 purification and compression unit	MWe	12.4	12.4
Utility & Offsite Units	MWe	10.4	10.6
ELECTRIC POWER CONSUMPTION	**MWe**	**207.5**	**207.9**
NET ELECTRIC POWER OUTPUT	**MWe**	**848.4**	**881.3**
(Step Up transformer efficiency = 0.997%) (B)	MWe	845.9	878.6
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**68.7%**	**70.9%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**55.1%**	**57.2%**
Gross electrical efficiency (C/(A+D) x 100) (based on LHV)	**%**	**68.7%**	**68.1%**
Net electrical efficiency (B/(A+D) x 100) (based on LHV)	**%**	**55.1%**	**55.0%**
Fuel Consumption per net power production	**MWth/MWe**	**1.82**	**1.75**
CO_2 emission per net power production	**kg/MWh**	**37.0**	**35.6**

Based on the above performance data and the design features changes summarised in the previous section, the following considerations can be drawn:

- The additional heat made available from outside the oxy-plant battery limits allows to achieve a higher flue gas recycle temperature to the combustion chamber. As a consequence, a higher flowrate is required to control the combustion temperature at the design value of 1150°C, implying a higher gas flowrate expanded in the gas turbine and finally a higher power production.

- It has to be noted that, for this case, steam conditions are selected in order to optimise the main heat exchanger and achieve the highest possible recycle temperature, keeping the same minimum design approach of the base case. In fact, as the main heat exchanger heat-temperature profile is one of the most critical design issues of the NET power cycle, not only the amount of heat available but also the other conditions shall be properly selected.

4.3. Investment cost estimate

The overleaf tables show the Total Plant Cost summary of the Modified S-Graz cycle based cases listed above, while Table 12 show the comparison between the capital cost figures of these integrated plants and the base study case 3b.

Table 12. Modified S-Graz cycle: TPC comparison

		Case 3b	Case A DH application	Case B Heat input from OBL
Power island	M€	515.6	454.9	554.4
CO2 purification unit	M€	98.3	98.3	98.3
ASU	M€	320.9	320.9	320.9
Utility unit	M€	201.7	100.5	333.2
Total Plant cost	**M€**	**1,136.5**	**974.6**	**1,306.8**
Specific TPC	**€/kW**	**1,500**	**1,500**	**1,500**

No significant investment cost increase, associated to the use of additional heat from OBL, is expected for the NET Power based case. In this case the plant economics are mostly affected by the increased operating cost for the incoming steam.

FOSTER WHEELER

OXY-TURBINE POWER PLANT
CASE A - MODIFIED S-GRAZ CYCLE
(District heating application)

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 POWER ISLAND	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	219,500,000	59,300,000	170,800,000	53,800,000	503,400,000	1) Gross power output: MW 867.8 Specific cost €/kW : 1,120
2	CONSTRUCTION and OTHER COSTS	161,700,000	19,300,000	95,900,000	26,900,000	303,800,000	2) Total Net Power : MW 648.6
3	EPC SERVICES	32,300,000	10,800,000	25,000,000	10,700,000	78,800,000	Specific cost €/kW : 1,500
4	**TOTAL INSTALLED COST**	**413,500,000**	**89,400,000**	**291,700,000**	**91,400,000**	**886,000,000**	
5	PROJECT CONTINGENCY	41,400,000	8,900,000	29,200,000	9,100,000	88,600,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**454,900,000**	**98,300,000**	**320,900,000**	**100,500,000**	**974,600,000**	

OXY-TURBINE POWER PLANT
CASE B - MODIFIED S-GRAZ CYCLE
(Heat available from OSBL)

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: FEBRUARY 2015
REV.: 0

POS.	DESCRIPTION	UNIT 3000 POWER ISLAND	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	267,600,000	59,300,000	170,800,000	178,400,000	676,100,000	1) Gross power output: MW 1116.3 Specific cost €/kW : 1,170
2	CONSTRUCTION and OTHER COSTS	197,100,000	19,300,000	95,900,000	89,000,000	401,300,000	2) Total Net Power : MW 868.7 Specific cost €/kW : 1,500
3	EPC SERVICES	39,300,000	10,800,000	25,000,000	35,500,000	110,600,000	
4	**TOTAL INSTALLED COST**	**504,000,000**	**89,400,000**	**291,700,000**	**302,900,000**	**1,188,000,000**	
5	PROJECT CONTINGENCY	50,400,000	8,900,000	29,200,000	30,300,000	118,800,000	
6	PROCESS CONTINGENCY	-	-	-	-	-	
7	**TOTAL PLANT COST (TPC)**	**554,400,000**	**98,300,000**	**320,900,000**	**333,200,000**	**1,306,800,000**	

5. Applications requiring compact plants

Some oxy-turbine plants are expected to be significantly more compact than conventional power plants with post combustion capture of CO_2, being a significant advantage for energy users with limited space.

Main design differences, that may have a significant impact on the plot area requirements, are summarised in the following Table 13, including the reference plant without capture, the combined cycle with post-combustion capture and the oxy-fuel cycles.

As there are several differences among the oxy-turbine cycles proposed in literature that have an impact on the lay-put, two different categories have been identified.

The first category includes the oxy-turbine cycles that are based on the 'combined cycle' concept, recovering the heat available from the gas turbine exhaust flue gas for steam generation. Among the cases technically assessed in the study, this category includes the SCOC-CC, the S-GRAZ cycle and the CES. The only potential advantage of these cycles in terms of space requirements is the lower plot area of the ASU and CPU with respect to the post-combustion capture unit.

The second category includes the regenerative cycle as the NET Power. The potential for space saving considering this technology is significant, as no steam cycle is installed and the equipment, such as the gas turbine and the main heat exchanger, are more compact due to the higher operating pressure. NET Power claims for their power and CO_2 cycle a footprint of about 1/3 the size of a combined cycle with a similar power output [[1]].

The above considerations mainly focus on process units. It is expected that the oxy-turbine cycle requires a lower amount of cooling water and consequently a lower area requirement for cooling tower installation or air cooler bays installation. No particular differences are expected for the water systems (demi and raw water) as the water make-up for combined cycle power plant is negligible, while for the waste water treatment, the layout is similar to a combined cycle with post combustion capture as these plants are expected to send the same amount of water condensate from the flue gas to the treatment.

[1] R.J. Allam at al., *High efficiency and low cost of electricity generation from fossil fuels while eliminating atmospheric emissions, including carbon dioxide*, GHGT-11

Table 13. Plot area requirements: oxy-turbine vs. NGCC with post-combustion

Reference plant	**Unit/equipment**	**Plot area** (*)
NGCC w/o CCS	• Gas turbine • HRSG • Steam turbine • NG receiving system • U&O (cooling water, demi water, raw water, WWT…)	250 x 360 m^2 (1)
Plant with CCS	**Delta Unit/equipment**	**Delta plot area** (*)
NGCC with post combustion capture	(+) Post combustion capture unit (+) Compression unit	+ 250 x 130 m^2 (1) **+ 32,500 m^2**
SCOC-CC type oxy-fuel cycle	(+) ASU (+) CPU including raw gas and CO_2 compression	+ 120 x 200 m^2 + 100 x 60 m^2 **+ 30,000 m^2**
NET Power type oxy-fuel cycle	(+) ASU (+) CPU including CO_2 compression (-) HRSG (+) Main heat exchanger (-) Steam cycle	+ 120 x 200 m^2 + 50 x 60 m^2 1/3 of conventional CC **+10,000 m^2**

(*) Size are referred to a 900 MWe combined cycle, based on two F-class equivalent gas turbines

Therefore, the oxy-fuel cycle based on the NET Power scheme seems to be the most promising technology from the space requirements point of view.

Main limitation to the use of the oxy-turbine technology in compact plants is the space requirement for the ASU, which alone accounts for around 25-30% additional space with respect to the conventional combined cycle.

A possible way to overcome this constraint is the construction of the oxy-fuel power plant in a location having a nearby oxygen pipeline.

[1] IEAGHG report 2012/8, *CO_2 capture at gas fired power plants*

CLIENT	:	IEAGHG
PROJECT NAME	:	OXY-COMBUSTION TURBINE POWER PLANTS
DOCUMENT NAME	:	OXY-TURBINES COMBINED WITH COAL GASIFICATION
FWI CONTRACT	:	1-BD-0764 A

ISSUED BY	:	N. FERRARI
CHECKED BY	:	L. MANCUSO
APPROVED BY	:	L. MANCUSO

Date	Revised Pages	Issued by	Checked by	Approved by

TABLE OF CONTENT

1. Introduction

This chapter of the report includes all technical information relevant to an IGCC plant based on the GE gasification technology, including a power island based on the semi-closed oxy-combustion combined cycle (SCOC-CC), with cryogenic purification and separation of the carbon dioxide. The plant is designed to process coal, whose characteristic is shown in chapter B, and produce electric power for export to the external grid.

The configuration of the plant is based on the following main features:

- High-pressure (65 barg) GE Energy Gasification process, with slurry-feed system and Radiant Syngas Cooler (RSC);
- COS hydrolysis;
- Removal of sulphuric acid gas (H_2S) based on Selexol physical solvent process;
- Oxygen-blown Claus unit, with tail gas catalytic treatment and recycle of the treated tail gas to the AGR;
- Cryogenic purification and separation of the carbon dioxide;
- SCOC-CC based configuration based on two parallel trains, each composed of one F-class equivalent oxy-fired gas turbine and one heat recovery steam generator (HRSG), generating steam at four levels of pressure, including a LP integrated deaerator. The generated steam feeds one steam turbine, water-cooled and condensing type, common to the two parallel trains.

The SCOC-CC is selected for the development of this IGCC case including an oxy-cycle based power island, as it resembles a conventional combined cycle and hence it is conventionally used as benchmark cycle.

It is to be noted that most of the cycle developers/licensor is evaluating the application of its own cycle to the coal-fired plant, with the gas turbine firing syngas, in particular focusing on optimising the plant configuration in terms of thermal integration to enhance electrical efficiency [1,2].

[1] W. Sanz, M. Mayr, H. Jericha et al, *Thermodynamic and Economic Evaluation of an IGCC Plant Based on the Graz Cycle for CO_2 Capture*, ASME Turbo Expo 2010, Glasgow, UK

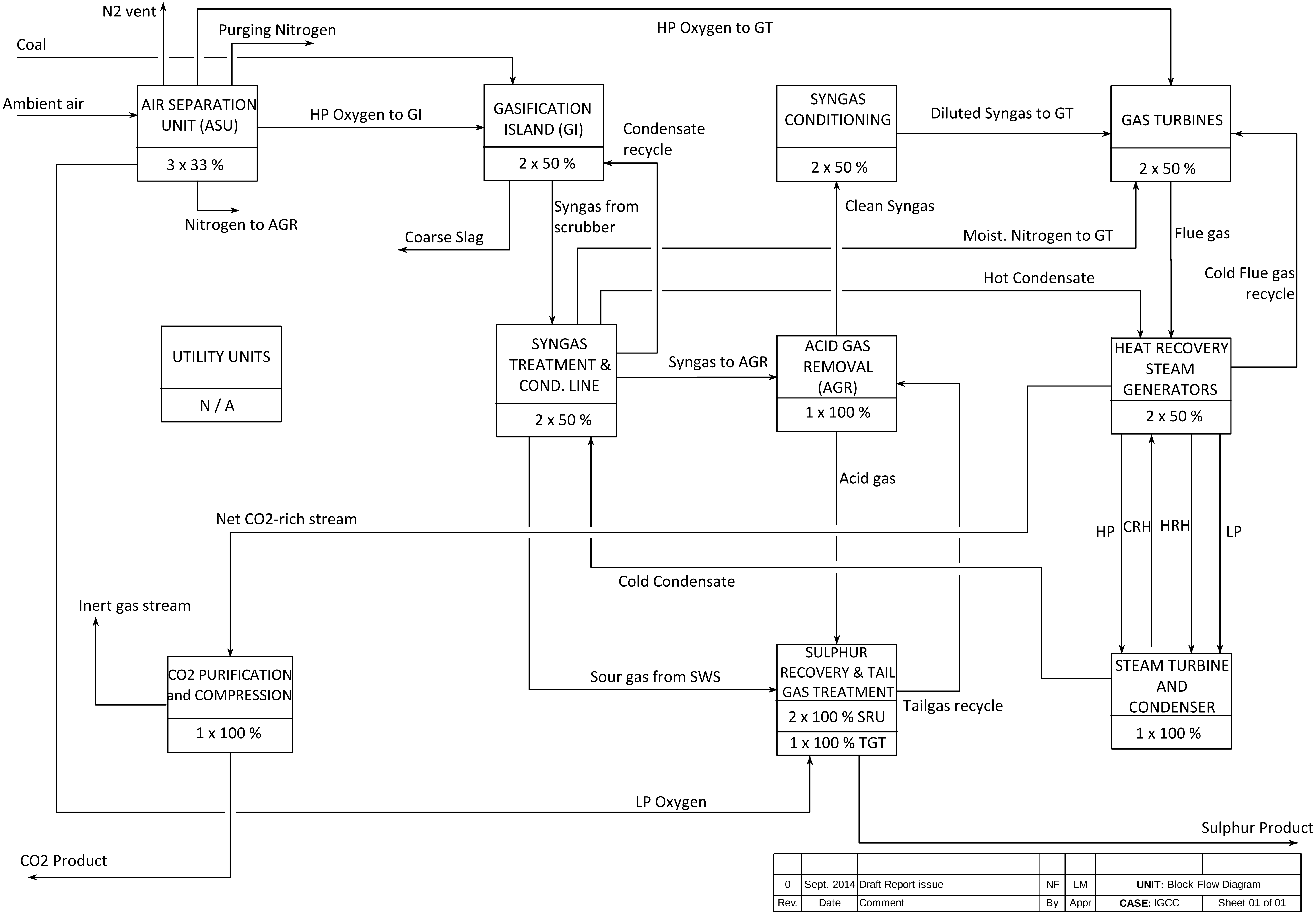
[2] X. Lu, *Flexible Integration of the sCO2AllamCycle with Coal Gasification for Low-Cost, Emission-Free Electricity Generation*, GCT 2014 Conference proceeding

1.1. Process unit arrangement

The arrangement of the main units is reported in the following Table 1. Reference is also made to the block flow diagram attached below.

Table 1. Oxy-turbine based IGCC – Unit arrangement

Unit	Description	Trains
900	Coal Handling & Storage	N/A
1000	Gasification	2 x 50%
	Coal Grinding & Slurry Preparation	
	Gasification (Radiant Syngas Cooler) and scrubber	
	Black Water Flash & Coarse Slag Handling	
	Grey Water & Fines Handling	
2100	Syngas Treatment and Conditioning Line	2 x 50%
	Sour Water Stripper (SWS)	1 x 100%
2200	Acid Gas Removal	1 x 100%
2300	Sulphur Recovery Unit	2 x 100%
	Tail Gas Treatment	1 x 100%
3000	Power Island	
	Gas Turbine	2 x 50%
	HRSG	2 x 50%
	Steam Turbine	1 x 100%
4000	CO_2 purification and compression	
	Raw gas compression	2 x 50%
	Auto-refrigerated inert removal section	1 x 100%
	CO_2 compressors	2 x 50%
5000	Air Separation Unit	3 x 33%
6000	Utility and Offsite	N/A



N2 vent
Purging Nitrogen
Coal
HP Oxygen to GT
Ambient air
AIR SEPARATION UNIT (ASU)
3 x 33 %
HP Oxygen to GI
GASIFICATION ISLAND (GI)
2 x 50 %
Condensate recycle
SYNGAS CONDITIONING
2 x 50 %
Diluted Syngas to GT
GAS TURBINES
2 x 50 %
Nitrogen to AGR
Coarse Slag
Syngas from scrubber
Clean Syngas
Moist. Nitrogen to GT
Flue gas
Hot Condensate
Cold Flue gas recycle
UTILITY UNITS
N / A
SYNGAS TREATMENT & COND. LINE
2 x 50 %
Syngas to AGR
ACID GAS REMOVAL (AGR)
1 x 100 %
HEAT RECOVERY STEAM GENERATORS
2 x 50 %
Acid gas
Net CO2-rich stream
HP
CRH
HRH
LP
Cold Condensate
Inert gas stream
CO2 PURIFICATION and COMPRESSION
1 x 100 %
Sour gas from SWS
SULPHUR RECOVERY & TAIL GAS TREATMENT
2 x 100 % SRU
1 x 100 % TGT
Tailgas recycle
STEAM TURBINE AND CONDENSER
1 x 100 %
LP Oxygen
Sulphur Product
CO2 Product
0
Sept. 2014
Draft Report issue
NF
LM
UNIT: Block Flow Diagram
Rev.
Date
Comment
By
Appr
CASE: IGCC
Sheet 01 of 01

2. Process description

2.1. Overview

The description reported in this section makes reference to the simplified Process Flow Diagrams (PFD) shown in section 3, while stream numbers refer to section 4, which provides heat and mass balance details for the numbered streams in the PFD.

2.2. Coal storage and handling

The scope of the feedstock receiving, handling and storage unit is to unload, convey, prepare, and store the coal delivered to the plant.

The coal is delivered from a port to the plant by train. The unloading is done by a wagon tipper that unloads the coal to the receiving equipment. Coal from each hopper is fed directly into a vibratory feeder and subsequently discharged onto a belt extractor. A conveyor and transfer tower system finally delivers the coal to the open stockyard (as-received coal).

The storage pile is designed to hold an inventory of 30 days of design consumption to allow the facility to hedge against delivery disruptions.

From the storage piles, the coal is discharged onto enclosed belt conveyors to two elevated feed hoppers, each sized for a capacity equivalent to two hours. Coal is discharged from the feed hoppers, at a controlled rate, and transported by belt feeders to parallel crushers, each sized for 100% of the full capacity. The crushers are designed to break down big lumps and deliver a coal with lump size not exceeding 35 mm. Coal from the crushers is then transferred by enclosed belt conveyors to the day silos close to the gasification island (as-fired coal).

Two magnetic plate separators for removal of tramp iron and two sampling systems are supplied for both the as-received coal and the as-fired coal. The recovered iron from the separators is delivered to a reclaim pile, while data from the analyses are used to support the reliable and efficient operation of the plant.

Enclosed belt conveyors, storage hoppers and silos, flow control feeders and other equipment handling coal are potential sources of air pollution, due to dispersion of fine powder. To control the plant environment all these items of equipment are connected to bag filters and exhaust fans that permit the capture of any coal powder generated in the coal handling area.

2.3. Unit 1000 – Gasification Island

The Gasification Island based on GE gasification includes the following units, briefly described in the following sections making reference to the block flow diagram in Figure 1.

- Coal Grinding & Slurry Preparation;

- Gasification based on radiant syngas cooler (RSC) technology & Scrubbing;
- Black Water Flash & Coarse Slag Handling;
- Grey Water & Fines Handling.

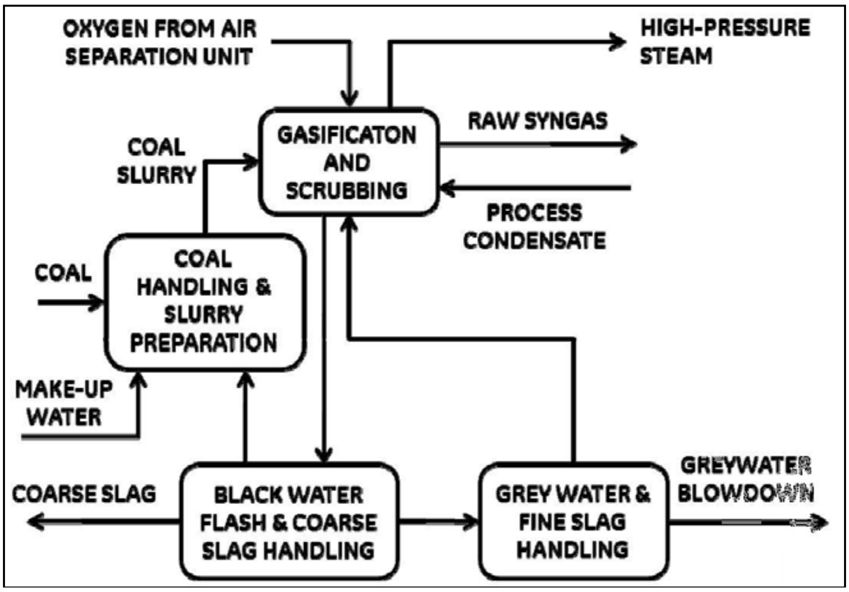


Figure 1. GE RSC - Preliminary Block Flow Diagram

Process data for the evaluation of the gasification island performance are as provided by GE Energy for the IEAGHG report 2014/3 '*CO_2 capture at coal based power and hydrogen plant*', developed by Foster Wheeler in 2013-2014.

2.3.1. Coal Grinding & Slurry Preparation

The coal grinding and slurry preparation system is dedicated to the preparation of the coal slurry feed for the gasifier.

Solid feedstock from offsite is continuously conveyed to a weigh feeder system that regulates the solid feed rate to the grinding mill. Fine slag may also be recycled into the grinding mill, which provides a means to prepare the solid as slurry feed for the gasifier. Grey water and/or fresh water are then used to slurry the grinding mill feed.

A fluxant system may be required based upon feedstock properties or if desired for future feedstock flexibility.

The slurry is transferred to a gasifier train by the slurry charge pump which provides a controlled flow of slurry to the gasifier feed injector.

2.3.2. *Gasification & Scrubbing*

A Gasification and Scrubbing train includes a Radiant-only Gasifier with high-pressure steam production, followed by a direct water quench cooling system.

The radiant-only gasifier is a refractory lined vessel capable of withstanding high temperatures and pressures. The coal slurry from the slurry run tank and oxygen from the air separation unit are fed and mixed through a gasifier feed injector and react at very high temperatures (approximately 1400°C). The pursuing partial oxidation reaction generates slag and syngas with a high hydrogen and carbon monoxide content, lesser content of water vapour, carbon dioxide, hydrogen sulphide, methane, nitrogen, and traces of carbonyl sulphide (COS) and ammonia. The ash content of the coal feed melts in the gasifier and transforms into slag.

The tip of the Feed Injector is protected from the high temperatures in the gasifier by a water jacket and cooling coils through which cooling water is continuously circulated.

From the gasifier vessel, the hot syngas and slag from the reaction chamber flow down into the Radiant Syngas Cooler (RSC) chamber. The RSC is a high-pressure steam generator equipped with a circulating boiler feed water wall to protect the vessel shell. Heat is transferred primarily by radiation from the hot syngas to the circulating BFW. The high-pressure steam flows to a liquid disengagement system and then to the high-pressure steam header for Power Generation and/or export to other applications. In the RSC chamber, the raw syngas from the radiant section is first cooled by direct contact with water, and then sent to the Syngas Scrubber for cooling, condensation of water vapour, and removal of particulates by scrubbing with water.

The syngas from the overhead of the Syngas Scrubber is routed to the Low Temperature Gas Cooling section (out of the GE’s scope). The bottom of the RSC vessel receives a portion of the syngas scrubber bottoms for cooling of the raw syngas and solidification of the molten slag. The water also wets the slag solids and assists its removal to the Lock Hopper.

2.3.3. *Black Water & Coarse Slag Handling*

The purpose of the black water flash system is to recover heat from the black water and to remove dissolved syngas, possibly requiring a deaerator (or sour water stripper) to provide deaerated return water to the syngas scrubber.

Black water from the RSC chamber and from the syngas scrubber is letdown and partially flashed through a series of flash stages the last of which is a vacuum flash stage. The flash stages serve to remove dissolved gases from the black water and to lower the black water temperature. The removed dissolved gases are routed to either a sour water stripper or offsite for treatment. Following the flashes, the black water is pumped to the gravity settler, which is part of the grey water handling system.

Black water from the vacuum flash vessel flows to the gravity settler where the solids are concentrated. A small amount of flocculent may be added upstream of the gravity settler to improve the settling efficiency. The gravity settler contains a slow moving rake that keeps the concentrated fine slag solids moving towards the bottom outlet. The concentrated gravity settler bottoms is either recycled to the grinding mill or pumped to filter feed tank in the fines filtration system.

The overflow water from the gravity settler (grey water) flows to the grey water tank. A slag crusher at the bottom of the radiant syngas cooler crushes the coarse slag which then gravity flows into the lockhopper, where an automated batch process is used for the collection of the solids. Once a solids collection cycle is complete, the lockhopper is isolated from the radiant gasifier, depressurized, flushed into a slag sump, re-pressurized, and opened to the radiant syngas cooler.

In the slag sump, slag settles onto a submerged slag drag conveyor, which separates the slag from the water. The coarse slag is dumped into trucks for removal offsite while the water removed from the slag is pumped to the black water flash system.

2.3.4. *Grey Water & Fines Handling*

The purpose of the grey water handling system is to concentrate solids in black water and to provide surge capacity for the grey water.

Black water from the vacuum flash vessel flows to the gravity settler where solids are concentrated. A small amount of flocculant may be added upstream of the gravity settler to improve settling efficiency.

The gravity settler contains a slow moving rake that keeps the concentrated fine slag solids moving towards the bottom outlet. The concentrated gravity settler bottoms is either sent to the filter (to be discharged as soot) and/or recycled to the grinding mill (to reduce soot disposal cost). The overflow water from the gravity settler (grey water) flows to the grey water tank, essentially free of particulates.

A low pressure grey water pump returns part of grey water to the lockhopper flash drum and the remaining water is blown down as grey water blowdown.

The high pressure grey water pump re-circulates grey water to the syngas scrubber through the deaerator.

2.4. Unit 2100 - Syngas treatment and conditioning line

Saturated raw syngas from the gasification scrubber, at approximately 64 barg, is cooled in the LP Steam Generator and in the VLP Steam Generator, to thermally recover heat and increase the overall power generation.

Raw syngas from the steam generators is reheated in the hydrolysis effluent exchanger and in the MP steam heater, before entering the hydrolysis reactor that converts COS to H_2S (section 2.4.1).

The reactor effluent is further cooled in the feed-effluent exchanger. Downstream the feed-effluent exchanger, the raw syngas is cooled in a second series of heat exchangers:

- LP Steam Generator,
- Condensate Pre-heater #1 and #2.

Final cooling of the syngas is made by cooling water. It is then passes through a sulphur-impregnated activated carbon bed to remove approximately 95% of the mercury (section 2.4.2). Cool, mercury-depleted syngas is then directed to the AGR.

During the cooling of the syngas, the process condensate is separated and collected in the process condensate accumulator. Before being sent to the accumulator, the condensate from the last syngas separator, upstream the AGR, plus a portion of the condensate from the upstream separator, is sent to the Sour Water Stripper (section 2.4.3) in order to avoid accumulation of ammonia and H_2S and other dissolved gases in the water recycle to the gasification section. The condensate from the accumulator is sent to the gasification scrubber for syngas saturation.

2.4.1. *COS hydrolysis*

The following simplified process description makes reference to the process flow diagram shown in Figure 2.

Approximately 95% of the sulphur in the syngas from the gasification island is in the form of hydrogen sulphide (H_2S), while the remaining 5% is carbonyl sulphide (COS). The Selexol solvent of the AGR removes some of the COS, though the solubility is much less than that of the H_2S. To remove the COS up to a level compatible with the sulphur emissions of the plant, the solvent circulation rate would be nearly doubled, thus increasing the capital and operating costs of the plant. It is therefore economically more attractive to perform the COS hydrolysis upstream of the AGR unit.

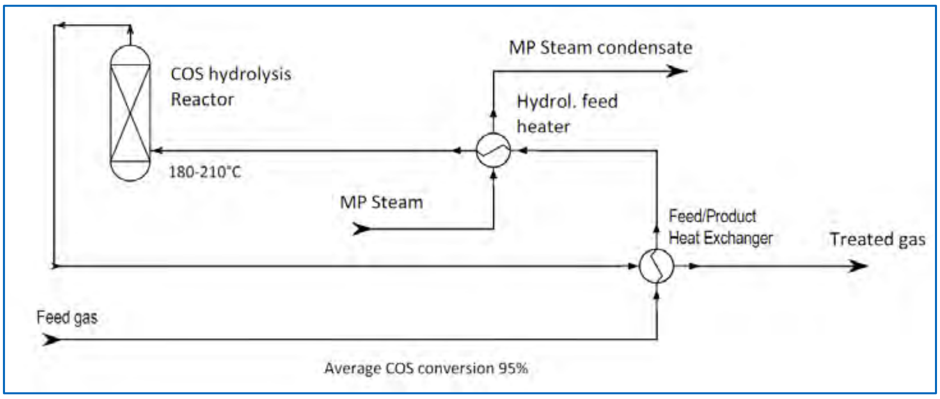


Figure 2. COS Hydrolysis: process flow diagram.

The COS hydrolysis reaction shown below is the reaction of the carbonyl sulphide with steam to produce carbon dioxide (CO_2) and H_2S:

$COS + H_2O \rightarrow H_2S + CO_2$

HCN is also removed from the syngas with the following reaction:

$HCN + H_2O \rightarrow NH_3 + CO$

The hydrolysis reactor optimum operating temperature is around 190°C. A feed/effluent exchanger and a final steam pre-heater with MP steam are included in the unit prior to the COS reactor, to ensure that the syngas is above the dew point since water condensation could damage the catalyst. The system is designed to allow 95% COS average conversion by controlling the temperature at the inlet of the reactor. The expected catalyst life is approximately 5 years.

The heat in the gas leaving the COS hydrolysis reactor is typically used to provide part of the pre-heat necessary for the process.

2.4.2. Mercury removal

The IGCC Complex includes a mercury removal system employed to eliminate mercury from the syngas before combustion in the gas turbine. This system uses sulphur-impregnated activated carbon beds capable of removing almost all the mercury in the syngas stream, thus meeting the environmental requirements.

The mercury removal package is located immediately upstream of the Acid Gas Removal unit, allowing operation of the system in its optimum conditions and enhancing the downstream AGR system performance and solvent life due to mercury and other contaminants removal.

Before entering the package, fuel gas is cooled to a temperature of about 34°C and the resulting process condensate is removed. Fuel gas is passed through one bed (one for each train) of sulphur-impregnated activated carbon.

2.4.3. Sour Water Stripper

The Sour Water Stripper (SWS) unit treats part of the contaminated condensate from the syngas treatment section and the blow-down from the AGR and the SRU, in order to avoid accumulation of ammonia and H_2S and other dissolved gases (e.g. CO_2, CO) in the water recycled to the gasification section. These dissolved gases, in particular the bulk of CO_2, H_2S and NH_3 contained in the sour water are removed by means of LP stripping steam supplied to the re-boiler.

Around 15% of the condensate from the syngas cooling is sent to the sour water stripper. All condensate from the last syngas separator, upstream the AGR, plus a portion of the condensate from the upstream separator, is sent to the stripper as solubility of NH_3 and H_2S increases at low temperature.

Before entering the stripper, sour condensate is heated against treated column bottoms, in order to enhance removal of gases from water. The warm stream enters via a distributor at the top of the stripper column.

The vapour stream from the top of the stripper is sent to an overhead system where vapour is condensed and the sour gases are separated from the condensate in the gas/liquid separator. The condensed water is routed back to the column as reflux, above the rectifying bed. The sour gases are routed to the SRU. The bottom from the column is pumped to the condensate collection vessel within the syngas treatment section, before being used as heating medium to pre-heat the stripper feed.

2.5. Unit 2200 - Acid gas removal

Cool, particulate-free syngas at around 56 barg enters the absorber of the AGR unit, which uses a physical solvent, like Selexol or Rectisol, or a chemical solvent, mainly amine-based, as an acid gas removing solvent. This unit is supplied by specialized Licensors, the following section makes reference to the process description of a typical Selexol unit, as licensed by UOP (Honeywell company).

In the absorber, the physical solvent preferentially removes H_2S and COS, together with some CO_2 and other gases such as hydrogen. The treated gas exits the top of the H_2S absorber and is sent to the syngas treatment and conditioning line for final conditioning. The H_2S-rich solvent stream from the absorber is regenerated in the stripper column.

The physical solvent washing matches the process specifications with reference to the H_2S and COS concentration of the treated gas exiting the unit.

CO_2 slippage into the treated syngas is virtually 100% and even CO_2 derived from the other minor acid streams fed to the SRU is recovered. The tail gas from the SRU is recycled back to the AGR (refer to next section). The acid gas H_2S concentration is approximately 41% dry basis, suitable to feed the downstream oxygen blown Claus process.

2.5.1. *UOP's process for H_2S removal*

Feed gas enters the unit battery and flows to the H_2S Absorber. The gas flows upward where it contacts regenerated lean solvent entering at the top of the absorber.

The contact between the gas phase and liquid phase is enhanced as they pass through the column, where primarily H_2S, COS, and some CO_2 and other gases such as hydrogen, are transferred from the gas phase to the liquid phase.

The treated gas exits the top of the H_2S Absorber and is sent to the battery limit of the unit.

The rich solvent from the bottom of the H_2S Absorber is sent to Lean / Rich Exchanger where the temperature is heated against the lean solvent from the bottom of the H_2S Stripper..

The rich solvent and desorbed vapours from the Lean/Rich Exchanger are routed to the H_2S Concentrator, where compounds such as CO_2, H_2, and CO with lower solubilities (with respect to H_2S) in Selexol are transferred from the liquid phase to the gas phase.

The gas exiting the H_2S Concentrator, primarily composed of CO_2, N_2, CO, and some H_2S, is compressed and finally recycled to the inlet of the H_2S Absorber.

The partially regenerated Selexol solvent is sent to the H_2S Stripper where the primary solvent regeneration occurs. The remaining H_2S, CO_2, N_2 and other compounds are transferred from the liquid phase to the gas phase by contact with the steam generated in the H_2S Stripper Reboiler.

The vapour phase exiting the stripper flows upward and it is contacted with counter-current flowing reflux water to cool and partially condense the hot overhead vapour, as well as reduce solvent entrainment. The overhead stream passes through a de-entrainment device and exits the top of the column. The overhead gas then passes through the Reflux Condenser in order to condense and recover a portion of the overhead steam. The liquid and vapour phases are separated in the Reflux Drum.

The H_2S-rich acid gas exits the Selexol unit battery limits, and the liquid is returned to the trayed section of the H_2S Stripper, via the Reflux Pump.

The heat source for the Stripper Reboiler is saturated low pressure steam. The reboiler is fed with the solvent collected below the section in the H_2S Stripper. The resulting steam generated in the reboiler re-enters the bottom of the Solvent Stripper, stripping the acid gas from the downflowing solvent. The total stripping heat duty of the reboiler is determined by the sensible heat requirement of the solvent, the Stripper overhead steam requirement, the heat necessary to transfer H_2S, CO_2 and other dissolved gasses from the solvent, and the heat required to vaporize the water absorbed from the feed gas.

The lean solvent from the H_2S Stripper is then sent to the hot side of the Lean / Rich Exchanger. The temperature of the lean solvent is further reduced in the Lean Solvent Cooler, and then returned to the top of the H_2S Absorber.

2.6. Unit 2300 - Sulphur Recovery Unit and Tail Gas Treatment

The Sulphur Recovery Unit (SRU) processes the main acid gas from the Acid Gas Removal unit, together with minor acid streams like the acid off-gas from the black water flash within the gasification island and the sour gases from the SWS. The SRU consists of two Claus Units for each line, each sized for approximately 100% of the maximum sulphur production in order to assure a satisfactory service factor. Low-pressure oxygen is used as oxidant of the Claus reaction.

A typical diagram of the Claus process is shown in Figure 3. The Claus plant consists of a Claus furnace and two catalytic conversion stages. The high operating temperature of the Claus furnace (approx 1100-1200°C) allows the destruction of the residual ammonia in the gas fed to the unit. In the furnace, H_2S is catalytically oxidized to SO_2 which is further reacted with H_2S to form H_2O and elemental sulphur.

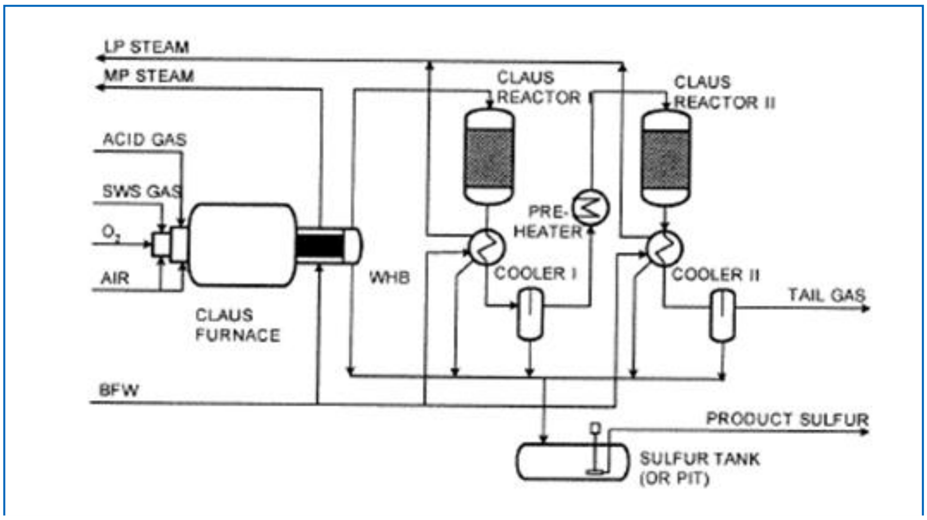


Figure 3. Claus unit diagram

The Claus process technology itself is limited by chemical equilibrium to approximately 98%, using multiple stages. To further enhance the sulphur recovery, it is therefore necessary to make provision for treating the tail gas, which contains the residual sulphur, in the Tail Gas Treatment Unit (TGT). This unit is designed as a single train, capable of processing 100% of the tail gas resulting from the possible SRU operating modes. The resulting overall sulphur recovery exceeds 98-99%.

The unit mainly includes a reduction reaction section, where the complete hydrogenation of SO_2, residual COS, CS_2, and elemental sulphur is achieved. The high hydrogen content in the shifted syngas results in a hydrogen content in the tail gas sufficient for complete hydrogenation of the tail gas stream itself. After quenching, tail gas is recycled back to the AGR at approximately 60 barg by means of a tail gas recycle compressor.

The overall sulphur production is approximately 62 tons per day.

2.7. Unit 3000 – Power Island

The unit is mainly composed of:

- Two F-class equivalent oxy-fired gas turbines.
- Two heat recovery steam generators (HRSG), generating steam at four levels of pressure, including a VLP integrated deaerator.
- Two recycle gas indirect contact cooling systems.
- One steam turbine, water-cooled and condensing type, common to the two parallel trains.

Technical information relevant to this unit is reported in chapter D, section 2.1.1, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

Main design parameters (e.g. combustion outlet temperature, cycle pressure level) and main characteristics of the stream at gas turbine boundary are summarised in the following Figure 4.

The syngas before the expander is heated using heat available from the flue gas compression in the CPU in order to enter the burners of the gas turbine at 117°C.

Oxygen is delivered from the ASU at the required pressure level and heated using heat available from the raw gas compressor in the CPU before entering the burners of the gas turbine at 200°C.

The pressure ratio of the SCOC-CC is higher than the one of an equivalent standard air blown commercial plant. Turbine inlet pressure of 43 bar has been selected to bring about the same temperature increase (385°C) across the compressor of the reference air-fired gas turbine. A pressure slightly higher than the ambient pressure (to avoid leakages into the CO_2 loop) has been selected so as to keep the design of the turbomachines closer to the current standards.

The gas turbine expander has 5 stages, so to have acceptable Mach number in the gas stream, peripheral velocity and blade height to mean diameter ratio of the last stage similar to those of the reference gas turbine, at a rotational speed of 3000 RPM.

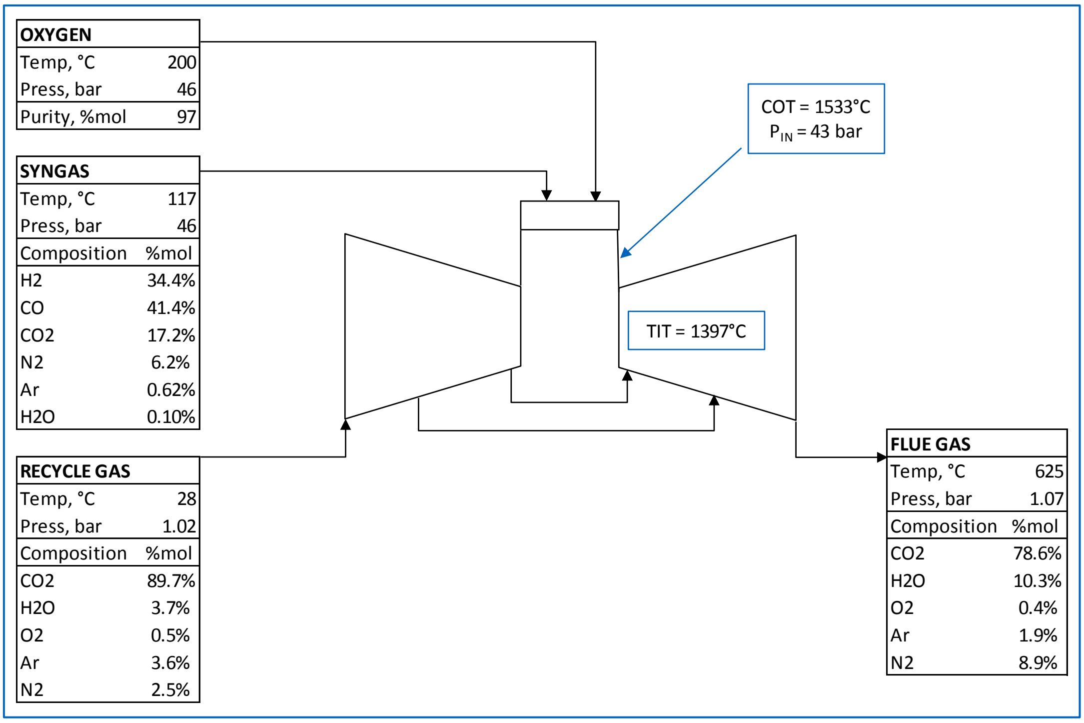


Figure 4. SCOC-CC syngas gas turbine

The gas turbine recycle flowrate is fixed by two design requirements to:

- control the combustion outlet temperature at 1533°C;
- provide the required cooling flow to the gas turbine blade in order to control the blade metal temperature as detailed below:

Turbine section	Maximum allowable temperature
1st rotor	860°C
1st stator	830°C
from 2nd rotor	830°C
from 2nd stator	800°C

The exhaust gases from the gas turbine enter the HRSG at 625°C. The HRSG recovers heat available from the exhaust gas producing steam at the following pressure levels for the steam turbine, including the steam generator with integral deaerator.

- HP steam: 134 bar, (SH: 565°C)
- MP steam: 44 bar, (RH: 565°C)
- LP steam: 9.0 bar
- VLP steam: 2.2 bar

The final exhaust gases are cooled down in a conventional contact cooler to the minimum temperature allowed by the cooling medium available. Most of the flue gas at 28°C from the top of the contact column (around 84%) is recycled back to the gas turbine compressors, while the remaining stream is sent to the downstream CO_2 purification and compression unit.

The combined cycle is thermally integrated with the other process units in order to maximize the net electrical efficiency of the plant. The main steam and water interfaces with the process units are given in Table 3.

2.8. Unit 5000 – Air Separation Unit

The ASU is based on the cryogenic distillation of atmospheric air at low pressure and it is designed to produce oxygen at 97%mol. O_2 purity and 50 bar. Oxygen pressure is set by the requirement of the gas turbine combustor. A dedicated compressor for each gasification train is foreseen to compress the oxygen up to the pressure required for the gasification.

The oxygen flowrate to the gas turbine is selected in order to lead the combustion reaction with an oxygen excess of 3% with respect to the stoichiometric, including the amount of oxygen in the recycle flowrate from the compressor.

Technical information relevant to this unit is reported in chapter D, section 2.4, while main process information of this case and the interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.9. Unit 4000 – CO_2 compression and purification

This unit is mainly composed of the following systems:

- Raw flue gas compression (1 - 34 bar);
- TSA unit;
- Auto-refrigerated inerts removal, including distillation column to meet the maximum oxygen content limit in the CO_2 product;
- The remaining part of the compression system up to 110 bar.

Technical information relevant to this system is reported in chapter D, section 2.3, while main process information of this case and interconnections with the other units are shown in the process flow diagram and in the heat and mass balance tables.

2.10. Unit 6000 - Utility Units

These units comprise all the systems necessary to allow the operation of the plant and the export of the produced power.

The main utility units include:

- Cooling Water system, based on two natural draft cooling towers, with the following main characteristics:

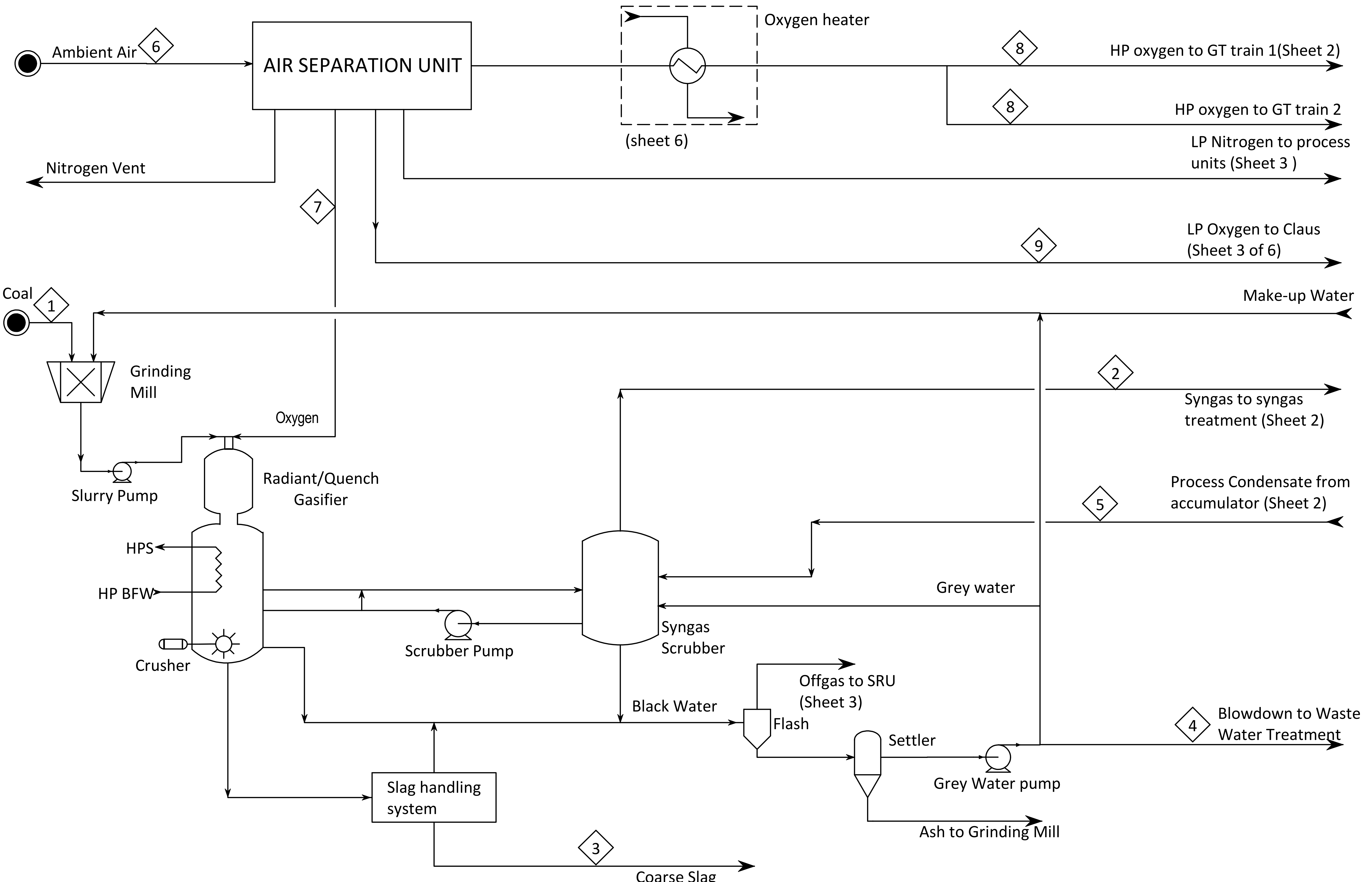
Basin diameter	100 m
Cooling tower height	210 m
Water inlet height	17 m

- Raw water system;
- Demineralised water plant;
- Waste Water Treatment
- Firefighting system;
- Instrument and Plant air.

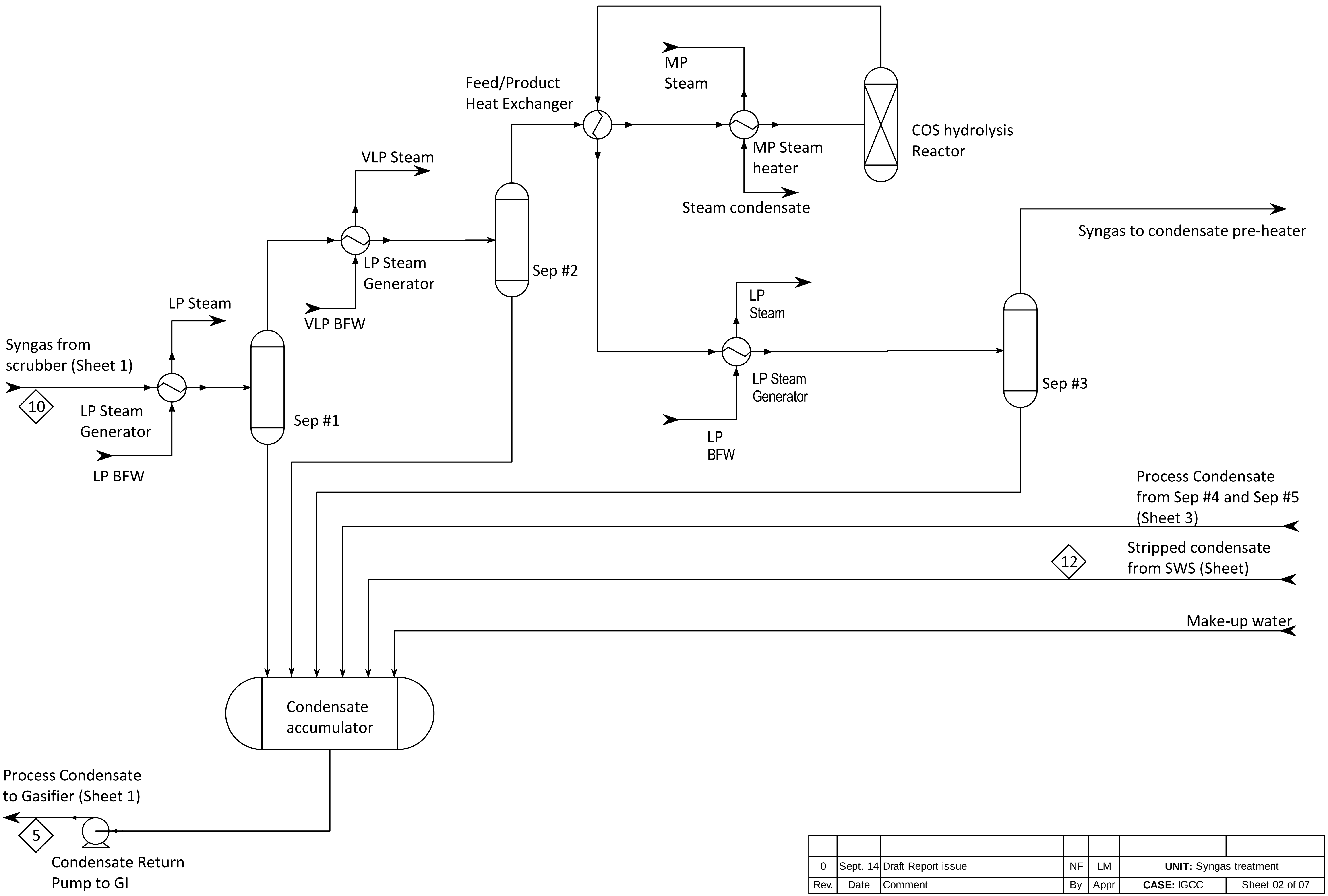
Process descriptions of the above systems are enclosed in chapter D, section 2.5.

3. Process Flow Diagrams

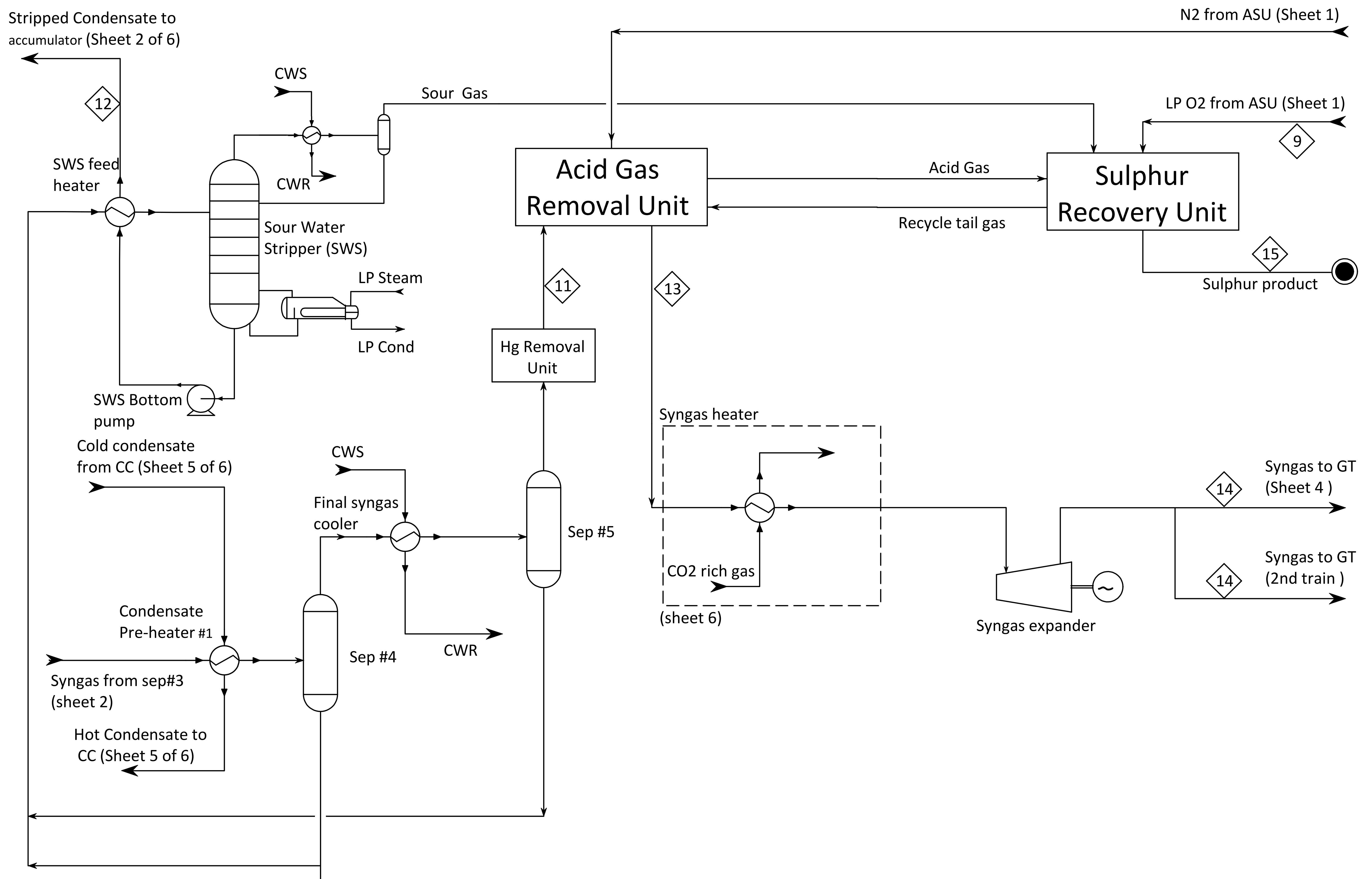
Simplified Process Flow Diagrams of this case are attached to this section. Stream numbers refer to the heat and material balance shown in the next section.



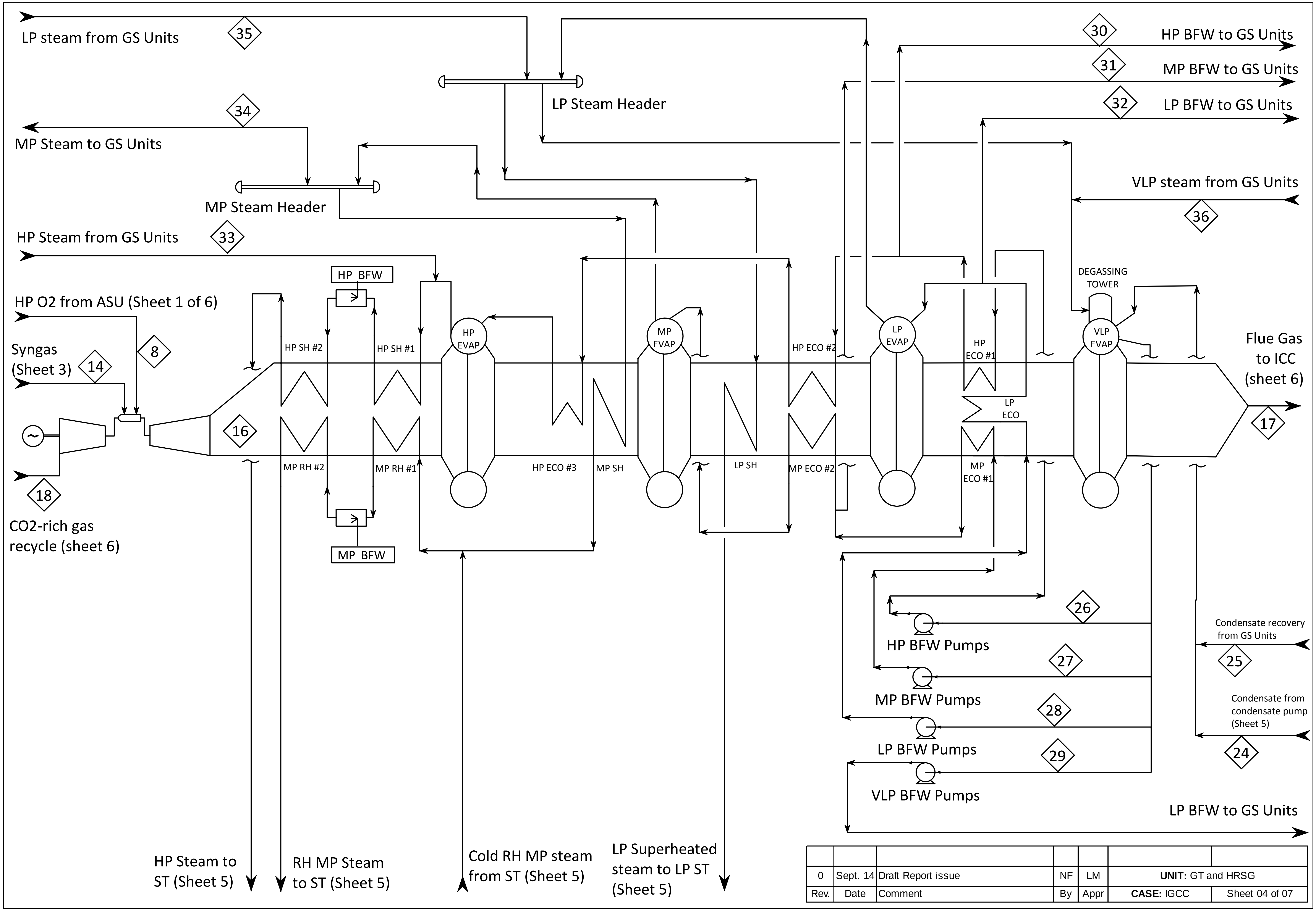
0	Sept. 14	Draft Report issue	NF	LM	**UNIT:** Gasification and ASU	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 01 of 07



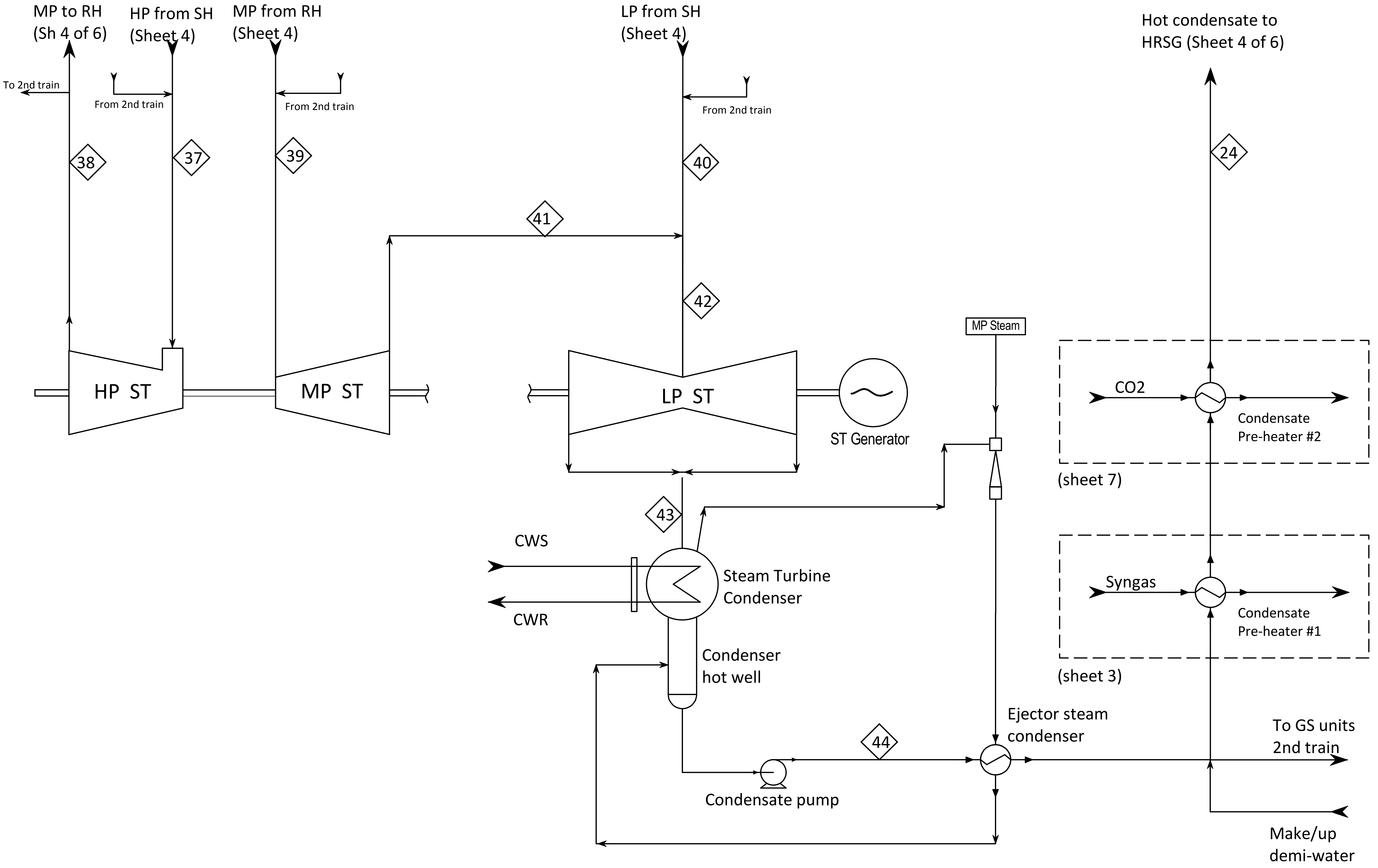
0	Sept. 14	Draft Report issue	NF	LM	**UNIT:** Syngas treatment	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 02 of 07



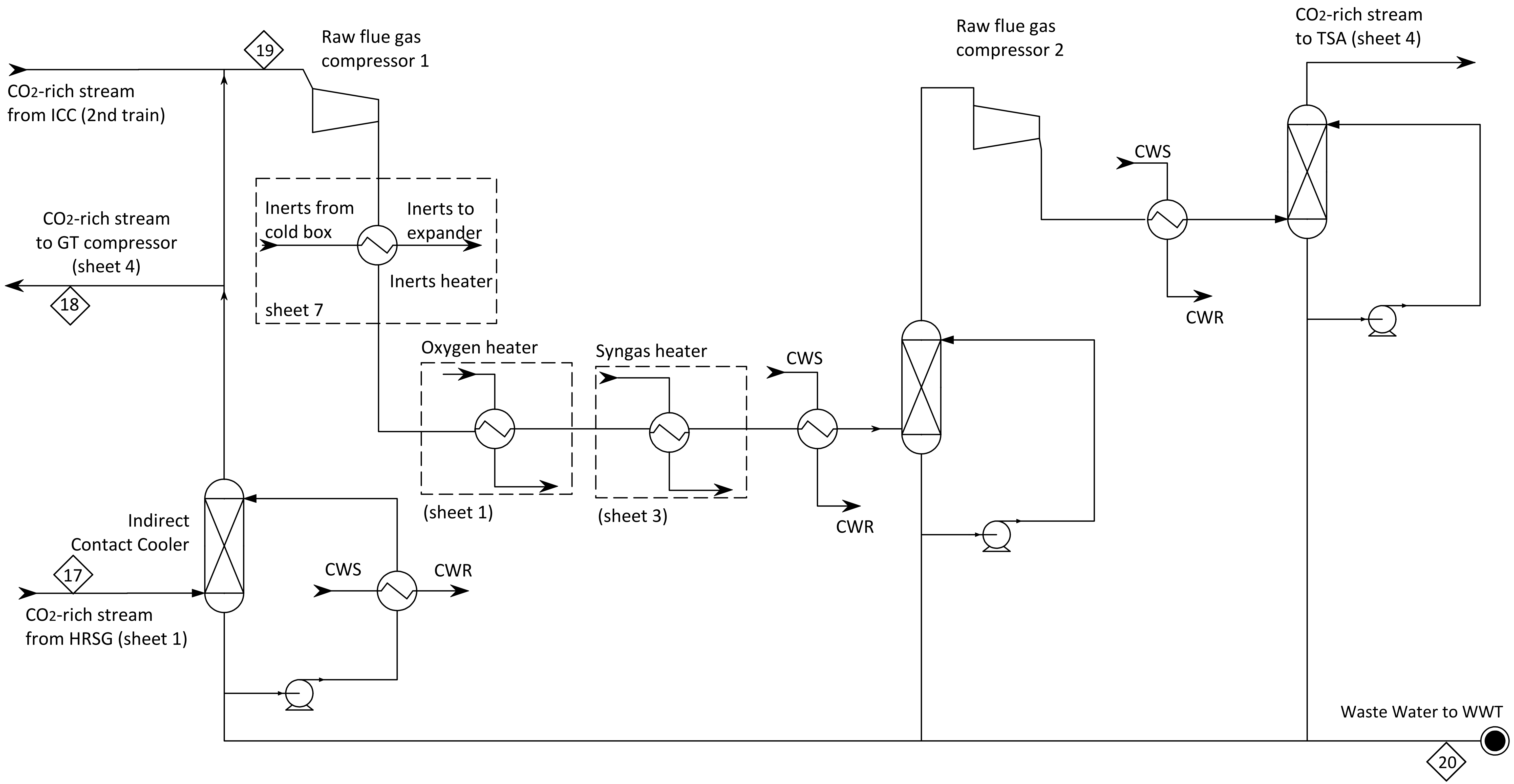
0	Sept. 14	Draft Report issue	NF	LM	**UNIT:** Syngas treat., AGR, SRU+TGT	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 03 of 07



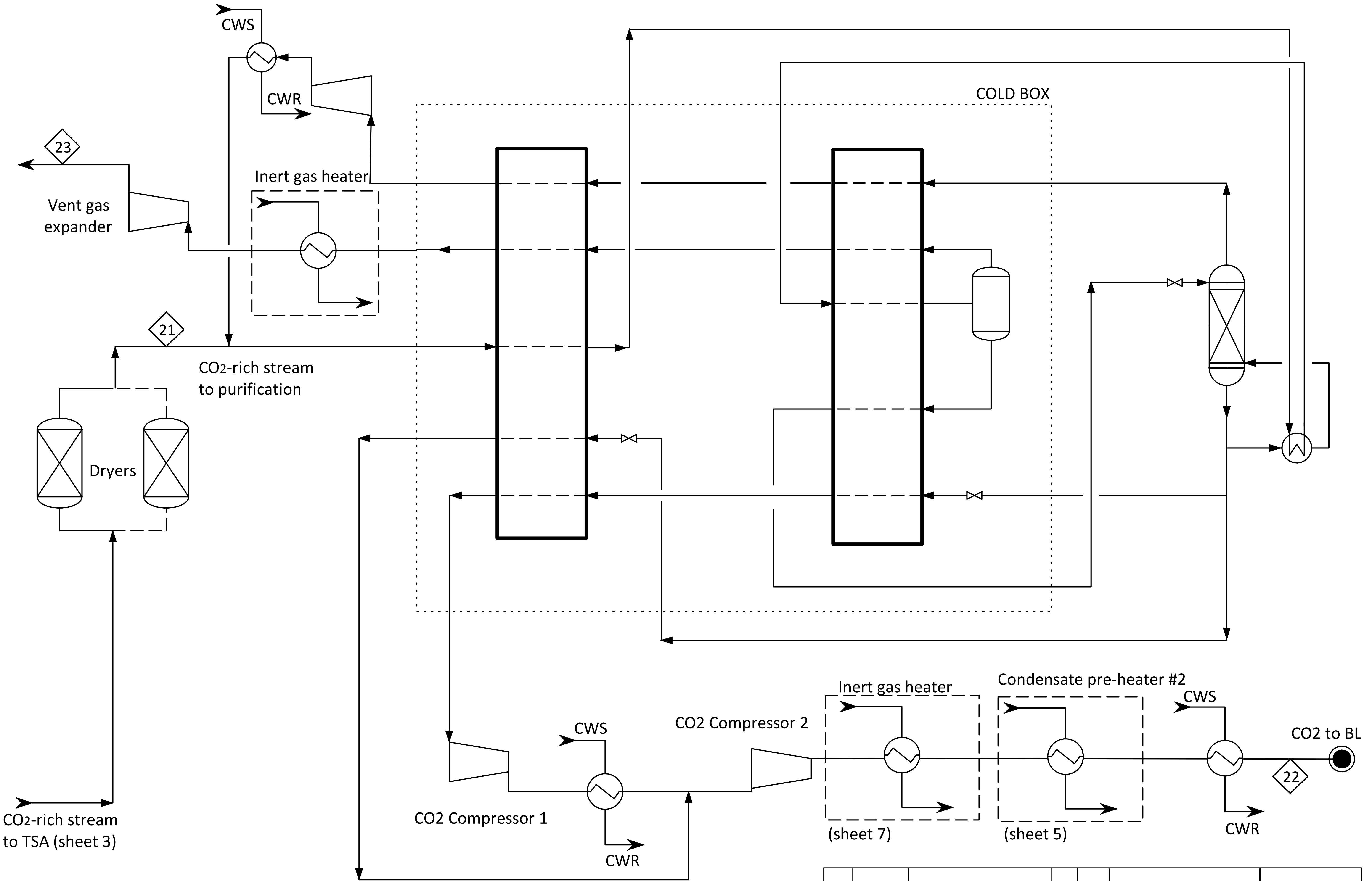
0	Sept. 14	Draft Report issue	NF	LM	**UNIT:** GT and HRSG	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 04 of 07



0	Sept. 14	Draft Report issue	NF	LM	**UNIT:** Steam Turbine and condenser	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 05 of 07



0	Sept. 2014	Draft report issue	NF	LM	**UNIT:** CO2 purification - compression section	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 06 of 07



0	Sept. 2014	Draft report issue	NF	LM	**UNIT:** CO2 purification - inerts removal section	
Rev.	Date	Comment	By	Appr	**CASE:** IGCC	Sheet 07 of 07

4. Heat and Material Balance

Heat & Material Balances reported make reference to the simplified Process Flow Diagrams shown in section 3.

	GE based IGCC with SCOC-CC - HEAT AND MATERIAL BALANCE		REVISION	0		
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	NF		
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM		
	PROJECT NO:	1-BD-0764 A	APPROVED	LM		
	LOCATION:	The Netherlands	DATE	September 2014		

HEAT AND MATERIAL BALANCE

STREAM	1	2	3	4	5	6	7	8	9	10
	Coal from BL	Syngas to syngas treatment	Slag from gasification	Effluent water from gasification	Process condensate to scrubber	Ambient air	HP oxygen to gasification	HP oxygen to GT*	LP oxygen to claus	Syngas to COS hydrolysis
Temperature (°C)	amb	N/D	80	amb	165	amb	60	200	15	190
Pressure (bar)	atm	64.6	atm	2.5	70	atm	75	46	6.5	62
TOTAL FLOW	Solid		Solid + water							
Mass flow (kg/h)	302,900	1,003,410	37,900	82,000	427,325	2,721,500	273,620	174,320	1,190	648,690
Molar flow (kmol/h)		50,280		4,550	23,720	94,300	8,520	5,430	37	30,630
LIQUID PHASE										
Mass flow (kg/h)				82,000	427,325					
GASEOUS PHASE										
Mass flow (kg/h)		1,003,410	37,900			2,721,500	273,620	174,320	1,190	648,690
Molar flow (kmol/h)		50,280				94,300	8,520	5,430	37	30,630
Molecular Weight (kg/kmol)		20.0				28.9	32.1	32.1	32.1	21.2
Composition (vol %)	%wt	dry basis	50% moisture							
H_2	C: 64.6%	35.77%		0.00%	0.00%	-	-	-	-	31.00%
CO	H: 4.38%	43.00%		0.00%	0.00%	0.00%	-	-	-	37.28%
CO_2	O: 7.02%	18.02%		0.00%	0.00%	0.04%	-	-	-	15.54%
H_2O	S: 0.86%	-		100.00%	100.00%	0.97%	-	-	-	13.41%
N_2	N: 1.41%	2.26%		0.00%	0.00%	77.32%	1.00%	1.00%	1.00%	1.95%
O_2	Cl: 0.03%	-		0.00%	0.00%	20.75%	97.00%	97.00%	97.00%	-
Ar	Moisture: 9.5%	0.64%		0.00%	0.00%	0.92%	2.00%	2.00%	2.00%	0.56%
H_2S + COS	Ash: 12.20%	0.31%		0.00%	0.00%	-	-	-	-	0.26%
Total		100.00%		100.00%	100.00%	100.00%	100.00%	100.00%	100.00%	100.00%

NOTE

1. Streams marked up with * correspond to the flowrate of a single train. The remaining figures are referred to the total flow of train

	GE based IGCC with SCOC-CC - HEAT AND MATERIAL BALANCE		REVISION	0		
FOSTER WHEELER	**CLIENT :**	IEAGHG	**PREP.**	NF		
	PROJECT NAME:	Oxy-turbine power plants	**CHECKED**	LM		
	PROJECT NO:	1-BD-0764 A	**APPROVED**	LM		
	LOCATION:	The Netherlands	**DATE**	September 2014		

HEAT AND MATERIAL BALANCE

STREAM	**11**	**12**	**13**	**14***	**15**	**16***	**17***	**18***	**19**	**20**
	Syngas to AGR	Stripped condensate from SWS	Syngas from AGR	Syngas to GT	Sulphur product	Flue gas to HRSG	Flue gas to ICC	Flue gas recycle	Flue gas to CPU	Waste water to WWT
Temperature (°C)	38	90	42	117	amb	625	133	28	28	25
Pressure (bar)	57	60	56	46.0	atm	1.07	1.04	1.03	1.02	2.5
TOTAL FLOW					Solid					
Mass flow (kg/h)	575,445	64,100	603,600	301,800	62 t/d	2,537,640	2,537,640	2,061,000	790,850	167,720
Molar flow (kmol/h)	26,565	3,560	27,580	13,790		63,775	63,775	49,850	19,160	9,310
LIQUID PHASE										
Mass flow (kg/h)		64,100								167720
GASEOUS PHASE										
Mass flow (kg/h)	575,445		603,600	301,800		2,537,640	2,537,640	2,061,000	790,850	
Molar flow (kmol/h)	26,565		27,580	13,790		63,775	63,775	49,850	19,160	
Molecular Weight (kg/kmol)	21.7		21.9	21.9		39.8	39.8	41.3	41.3	
Composition (vol %)					99.9%wt S					
H_2	35.74%	0.00%	34.43%	34.43%		-	-	-	-	-
CO	42.97%	0.00%	41.39%	41.39%		0.00%	0.00%	0.00%	0.00%	-
CO_2	17.91%	0.00%	17.25%	17.25%		78.58%	78.58%	84.42%	84.42%	0.00%
H_2O	0.18%	100.00%	0.10%	0.10%		10.33%	10.33%	3.67%	3.67%	100.00%
N_2	2.25%	0.00%	6.21%	6.21%		8.85%	8.85%	9.51%	9.51%	0.00%
O_2	-	0.00%	-	-		0.35%	0.35%	0.38%	0.38%	0.00%
Ar	0.64%	0.00%	0.62%	0.62%		1.89%	1.89%	2.03%	2.03%	0.00%
H_2S + COS	0.30%	0.00%	0.00%	0.00%		-	-	-	-	-
Total	100.00%	100.00%	100.00%	100.00%		100.00%	100.00%	100.00%	100.00%	100.00%

NOTE

1. Streams marked up with * correspond to the flowrate of a single train. The remaining figures are referred to the total flow of train

	GE based IGCC with SCOC-CC - HEAT AND MATERIAL BALANCE		REVISION	0		
FOSTER WHEELER	CLIENT :	IEAGHG	PREP.	NF		
	PROJECT NAME:	Oxy-turbine power plants	CHECKED	LM		
	PROJECT NO:	1-BD-0764 A	APPROVED	LM		
	LOCATION:	The Netherlands	DATE	September 2014		

HEAT AND MATERIAL BALANCE

STREAM	21	22	23							
	CO2-rich stream to purification	CO2 to BL	Inert gas stream							
Temperature (°C)	28	30	80							
Pressure (bar)	33	110	1.1							
TOTAL FLOW										
Mass flow (kg/h)	777,815	642,365	135,450							
Molar flow (kmol/h)	18,445	14,595	3,850							
LIQUID PHASE										
Mass flow (kg/h)		642,365								
GASEOUS PHASE										
Mass flow (kg/h)	777,815		135,450							
Molar flow (kmol/h)	18,445		3,850							
Molecular Weight (kg/kmol)	42.2		35.2							
Composition (vol %)										
H_2	-	-	-							
CO	-	-	-							
CO_2	87.63%	99.96%	40.88%							
H_2O	-	-	-							
N_2	9.87%	0.00%	47.27%							
O_2	2.10%	0.01%	10.04%							
Ar	0.40%	0.03%	1.81%							
H_2S + COS	-	-	-							
Total	100.00%	100.00%	100.00%							

NOTE

1. Streams marked up with * correspond to the flowrate of a single train. The remaining figures are referred to the total flow of train

	GE based IGCC with SCOC-CC - HEAT AND MATERIAL BALANCE		Revision	0	
	CLIENT:	IEAGHG	**Prepared**	NF	
	PROJECT NAME:	Oxy-turbine power plants	**Checked**	LM	
	PROJECT NO:	1-BD-0764 A	**Approved**	LM	
	LOCATION:	The Netherlands	**Date**	September 2014	

HEAT AND MATERIAL BALANCE

Stream	Description	Flowrate t/h	Temperature °C	Pressure bar a	Entalphy kJ/kg
24*	Condensate from condensate pump	613.4	29	10	122.5
25*	Condensate recovery from process unit	38.2	94	10	394.6
26*	BFW to HP BFW Pump	391.7	123	2.2	517.6
27*	BFW to MP BFW Pump	65.4	123	2.2	517.6
28*	BFW to LP BFW Pump	199.4	123	2.2	517.6
29*	BFW to VLP BFW Pump	31.3	123	2.2	517.6
30*	HP BFW to process units	193.8	123	165	529.0
31*	MP BFW to process units	2.6	123	57	521.4
32*	LP BFW to process units	156.5	123	15	518.5
33*	HP steam from process unit	192.7	333	134	2653
34*	MP steam to process unit	2.5	256	44	2799
35*	LP steam from process unit	124.8	175	9.0	2773
36*	VLP steam from process unit	26.8	145	4.2	2740
37	HP steam to steam turbine	779.3	563	129	3506
38	Exhaust from MP Steam Turbine	779.3	380	39	3169
39	Hot reheat to steam turbine	901.7	563	35	3595
40	LP steam from HRSG	316.6	235	6.0	2926
41	Exhaust from MP Steam Turbine	901.7	310	6.0	3083
42	LP steam to LP Steam Turbine	1218.3	290	6.0	3041
43	Exhaust from LP steam turbine	1218.3	29	0.04	2304.2
44	Condensate from condenser	1226.7	29	0.04	121.6

NOTE
1. Streams marked up with * correspond to the flowrate of a single train. The remaining figures are referred to the total flow of train

5. Utility and chemicals consumption

Main utility consumption of the process and utility units is reported in the following tables. More specifically:

- Steam / BFW / condensate interface summary is reported in Table 2;
- Water consumption summary, reported in Table 3;
- Electrical consumption summary, shown in Table 4.

Table 2. Oxy-turbine based IGCC – Steam/BFW/condensate interface summary

CLIENT: IEAGHG	REVISION	Rev. Draft	Rev. 0	Rev. 1	Rev. 2
PROJECT: Oxy-turbine power plant	DATE	Sep-14			
LOCATION: The Netherlands	ISSUED BY	NF			
FWI Nº: 1-BD-0764A	CHECKED BY	LM			
	APPROVED BY	LM			

Oxy-turbine based IGCC

STEAM, BOILER FEED WATER, CONDENSATE BALANCE

UNIT	DESCRIPTION UNIT	HP Steam barg 133 [t/h]	MP Steam barg 43 [t/h]	LP Steam barg 8.0 [t/h]	VLP Steam barg 3.20 [t/h]	HP BFW [t/h]	MP BFW [t/h]	LP BFW [t/h]	VLP BFW [t/h]	condensate recovery [t/h]	Losses [t/h]
	GASIFICATION AND PROCESS UNITS										
1000	Gasification island	-385.4				387.6					2.2
2200	Syngas treatment and Conditioning Line		7.1	-309.9	-51.7			313.0	60.8	-15.6	3.7
2300	Acid Gas Removal (AGR)			45.3						-45.3	0.0
2400-2500	Sulphur Recovery Unit (SRU) and Tail Gas Treatment (TGT)		-4.6		-1.8		5.2		1.8	-0.5	0.1
4000	CO2 purification unit										
5000	Air Separation Unit (ASU)										
	POWER ISLAND										
3000	Combined cycle	385.4	-2.5	249.7	53.5	-387.6	-5.2	-313.0	-62.6	76.5	-6.0
	UTILITY and OFFSITE UNITS										
6000	Utility and offsite units			15.0						-15.0	0.0
	BALANCE	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Note: (1) Negative figures represent generation

Table 3. Oxy-turbine based IGCC – Water consumption summary

CLIENT:	IEAGHG	REVISION	0
PROJECT NAME:	Oxy-turbine power plant	DATE	Sep-14
PROJECT No. :	1-BD-0764A	MADE BY	NF
LOCATION :	The Netherlands	APPROVED BY	LM

Oxy-turbine based IGCC

WATER CONSUMPTION

UNIT	DESCRIPTION UNIT	Raw Water [t/h]	Demi Water [t/h]	Cooling Water [DT = 11°C] [t/h]
	PROCESS UNIT			
1000	Gasification island	125		3,360
2100	Syngas treatment and Conditioning Line			1,000
2200	Acid gas removal (AGR)		0	2,250
2300-2400	Sulphur Recovery Unit (SRU) and Tail Gas Treatment (TGT)			250
3000	**OXY-TURBINE CYCLE**			
	Condenser			58,290
	Turbine and generator Auxiliaries		10	4,000
	Indirect contact cooler			19,290
5000	**AIR SEPARATION UNIT**			
	MAC intercoolers			12,620
	BAC intercoolers			1,960
4000	**CO_2 PURIFICATION UNIT**			
	CO2 purification unit			6,760
6000	**UTILITY and OFFSITE UNITS**			
	Cooling Water System	1,850		
	Demineralized water unit	15	-10	
	Waste Water Treatment and Condensate Recovery	-240		
	Balance of plant			
	BALANCE	1,750	0	102,920

Note: (1) Minus prior to figure means figure is generated

Table 4. Oxy-turbine based IGCC – Electrical consumption summary

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Sep-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Oxy-turbine based IGCC

ELECTRICAL CONSUMPTION

UNIT	DESCRIPTION UNIT	Electric Power [kW]
	CO_2 PURIFICATION UNIT	
1000	Gasification island	**7,630**
2100	Syngas treatment and Conditioning Line	**450**
2200	Acid gas removal (AGR)	**2,920**
2300-2400	Sulphur Recovery Unit (SRU) and Tail Gas Treatment (TGT)	**910**
3000	**OXY-TURBINE CYCLE**	
	BFW and condensate pumps	**5,440**
	Turbine Auxiliaries + generator losses	**6,440**
5000	**AIR SEPARATION UNIT**	
	Main Air Compressors	**174,600**
	Booster air compressor and miscellanea	**25,980**
	Oxygen compressor to GI	**3,200**
4000	**CO_2 PURIFICATION UNIT**	
	Flue gas compression section	**70,400**
	Autorefrigerated inerts removal unit compression consumption	**31,900**
	Autorefrigerated inerts removal unit expander production	**-7,480**
6000	**UTILITY and OFFSITE UNITS**	
	Cooling Water System	**12,880**
	Balance of plant	**1,460**
	BALANCE	**336,730**

6. Overall performance

The following table shows the overall performances of the IGCC plant, while table below shows the CO_2 balance and removal efficiency.

CLIENT:	IEAGHG	**REVISION**	0
PROJECT NAME:	Oxy-turbine power plant	**DATE**	Sep-14
PROJECT No. :	1-BD-0764A	**MADE BY**	NF
LOCATION :	The Netherlands	**APPROVED BY**	LM

Oxy-turbine based IGCC

OVERALL PERFORMANCES		
Coal flow rate (as received)	t/h	302.9
Coal LHV (as received)	kJ/kg	25870
Coal HHV (as received)	kJ/kg	27060
THERMAL ENERGY OF FEEDSTOCK (based on LHV) (A)	**MWth**	**2177**
THERMAL ENERGY OF FEEDSTOCK (based on HHV) (A')	**MWth**	**2277**
Thermal Power of Raw Syngas exit Scrubber (D)	MWth (LHV)	1549
Thermal power of syngas to AGR	MWth (LHV)	1548
Thermal Power of Clean Syngas to Gas Turbines (E)	MWth (LHV)	1536
Syngas treatment efficiency (E/D x 100)	% (LHV)	99.2
Gas turbine power output (@ gen terminals)	MWe	633.0
Steam turbine power output (@ gen terminals)	MWe	441.9
Syngas expander	MWe	3.8
GROSS ELECTRIC POWER OUTPUT (C)	**MWe**	**1078.7**
Process unit, including gasification island	MWe	11.9
Oxy-turbine cycle	MWe	11.9
Air separation unit	MWe	203.8
CO2 purification and compression unit	MWe	94.8
Utility & Offsite Units	MWe	14.3
ELECTRIC POWER CONSUMPTION	**MWe**	**336.7**
NET ELECTRIC POWER OUTPUT	**MWe**	**742.0**
(Step Up transformer efficiency = 0.997%) (B)	MWe	739.7
Gross electrical efficiency (C/A x 100) (based on LHV)	**%**	**49.6%**
Net electrical efficiency (B/A x 100) (based on LHV)	**%**	**34.0%**
Gross electrical efficiency (C/A' x 100) (based on HHV)	**%**	**47.4%**
Net electrical efficiency (B/A' x 100) (based on HHV)	**%**	**32.6%**
Fuel Consumption per net power production	**MWth/MWe**	**2.94**
CO_2 emission per net power production	**kg/MWh**	**93.7**

CO_2 removal efficiency	Equivalent flow of CO_2 kmol/h
INPUT	
Fuel Mix (Carbon AR)	16292
TOTAL (A)	16292
OUTPUT	
Slag + Waste water (B)	127
Total to storage (C)	14591
Emission	1574
TOTAL	16292
Overall Carbon Capture, % ((B+C)/A)	90.3

7. Environmental impact

The oxy-combustion IGCC plant shall be designed in order to reach high electrical generation efficiency, while minimizing impact to the environment. Main gaseous emissions, liquid effluents, and solid wastes from the plant are summarized in the following sections.

7.1. Gaseous emissions

During normal operation at full load, main continuous emissions are the inerts vent stream from the CO_2 purification unit. Table 5 summarizes the expected flow rate and composition of the inerts vent.

Minor gaseous emissions are created by process vents and fugitive emissions. Some of the vent points emit continuously; others during process upsets or emergency conditions only. All vent streams containing, potentially, undesirable gaseous components are sent to a flare system. Venting via the flare will be minimal during normal operation, but will be significant during emergencies, process upsets, start up and shutdown. Fugitive emissions are related to the milling, storage and handling of solids (mainly consisting of air containing coal particulate) and to seal losses in the process units.

Table 5. Oxy-turbine based IGCC – Plant emission during normal operation

Inert gas from CPU	
Emission type	Continuous
Conditions	
Wet gas flowrate, kg/h	135,450
Flow, Nm^3/h	86,300
Composition	(%mol)
Ar	1.81
N_2	47.27
O_2	10.04
CO_2	40.88
H_2O	-
NOx	< 1 ppmv
SOx	< 1 pmmv

7.2. Liquid effluent

The plant main liquid effluent is cooling tower continuous blow-down, necessary to prevent precipitation of dissolved solids. The process units blow-downs (e.g. from flue gas final contact cooler and CO_2 purification unit) are treated to recover water, with a consequent effluent from the Waste Water Treatment.

Cooling Tower blowdown

Flowrate : 440 m^3/h

Waste Water Treatment effluent

Flowrate : 20 m^3/h

7.3. Solid effluents

The IGCC plant is expected to produce the following solid by-product:

Slag from gasifier

Flowrate : 38 t/h (dry basis)

Moisture content : 50%

Slag product has a potential use as major components in concrete mixtures to make road, pads, storage bins.

8. Equipment list

The list of main equipment and process packages is included in this section.

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 900 - Solid handling

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	SOLID HANDLING							
PK- 901	**Coal Handling**		Coal flowrate: 305 t/h					***30 days storage***
	Including:							*Storage piles: 2 x 110,000 t each*
	Wagon tipper							
	Receiving Hopper, vibratory feeder and belt extractor							
	Conveyors	Belt						
	Transfer Towers	Enclosed						
	As-Received Coal Sampling System	Two-Stage						
	As-Received Magnetic Separator System	Magnetic Plates						
	Conveyors	Belt						
	Transfer Towers	Enclosed						
	Crushers Towers	Impactor reduction						
	As-Fired Coal Sampling System	Swing hammer						
	As-Fired Magnetic Separator System	Magnetic Plates						
	Coal Silo		2 x 5300 m3					*for daily storage*

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	

REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
DATE	Aug-14	Feb-15		
ISSUED BY	NF	NF		
CHECKED BY	LM	LM		
APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 1000 - GE Gasification island

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	GASIFICATION ISLAND							
PK- 1000 - 1/2	**GE Energy Coal gasification package** *Each including:* Coal grinding and slurry preparation Gasifiers (RSC) Scrubber Black Water flash Coarse slag handling Grey water systeam and fines handling		2 x 3650 t/d coal (as received) to burners 2 x 775 MWth (LHV basis) syngas at scrubber outlet					***2 x 50% gasification package***

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC					
		REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 2100 - Syngas Treatment and conditioning line (2X50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	EXCHANGERS (*)							
E- 2101	Feed / Product Heat Exchanger	shell and tube						
E- 2102	LP steam generator	kettle						
E- 2103	VLP steam generator #1	kettle						
E- 2104	VLP steam generator #2	kettle						
E- 2105	MP steam heater	kettle						
E- 2106	Condensate pre-heater	shell and tube						
E- 2107	Final syngas cooler	shell and tube						
	DRUM (*)							
D- 2101	Flash drum #1	vertical						
D- 2102	Flash drum #2	vertical						
D- 2103	Flash drum #3	vertical						
D- 2104	Flash drum #4	vertical						
D- 2105	Flash drum #5	vertical						
D- 2106	Condensate accumulator	horizontal						
	REACTORS (*)							
R- 2101	CO hydrolysis	vertical						
	EXPANDER (*)							
T- 2101	Syngas expander							
	MISCELLANEA (*)							
X- 2101	Mercury adsorber	sulphur impregnated activated carbon beds						

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 2100 - Syngas Treatment and conditioning line (2X50%)

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	SWS stripper package (1x100%)							
	Sour water stripper SWS reboiler SWS condenser Stripper feed product exchanger SWS pump							

(*) Equipment list referred to one train only

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 2200 - Acid gas removal unit

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	ACID GAS REMOVAL							
PK- 2200	**Acid Gas Removal Unit**	Solvent: Selexol	Feed gas: 600,000 Nm3/h; 56 barg; 34 °C					
	Absorption section		0.3% S in the syngas feed < 15 ppm in the syngas product					
	H2S concentrator		N2 stripping					
	Solvent regeneration		40% H2S content in acid gas to SRU					

FOSTER WHEELER	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
	LOCATION: The Netherlands	DATE	Aug-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: GE based IGCC with SCOC-CC	APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 2300 - Sulphur Recovery unit and Tail Gas Treatment

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	ACID GAS REMOVAL							
PK- 2200	**Sulphur Recovery Unit and Tail Gas Treatment Package** - two Sulphur Recovery Unit, each sized for 100% of the capacity - one Tail Gas Treatment Unit sized for 100% of capacity (including Reduction Reactor and Tail Gas Compressor)		Sulphur Prod.= 62 t/d Acid Gas from AGR = 4910 Nm3/h Expected Treated Tail Gas = 2930 Nm3/h					Sulphur content > 99,9 % min (dry basis)

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST

Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	GAS TURBINE							
PK- 3101-1/2	**Gas turbine and Generator Package**							***2 x 50% gas turbine package***
	Each including:							
	Gas turbine		320 MWe					*One per train, two in total*
	Gas turbine generator							*One per train, two in total*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	HEAT RECOVERY STEAM GENERATOR							
PK- 3201-1/2	**Heat recovery steam generator**	Horizontal, Natural Circulated, 4 Pressure Levels, Simple Recovery, Reheated						***2 x 50% HRSG package***
	Each including:							
D- 3201	HP steam drum							
D- 3202	MP steam drum							
D- 3203	LP steam drum							
D- 3204	VLP steam drum with degassing section							
E- 3201	HP Superheater 2nd section							
E- 3202	MP Reheater 2nd section							
E- 3203	HP Superheater 1st section							
E- 3204	MP Reheater 1st section							
E- 3205	HP Evaporator							
E- 3206	HP Economizer 3rd section							
E- 3207	MP Superheater							
E- 3208	MP Evaporator							
E- 3209	LP Superheater							
E- 3210	HP Economizer 2nd section							
E- 3211	MP Economizer 2nd section							
E- 3212	LP Evaporator							
E- 3213	Condensate heater							
E- 3214	HP Economizer 1st section							
E- 3215	MP Economizer 1st section							
E- 3216	LP Economizer 1st section							
E- 3217	VLP Evaporator							
X- 3201	HP steam desuperheater							
X- 3202	MP steam desuperheater							
X- 3203	Start-up Stack							*Including silencer and CEMS*

	CLIENT: IEAGHG	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
FOSTER WHEELER	LOCATION: The Netherlands	DATE	Aug-14	Feb-15		
	PROJ. NAME: Oxy-turbine power plants	ISSUED BY	NF	NF		
	CONTRACT N.: 1-BD-0764 A	CHECKED BY	LM	LM		
	CASE.: GE based IGCC with SCOC-CC	APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	HEAT RECOVERY STEAM GENERATOR							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3201 A/B	HP BFW pumps	Centrifugal	460 m3/h x 1730 m	2600 kW				*One operating one spare, per each train*
P- 3202 A/B	MP BFW pumps	Centrifugal	70 m3/h x 580 m	160 kW				*One operating one spare, per each train*
P- 3203 A/B	LP BFW pumps	Centrifugal	220 m3/h x 120 m	90 kW				*One operating one spare, per each train*
P- 3204 A/B	VLP BFW pumps	Centrifugal	40 m3/h x 80 m	15 kW				*One operating one spare, per each train*
	HEAT EXCHANGER							
	Blowdown cooler							
	DRUM							
	Continuous Blowdown drum							
	Intermittent Blowdown drum							
	PACKAGES (Common to both train)							
PK- 3202	**Fluid Sampling Package**							
PK- 3203	**Phosphate Injection Package**							
	Phosphate storage tank							
	Phosphate dosage pumps							*One operating one spare*
PK- 3204	**Oxygen scavenger Injection Package**							
	Oxygen scavenger storage tank							
	Oxygen scavenger dosage pumps							*One operating one spare*
PK- 3204	**Amine Injection Package**							
	Amine storage tank							
	Amine dosage pumps							*One operating one spare*
PK- 3101-1/2	**Indirect contect cooler**							

FOSTER WHEELER	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 3000 - Oxy-turbine Cycle

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	STEAM TURBINE							
PK- 3301	**Steam turbine and Generator Package**							***1 x 100% package***
	Including:							
ST- 3301	Steam turbine		450 MWe					*Including:* *Lube oil system* *Cooling* *Idraulic control system* *Seals system* *Drainage system*
G- 3301	Steam turbine generator		550 MVA					*Including relevant auxiliaries*
	Inlet/after condenser Gland Condenser							
PK- 3302	**Steam Condenser Package** Each including:							***1 x 100% condenser package***
	Steam condenser		745 MWth					*Including:* *Condenser hotwell* *Ejector* *Start-up Ejector*
PK- 3303	**Steam Turbine by-pass system**							
	PUMPS		**Q [m3/h] x H [m]**					
P- 3301 A/B	Condensate pumps	Centrifugal	1110 m3/h x 170 m	670 kW				*One operating one spare*

EQUIPMENT LIST

Unit 4000 - CO2 compression and purification

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	CO2 COMPRESSION AND PURIFICATION							
PK - 4001	**CO2-rich gas compression**							
	Including:							
	Raw flue gas compressors							
	- *Raw flue gas compressor #1*	axial	Flowrate: 2 x 215,000 Nm3/h Pin: 1.02 bar; Pout: 15 bar Compression ratio: 14.7	*2 x 30 MWe*				*2x50%*
	- *Raw flue gas compressor #2*	axial	Flowrate: 2 x 230,000 Nm3/h Pin: 14.4 bar; Pout: 35 bar Compression ratio: 2.44	*2 x 8.5 MWe*				*2x50%*
	Condesate separators							
	Intercoolers							
	Inert gas heater							
	Oxygen heater							
	Syngas heater							
	Cooling water intercoolers							
PK - 4002	**Dual Bed dessicant system**							*1x100%*
PK - 4003	**Cold box for inerts removal**							*1x100%*
	Including:							
	- Main heat exchangers							
	- CO2 liquid separator							
	- CO2 distillation column							
	- CO2 compressor 1	centrifugal	Flowrate: 2 x 82,000 Nm3/h	*2 x 3.5 MWe*				*2x50%*
	- CO2 compressor 2	centrifugal	Flowrate: 2 x 164,000 Nm3/h	*2 x 14 MWe*				*2x50%*
	- Inerts heater							
	- Inerts expander		7600 kW					*1x100%*
	- Overhead recycle compressors (optional)							
	- Intercoolers							
	Inert gas heater							
	Condensate heater							
	Cooling water intercoolers							

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 5000 - Air Separation Unit

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [MW]	P des [barg]	T des [°C]	Materials	Remarks
	AIR SEPARATION UNIT							
PK - 5001-1/2	**Air Separation unit**		3 x 4990 t/d Oxygen purity: 97%mol Oxygen pressure: 50 bar					*3x33% unit*
	Each including							
	- Main Air Compressors	Centrifugal		2 x 32.5 MWe				*Four stages, intercooled*
	- Booster air compressor	Centrifugal		8.5 MWe				
	- Compander							
	- Air purification system							
	- Main heat exchanger							
	- ASU compander							
	- ASU Column System							
	- Pumps	Centrifugal						
	- ASU chiller							
	- Oxygen to GI compressors	Centrifugal		1.6 MWe				*One for each GI train*
TK - 5001	LOX storage tank		1800 t					*Common to all trains. Corresponding to 3 hours O2 production from one ASU, in order to enhance ASU reliability*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC					REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
						DATE	Aug-14	Feb-15		
						ISSUED BY	NF	NF		
						CHECKED BY	LM	LM		
						APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	COOLING SYSTEM							
CT- 6001 A/B	**Cooling Tower**	Natural draft	2 x 660 MWth				concrete	
	including:		Diameter: 95 m,					
	Cooling water basin		Height: 180 m,					
			Water inlet: 17 m					
	PUMPS		**Q [m3/h] x H [m]**					
P- 6001 A/…/I	**Cooling Water Pumps**	Centrifugal	13,500 m3/h x 40 m	1800 kW				*Eight in operation, one spare*
P- 6002 A/B /C	**Cooling tower make up pumps**	Centrifugal	1,000 m3/h x 30 m	125 kW				*Two operating, one spare*
	PACKAGES							
	Cooling Water Filtration Package							
	Cooling Water Sidestream Filters		10300 m3/h					*Common for two towers*
	Sodium Hypochlorite Dosing Package							
	Sodium Hypochlorite storage tank							
	Sodium Hypochlorite dosage pumps							
	Antiscalant Package							
	Dispersant storage tank							
	Dispersant dosage pumps							
	RAW WATER SYSTEM							
PK- 6001	Raw Water Filtration Package							
T- 6001	Raw Water storage tank							*12 hour storage*
P- 6003 A/B	Raw water pumps	centrifugal						*One operating, one spare*
T- 6002	Potable storage tank							*12 hour storage*
P- 6004 A/B	Portable water pumps	centrifugal						*One operating, one spare*

	CLIENT: IEAGHG LOCATION: The Netherlands PROJ. NAME: Oxy-turbine power plants CONTRACT N.: 1-BD-0764 A CASE.: GE based IGCC with SCOC-CC	REVISION	Rev.: Draft	Rev.: 0	Rev.1	Rev.2
		DATE	Aug-14	Feb-15		
		ISSUED BY	NF	NF		
		CHECKED BY	LM	LM		
		APPROVED BY	LM	LM		

EQUIPMENT LIST
Unit 6000 - Utility units

ITEM	DESCRIPTION	TYPE	SIZE	Motor rating [kW]	P des [barg]	T des [°C]	Materials	Remarks
	DEMINERALIZED WATER SYSTEM							
PK- 6002	Demin Water Package, including:							
	- Multimedia filter							
	- Reverse Osmosis (RO) Cartidge filter							
	- Electro de-ionization system							
T- 6003	Demin Water storage tank							*12 hour storage*
P- 6005 A/B	Demin water pumps	centrifugal						*One operating, one spare*
	FIRE FIGHTING SYSTEM							
T- 6004	Fire water storage tank							
	Fire pumps (diesel)							
	Fire pumps (electric)							
	FW jockey pump							
	OTHER UTILITIES							
	Natural gas receiving station							
	Plant & instrument air							
	Emergency diesel generator system							
	Waste water treatment							
	Electrical equipment							
	Buildings							
	Auxiliary boiler							
	Condensate polishing system							
	Flare system							
	Sulphur storage and handling		62 t/d Sulphur prod.					

9. Economic assessment

The purpose of this section is to present the results of the economic analysis, carried out to evaluate the Levelized Cost of Electricity (LCOE) and the CO_2 Avoidance Cost (CAC) of the IGCC study case.

Capital cost and operating & maintenance (O&M) costs have been evaluated and are presented in this section, along with the results of the simplified financial model.

All economical inputs used to perform this analysis are set in accordance with the economic bases reported in chapter B of this report.

The reference plant considered for the financial assessment of the IGCC with oxygen fired gas turbine (SCOC-CC) is the SC PC plant w/o CCS, as assessed in the IEAGHG report 2014/3, '*CO_2 capture at coal based power and hydrogen plant*'.

9.1. Capital costs

The main cost estimating bases are described in chapter B of this report, while definition of Total Plant Cost (TPC), Total Capital Requirements (TCR) and the estimating methodology are reported in chapter E of the report, section 2.

Table 7 shows the TPC of the IGCC, divided into the main process units. The table is followed by the related pie chart of the total plant cost to show the percentage weight of each unit on the overall capital cost of the plant.

A summary of the main investment cost figures, including also the specific TPC and TCR per net power production are also reported in the below Table 6, showing also the comparison with the conventional IGCC plant based on the same gasification technology.

Table 6. GE based IGCC – SCOC-CC vs Conventional CC

		GE based IGCC w SCOC-CC	GE based IGCC (conventional CC)
TPC	**M€**	**2,611.7**	**2,668.1**
Syngas generation train (*)	M€	942.2	1166.7
Air Separation Unit	M€	449.6	364.5
CO_2 purification and compression	M€	198.2	83.7
Combined cycle	M€	664.2	676.9
Utility and Offsites	M€	357.5	396.3
TCR	**M€**	**3596.7**	**3704.6**
Specific cost (TPC/Net Power)	**€/kW**	**3,540**	**3,080**
Specific cost (TCR/Net Power)	**€/kW**	**4,880**	**4,240**

* including solid handling gasification, syngas treat. AGR, SRU

Table 7. Oxy-turbine based IGCC – Total plant cost

OXY-TURBINE POWER PLANT
GE BASED IGCC + SCOC-COMBINED CYCLE

CONTRACT: 1-BD-0764A
CLIENT: IEAGHG
LOCATION: THE NETHERLANDS
DATE: SEPTEMBER 2014
REV.: 0

POS.	DESCRIPTION	UNIT 900 SOLID HANDLING	UNIT 1000 GASIFICATION ISLAND	UNIT 2100 SYNGAS TREATMENT	UNIT 2200+2300 AGR, SRU + TGT	UNIT 3000 COMBINED CYCLE	UNIT 4000 CO2 PURIFICATION UNIT	UNIT 5000 ASU	UNIT 6000 UTILITY UNITS	TOTAL COST EURO	NOTES / REMARKS
1	DIRECT MATERIAL	49,800,000	312,500,000	37,450,000	86,800,000	320,600,000	119,600,000	239,300,000	191,400,000	1,357,450,000	1) Gross power output: MW 1075.6 Specific cost €/kW : 2,430
2	CONSTRUCTION + OTHER COSTS	21,600,000	153,300,000	21,000,000	53,700,000	236,100,000	38,900,000	134,400,000	95,500,000	754,500,000	
3	EPC SERVICES	10,900,000	71,000,000	10,300,000	28,100,000	47,100,000	21,700,000	35,000,000	38,100,000	262,200,000	2) Total Net Power : MW 737.5 Specific cost €/kW : 3,540
4	**TOTAL INSTALLED COST**	**82,300,000**	**536,800,000**	**68,750,000**	**168,600,000**	**603,800,000**	**180,200,000**	**408,700,000**	**325,000,000**	**2,374,150,000**	
5	PROJECT CONTINGENCY	8,200,000	53,700,000	6,900,000	16,900,000	60,400,000	18,000,000	40,900,000	32,500,000	237,500,000	
6	PROCESS CONTINGENCY	EXCLUDED	EXCLUDED	EXCLUDED	EXCLUDED	EXCLUDED	EXCLUDED	EXCLUDED	EXCLUDED	-	
7	**TOTAL PLANT COST (TPC)**	**90,500,000**	**590,500,000**	**75,650,000**	**185,500,000**	**664,200,000**	**198,200,000**	**449,600,000**	**357,500,000**	**2,611,650,000**	

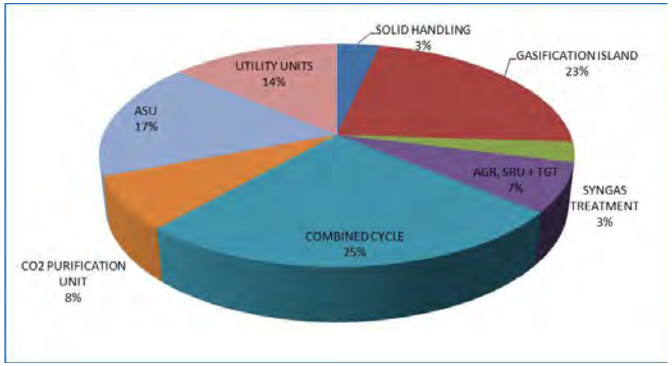


Figure 5. Unit percentage weight on TPC

9.2. Operating and Maintenance costs

The definition of the Operating and Maintenance (O&M) costs is given in chapter B of the report. The following sections provide estimated operating and maintenance costs for the IGCC case, including variable and fixed costs.

9.2.1. Variable costs

The following Table 8 reports a summary of the variable costs for the IGCC cases, including:

- Coal (2.5 €/GJ LHV basis)
- Raw water make-up
- Solvent
- Catalysts
- Chemicals.

The consumption of the various items and the corresponding costs are yearly, based on the expected equivalent availability of the plant (85%).

Table 8. Oxy-turbine based IGCC – Variable operating costs

FOSTER WHEELER			Yearly Variable Costs		
Yearly Operating hours = 7446			IGCC with SCOC-CC		
Consumables	**Unit Cost**		**Consumption**		**Oper. Costs**
			Hourly	Yearly	
		€/t	kg/h	t/y	€/y
Feedstock					
Coal		64.7	302,905	2,255,428	145,869,800
Fluxant (Limestone)		20.0	0	0	0
Auxiliary feedstock					
Make-up water		0.2	1,750,000	13,030,500	2,606,100
Catalysts		not displayable	-	-	1,500,250
Chemicals (including Solvents[1])		not displayable	-	-	2,421,650
Waste Disposal					
Slag disposal (wet)		0.0	0.0	0	0
Solvent disposal		-	-	-	0
TOTAL YEARLY OPERATING COSTS		**Euro/year**			**152,397,800**

(1) Solvent cost and make-up: confidential information

9.2.2. *Fixed costs*

Fixed costs include:

- Operating Labour Costs
- Overhead Charges
- Maintenance Costs.

The following report the labour force for the different configurations, along with the direct labour cost, virtually divided into the following main areas of operation:

- Syngas generation trains and CPU
- ASU and utilities
- Power island

Table 9. Oxy-turbine based IGCC – Operating labor costs

IGCC w SCOC-CC					
	ASU & Utilities	**Syngas generation and CPU**	**Power Island**	**TOTAL**	**Notes**
OPERATION					
Area Responsible	1	1	1	3	daily position
Assistant Area Responsible	1	1	1	3	daily position
Shift Superintendent		5		5	1 position per shift
Electrical Assistant		5		5	1 position per shift
Shift Supervisor	5	5	5	15	3 positions per shift
Control Room Operator	5	10	10	25	5 positions per shift
Field Operator	10	30	20	60	12 positions per shift
Subtotal				**116**	
MAINTENANCE					
Mechanical group		4		4	daily position
Instrument group		7		7	daily position
Electrical group		5		5	daily position
Subtotal				**16**	
LABORATORY					
Superintendent+Analysts		6		6	daily position
Subtotal				**6**	
TOTAL				**138**	
Cost for personnel					
Yearly individual average cost =		60,000	Euro/year		
Total cost =		8,280,000	Euro/year		

The overhead charges cost is evaluated as 30% of the operating labour and maintenance labour cost. The annual maintenance cost of the plant is estimated as 2.5% of the Total Plant Cost.

9.2.3. *O&M summary*

The following table reports the summary of the O&M costs for IGCC case.

	IGCC w SCOC-CC O&M COSTS (2014) €/year
Fixed Costs	
Direct labour	8,280,000
Adm./gen overheads	10,319,000
Insurance & Local taxes	26,116,500
Maintenance	65,291,300
Subtotal	**110,006,800**
Variable Costs (Availability = 85%)	
Feedstock	145,869,800
Water Makeup	2,606,100
Catalyst	1,500,250
Chemicals (including Solvent)	2,421,650
Waste disposal (incl. Solvent)	0
Subtotal	**152,397,800**
TOTAL O&M COSTS	**262,404,600**

9.3. Financial analysis

This section summarizes the results of the simplified financial analysis performed for the IGCC study case. Main input data and definitions are reported in chapter E, section 4 of the report, while principal financial bases assumed for the financial modelling are reported also hereafter for reader's convenience.

ITEM	DATA
Type of fuel	Coal at 2.5 €/GJ (LHV)
Discount Rate	8%
Capacity factor	85%
CO_2 transport & storage cost	10 €/t STORED
CO_2 emission cost	0 €/t EMITTED
Inflation Rate	Constant Euro
Currency	Euro reported in 2Q2014

Figure 6 and Figure 7 report the LCOE and CAC for the oxy-turbine IGCC case, compared with the conventional IGCC. Reference plant for the CAC calculation is the SC PC plant w/o capture as reported in IEAGHG report 2014/3. LCOE figures also show the relative weight of:

- Capital investment,
- Fixed O&M,
- Variable O&M,
- Fuel,
- CO_2 transportation & storage,
- CO_2 emission.

A summary of the economical modelling results is also reported in the following Table 10.

Table 10. Financial results summary: LCOE and CO_2 avoidance cost

Case	LCOE €/MWh	CAC €/t
SC PC w/o CCS	52.0	-
IGCC w SCOC-CC	126.8	114.8
IGCC w conventional CC	114.4	95.8

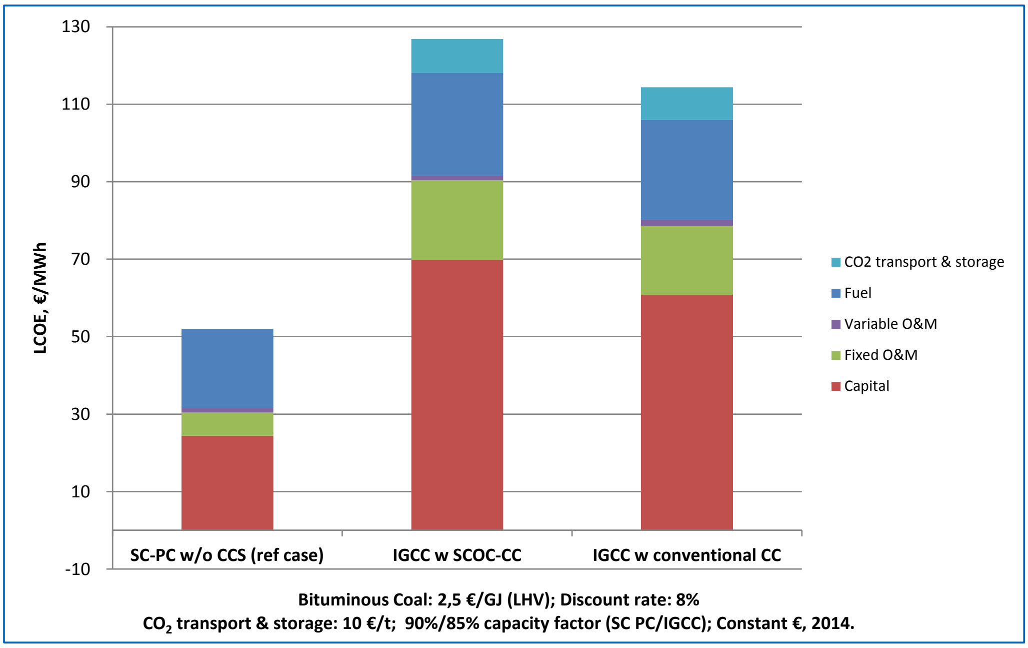


Figure 6. LCOE

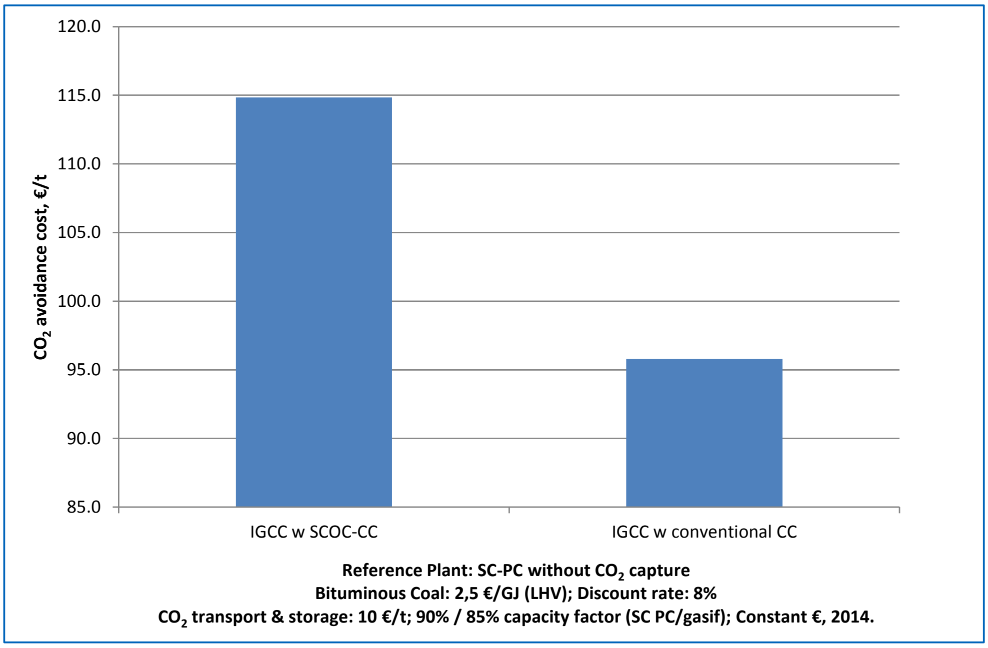


Figure 7. Cost of CO_2 avoidance