A THERMODYNAMIC COMPARISON OF THE OXY-FUEL POWER CYCLES WATER-CYCLE, GRAZ-CYCLE AND MATIANT-CYCLE

Olav Bolland, Norwegian University of Science and Technology, N-7491 Trondheim, Norway,
Tel: +47 73591604 - Fax: +47 73598390 – E-mail: Olav.Bolland@tev.ntnu.no
Hanne M. Kvamsdal, SINTEF Energy Research, N-7465 Trondheim, Norway,
Tel: +47 73550465 - Fax: +47 73593580 – E-mail: Hanne.Kvamsdal@energy.sintef.no
John C. Boden, (CO_2 Capture Project), BP Oil International, Sunbury-on-Thames, TW16 7LN, UK,
Tel: +44 1932 771929 – Fax: +44 1932 763439– E-mail: bodenjc@bp.com

INTRODUCTION

With the aim of developing technologies to reduce the cost of capture and geological storage of carbon dioxide from fossil fuel combustion, an international programme, The CO_2 Capture Project, funded by nine leading energy companies, has been established. One of the technology areas targeted is oxy-fuel combustion, since this generates a flue gas consisting largely of carbon dioxide and water from which carbon dioxide is easily separated. The use of oxy-fuel combustion in gas turbine based power generation will require new equipment, but also provides an opportunity to develop new cycles which may offer higher efficiencies than current air-based combined cycle systems, thus partially offsetting the additional cost of oxygen production. Several such cycles have been proposed in the literature and this theoretical study compares the thermodynamic performance of three of them, referred to here as the Water-cycle, the Graz-cycle and the Matiant-cycle.

An evaluation of the three concepts is given in the following sections. The evaluation of the Water cycle concept and the Graz-cycle concept is based on simulations performed by the simulation tool PRO/II (SIMSCI Inc.) while the evaluation of the Matiant-cycle is based on literature statements only.

Computational assumptions. The following assumptions are common for the two computational parts of the present study:

- The SRK (Soave-Redlich-Kwong) thermodynamic system including use of steam tables in PRO/II was used for calculation of thermodynamic properties.
- The fuel is natural gas with the following composition: C1; 81.2 vol%, C2; 8.97 vol%, C3; 4.26 vol%, C4; 2.35 vol%, C5+; 1.0 vol%. Fuel pressure; 50 bar.
- The oxygen for the combustion is produced at atmospheric pressure with an energy requirement of 906 kJ/kg oxygen (0.25 kWh/kg).
- The heat exchangers were calculated with a pressure drop of 3%. The combustor(s) were calculated with 4% pressure drop. The adiabatic efficiency of 88% was assumed for all compressors while the turbines were calculated with an adiabatic efficiency of 85% (turbine cooling losses included).
- In the present work the compression of CO_2 from the condenser pressure to 1 bar is calculated, using a polytropic efficiency of 75%-85%. For intercooled compression of CO_2 from 1 bar to the specified end pressure of 100 bar, a specific value of 390 kJ/kg (about 0.11 kWh/kg) was used ([1]).

WATER-CYCLE

The evaluation of the water cycle concept is to a large extent focused on publications given by [2]-[4]. The Water-cycle can be categorised as a Rankine type power cycle. The working fluid (approximately 90% water) is compressed in the liquid phase, and hot gases are expanded to provide work. In the publications ([2] to [4]), there are various schemes for the cycle configuration with respect to the reheat arrangement. Both single and double reheat is applied.

A flowsheet diagram of the process applied in the present study is shown in Figure 1. Production of oxygen and the compression up to the specific pressure levels are not shown here, but the energy requirement is included in the total energy efficiency calculations. The fuel is compressed (FUEL_COMP) and preheated (E4) before the high-pressure combustion takes place (HP_COMBUSTOR). Oxygen (O2_COMB), from a cryogenic air separation unit, is fed in a stoichiometric ratio with the fuel in the combustor. The combustor exit temperature is controlled by adding water (H2O_MIXED). The combustor exit flow (HP_TI) is expanded in a turbine (HP_TURBINE). The turbine exit stream flows to a secondary, or reheat, combustor (REHEATER). By adding fuel (MP_FUEL) and oxygen (O2_REHEATER) to the reheater, the exit temperature of this unit is controlled. The HP_TURBINE inlet temperature is 871ºC which represent a very advanced steam turbine technology based on an uncooled, high pressure turbine while the REHEATER, in which the inlet temperature is 1427ºC, represents a standard gas turbine technology based on a cooled, medium pressure turbine. The reheater exit stream is expanded in a turbine (LP_TURBINE). The temperature of this stream (LP_EXHAUST1) is rather high, 730-960°C depending upon combustion pressure and temperature. The turbine exhaust consists of typically 90% steam and 10% CO_2. The exhaust is cooled down by fuel preheating (E1) and water heating (EX_COOLER). The water (H2O_MIXED) to the high-pressure combustor is preheated in the EX_COOLER. The exhaust starts to condense in the EX_COOLER and is further cooled by cooling water in the heat exchanger E3. Water and CO_2 is split in the CONDENSER. The computational units E3 and CONDENSER would in practice be one condenser unit. The CO_2 (in gaseous phase) is compressed to 1 bar. The water from the condenser is recycled (H2O_RECYCLE) back to the combustor, after compression (H2O_PUMP) and heating (EX_COOLER). However, a fraction of the water (H2O), equal to the amount of water formed in the combustion, is bled off from the process.

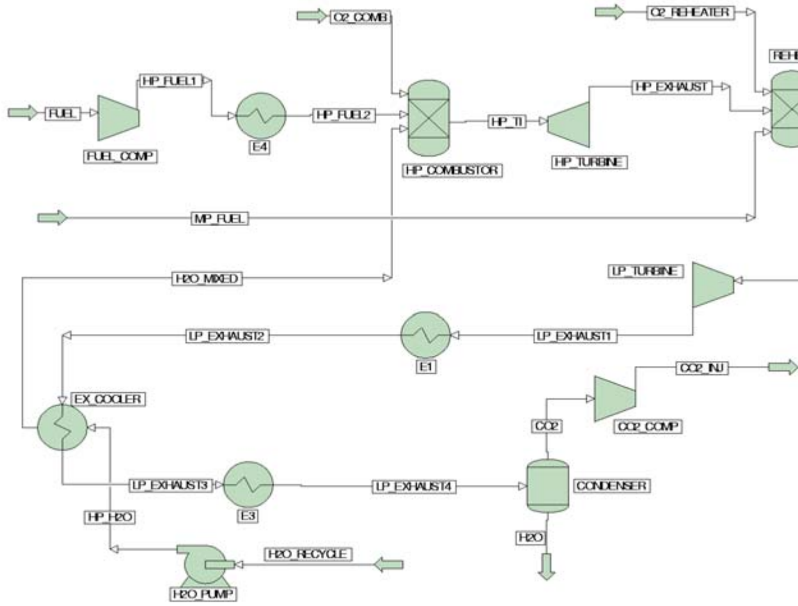

Figure 1 Flowsheet diagram of the Water-cycle as modelled in PRO/II.

Case specific assumptions. When varying the high-pressure combustor (HP_COMBUSTOR) pressure the reheat combustor (REHEATER) pressure is adjusted such that the pressure ratios over the turbines (HP_TURBINE and LP_TURBINE) are kept equal. In the large heat exchanger EX_COOLER a minimum internal temperature difference of 20 K is used. The exhaust is partly condensed in this heat exchanger.

Heat and mass balances. A base case was defined with the assumptions given earlier. This base case is meant to resemble the CES "near-term" cycle, as published by [2], using a single reheat cycle. A heat and mass balance was calculated for this base case. Additionally, calculations were carried out for a variation in some parameters:

1 High-pressure combustor (HP_COMBUSTOR) exit temperature (600-1450°C, base case =871°C)
2 High-pressure combustor (HP_COMBUSTOR) pressure (82.7-200 bar, base case =82.7 bar)
3 Condenser (CONDENSER) pressure (0.15-1.0 bar, base case =0.83 bar).

The reheat pressure was set to 8.27 bare in the base case and varied assuming constant HP/LP turbine pressure ratios.
The results from the three-parameter variations are presented in Figures 2–4, respectively. The upper line in the figures represents the gross efficiency (for the power cycle itself; shaft power minus pump work). The other lines show the efficiency when including a) the energy penalty for producing atmospheric gaseous oxygen, b) the energy penalty for compressing oxygen to the combustor pressure, and c) the energy penalty for compressing CO_2 from atmospheric pressure to 100 bar. The energy penalty, in terms of efficiency reduction, can be seen as the difference between the curves.

The gross plant efficiency (shaft work minus pump work related to fuel lower heating value) was calculated to be 51.7% for the base case. The net plant efficiency, including fuel compression (very small), oxygen production and compression of CO_2, was calculated to be 40.5% for the base case. When including generator efficiency (≈98.7%) and all plant auxiliaries, the net plant efficiency was estimated as 39.5%. This result is significantly lower than the claimed efficiency by [2] (=56%). That is even higher than the gross plant efficiency of the present study. It is not easy to extract the exact computational assumptions used in the publications by [2]-[4]. It is unclear whether the energy penalty for oxygen production and compression is included. A publication by [5] refers to a communication with CES indicating that the condenser pressure is 0.15 bar rather than the 0.83 given by [2].

The effect of varying the high-pressure combustor (HP_COMBUSTOR) exit temperature is shown in Figure 2. The net plant efficiency increases by 0.7-1.3 percentage points for every 100 °C of increased temperature. At 1400 °C exit temperature, the net plant efficiency is increased from 40.5% to about 45.5%. Note that in this parameter variation, the exit temperature of the reheater (REHEATER) is kept constant at 1427 °C.

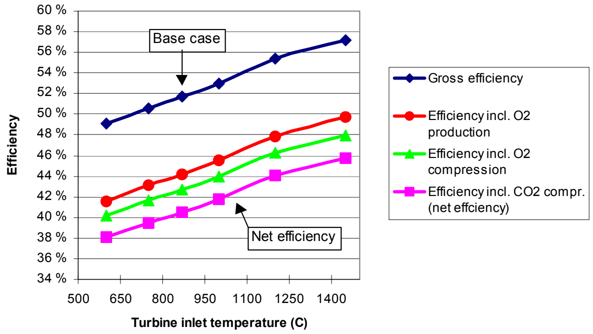

Figure 2 - Efficiency for the Water-cycle. High-pressure combustor (HP_COMBUSTOR) exit temperature varied in the range 600-1450 °C, base case temperature is 871 °C).

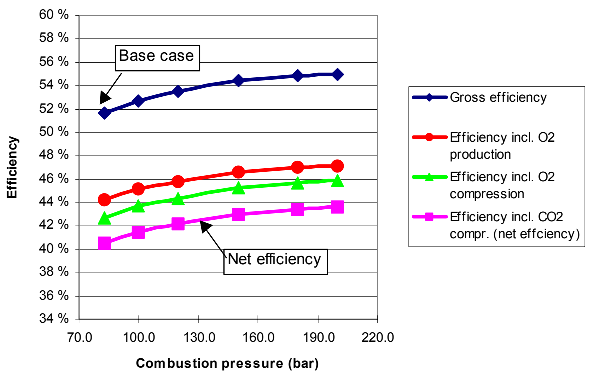

Figure 3 - Efficiency for the Water-cycle. High-pressure combustor (HP_COMBUSTOR) pressure varied in the range 82.7-200 bar, base case pressure is 82.7 bar.

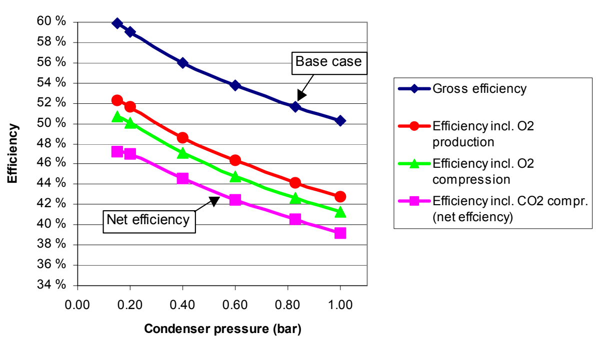

Figure 4 - Efficiency for the Water-cycle. Condenser (CONDENSER) pressure varied in the range 0.15-1.0 bar, base case pressure is 0.83 bar.

The effect of varying the high-pressure combustor (HP_COMBUSTOR) pressure is shown in Figure 3. The net plant efficiency increases about 3 %-points when increasing the pressure from the base value of 82.7 bar to around 200 bar. The effect of varying the condenser (CONDENSER) pressure is shown in Figure 4. The net plant efficiency increases by 0.5-1.2 percentage points for every 100 mbar (0.1 bar) change in efficiency.

The combined effects of increasing the high-pressure combustor exit temperature, increasing the combustion pressure, and reducing the condenser pressure were examined. Assuming that the high-pressure combustor exit temperature is set to 1200 °C, high-pressure combustor pressure set to 150 bar and condenser pressure is set to 0.15 bar, the net plant efficiency would be about 50.8% (base case 40.5%). If the high-pressure combustor exit temperature is set even higher, 1400 °C, and the pressure is increased to over 200 bar, an efficiency of about 53% is achievable combining the technology of a large steam turbine for the pressure and the high-temperature technology from advanced gas turbines. This assumption is a big challenge and is definitely "long-term". Efficiencies around 50-53% (shaft, lower heating value) would be comparable to (slightly above) the best currently available CO_2 capture technology (exhaust gas scrubbing by amine absorption) in a large modern Combined Cycle. It is consequently concluded that the Water-cycle does not offer any efficiency advantage over traditional Combined Cycle technology with exhaust gas scrubbing by amine absorption.

GRAZ-CYCLE

The evaluation of the Graz-cycle in the present study, including most parameter values, is based on publications by [6] and [7]. A flowsheet diagram of the process is shown in Figure 5. The high-temperature section of the Graz-cycle consists of a combustor (COMBUSTOR), which is fed with natural gas (FUEL), oxygen (O_2), a mixture of CO_2 and steam (CO_2+STEAM_2) and a stream of pure steam (STEAM_2). The combustor pressure is 50 bar. The combustor exit stream (HPT_IN), at 1440 ºC, is expanded in the high-pressure turbine (HPT). The turbine exit stream (HPT_EX) is split (SP1) after cooling in HX2, and is partly (≈38%) recycled back to the combustor after being compressed (COMP). The loop, in which this recycle stream flows, can be regarded as a recuperated steam injected oxy-fuel gas turbine cycle. The recuperation of exhaust gas heat takes place in HX2 and HX1. In these heat exchangers heat is transferred to water (H2O_4 and H2O_5), which is evaporated and superheated to 700 °C (same as reported by [6]). This steam temperature is very high compared to current design practice, which would rather indicate a temperature below 600 °C. The steam (STEAM_1) is then expanded (STEAM_EXP) from a pressure of 150 bar to the combustor pressure of 50 bar. The other stream from the splitter SP1 flows to a low-pressure system, consisting of two turbines (IPT and LPT) and a condenser (COOLER_1 and LP-CONDENSER). The water, to be used for steam generation and injection into the combustor (STEAM_2), is preheated by a bleed stream (≈10%) from the stream (S1) between the IPT and LPT turbines. This is very similar to regenerative feedwater preheating in steam power plants. The bleed stream contains steam and CO_2. The condensed water and CO_2 are separated in HP-CONDENSER. CO_2 gas is then present at two different pressures from the LP_CONDENSER and HP_CONDENSER. The low-pressure stream (CMP_1_IN) is compressed, cooled (COOLER_2) and drained for water (W-CONDENSER), before the two CO_2 streams (CO2_ATM_1 and CO2_ATM_2) are mixed (M1). Further compression of CO2_ATM_3 to 100 bar is not shown in Figure 5. The water collected from the streams WATER_1, WATER_2 and WATER_3 is partly recycled back to the cycle (H2O_1). Excess water formed in the combustion is removed.

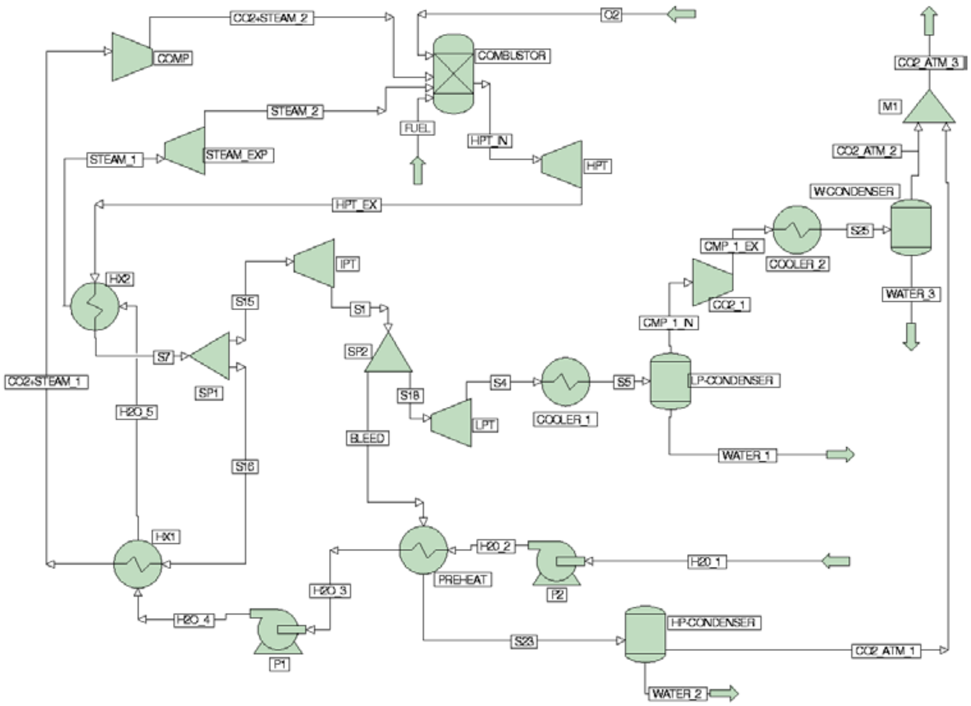

Figure 5 - Flowsheet diagram of the Graz-cycle as modelled in PRO/II

Case specific assumptions. In the large heat exchanger HX2 a minimum internal temperature difference of 76 K is used [6]. The lower pressure in the high-pressure cycle (HPT_EX) was set to 3.18 bar, according to [6].

Heat and mass balances. Calculations were carried out for a variation in condenser pressure. The presentation of the results in Figure 6 shows the net plant efficiency based on shaft power output minus pump work, energy requirement for oxygen production and work related to compression of CO_2, and lower heating value. As seen from Figure 6, the net efficiency decreases with increasing condenser pressure. The reason is that the LPT expansion work is higher than the CO_2 compression work such that the reduced expansion work only partly counteracts the decreased CO_2 compression work.
The net plant efficiency for the base case was calculated as 50.4% (condenser pressure 0.06 bar). The efficiency reported by [6] is 63.1%, without any energy penalty for the oxygen production and compression. In the present study, the energy penalty for the oxygen production and compression was calculated to about 10%-points, and the CO2 compression to about 2%-points. The calculated efficiency of the present work is therefore similar to that reported by [6].

The steam turbine inlet temperature of 700°C, and the condenser pressure of 0.06 bar seem to be beyond current technology. However, achieving a technology-enabling design with these parameters is not that far away.

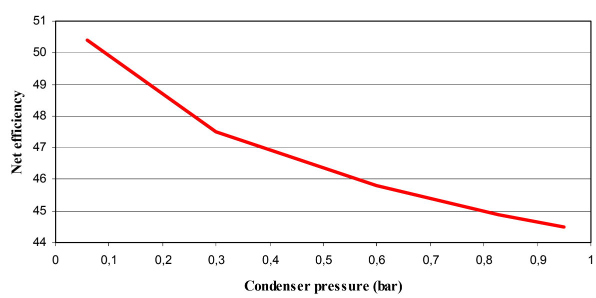

Figure 6 - Net efficiency for the Graz-cycle. The condenser (CONDENSER) pressure was varied in the range 0.06-0.83 bar (bleed pressure of 1.27 bar). TIT is 1440 ºC.

MATIANT-CYCLES

The Matiant-cycle originated in a Russian patent ([8]), which Professor Iantovski of the Moscow Institute of Energy Research ([9]) presented about 10 years ago. Later, both Iantowski and Mathieu have worked on research and concept development of the Matiant-cycle ([10] - [13]).

The Matiant-cycle is shown in a TS-diagram in Figure 7. This cycle is a recuperative Brayton-like cycle. The working fluid is compressed in the gaseous phase with intercooling (1-2). Then it is cooled such that a dense phase (as a liquid) can be pumped from point 3-4. At point 4, excess CO_2 is taken out of the process at the highest pressure (about 300 bar according to [11]). Then the working fluid is heated in a heat exchanger from point 4-5 and expanded to about 40 bar prior to reheating in a heat exchanger (6-7) and fed to a combustor (7-8). Then there is an expansion (8-9), a reheat combustor (9-10), and an expansion (10-11). The exhaust temperature after the last expansion is between 900-1000 °C. The heat of the exhaust is utilised for preheating of the compressed fluid 4-5 and 6-7.The Matiant-cycle involves a large amount of internal heat exchange between streams. The exhaust stream to be cooled (11-12) is at atmospheric pressure and at high temperature (900-1000 °C), and thus implies several potential problems related to heat exchanger technology. The net plant efficiency is calculated by [11] to be about 44-45%. The turbine inlet temperature (points 8 and 10) is set to 1300 °C, and penalties for turbine cooling and oxygen production (energy requirement: 0.28 kWh/kg O_2) are included.

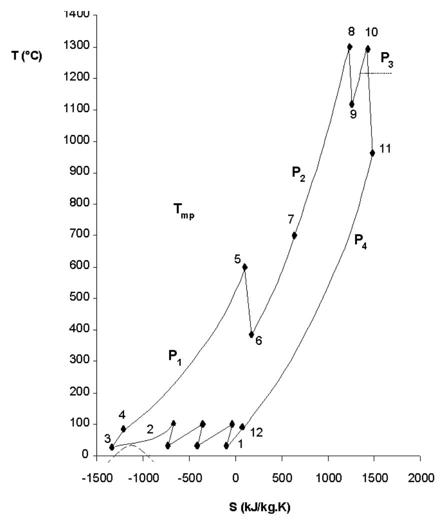

Figure 7 - TS diagram for the basic Matiant-cycle [13]. Supercritical part (2-6), with reheat (6-8-11-12-2), sequential combustion (7-8 and 9-10), staged expansion (8-9 and 10-11), recuperator (hot side: 11-12, cold side 4-5 and 6-7), water cooler/separator (12): 6% H_2O+0.02% of CO_2 recycled, staged compression with intercooling (1-3), CO_2 purge (4)=8% of CO_2 recycled.

The basic Matiant-cycle does not exhibit any thermodynamic advantage when taking into account parasitic losses (as in [11]). This is mainly due to the HP expander, however, further development of the basic Matiant-cycle has resulted in two different concepts called the E-Matiant-cycle and the CC-Matiant-cycle respectively. The E-Matiant-cycle is a fully Brayton-type cycle, with the whole cycle in the gas phase. It resembles very much an intercooled recuperative gas turbine cycle.

The CC-Matiant-cycle (see Figure 8) resembles several previous cycle proposals involving a CO_2/O_2 gas turbine (stoichiometric combustion with oxygen from an air separation unit) combined with a steam bottoming cycle. An example is the Norwegian HiOx concept. In this cycle, the compression is adiabatic instead of intercooled. The only difference between the CC-Matiant-cycle and the Combined Cycle with a CO_2/O_2 gas turbine and a steam bottoming cycle, is a recuperator between the hottest exhaust and the compressor discharge stream. The exhaust temperature of a CO_2/O_2 gas turbine is higher compared to an air gas turbine, for a given pressure ratio. This is because of different gas properties between the two cases. The combustion is stoichiometric. The oxidising agent is O_2 rather than air. The turbine inlet temperature is 1300°C. The novel idea of the CC-Matiant-cycle is to utilise the exhaust gas temperature between 600-700 °C for preheating of the compressor discharge flow, and thereby avoid a large temperature difference in the superheating of steam in the heat recovery steam boiler. It is difficult to see that this recuperator could give any advantage. The reason is that the compressor discharge temperature is around 500 °C (for a given pressure ratio in this case), and the compressor discharge stream is preheated to about 600 °C. This is not very different from the temperatures of the high-pressure steam superheater in the heat recovery boiler. When taking into account the parasitic losses (mainly pressure losses) of the recuperator, the ducting and the valves, it is questionable whether the recuperator contributes anything with respect to efficiency. The net plant efficiency is calculated by [13] to about 47-49%. This is comparable to other publications ([4], [14], [15] and [16]) for this specific cycle configuration.

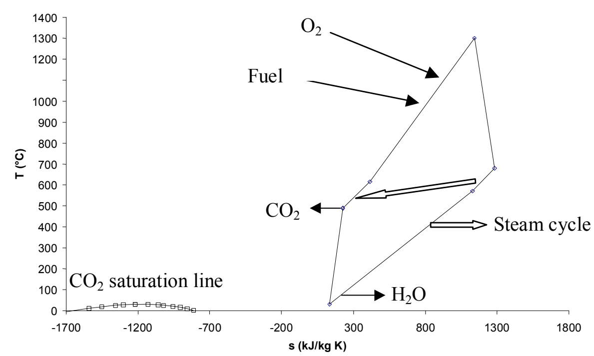

Figure 8 – TS diagram for the CC-Matiant-cycle [13]. Characterised by: Brayton like gas cycle, adiabatic compression and expansion, subcritical steam cycle (not shown), recuperator for hot exhaust → 600°C

The Matiant-cycle implies a series of technical challenges. These are mainly related to high-temperature technology and in particular to heat exchangers and turbines requiring cooling.

CONCLUSIONS

Three oxy-fuel power generation concepts (Water-cycle, Graz-cycle and Matiant-cycle), based on direct stoichiometric combustion with oxygen, are evaluated in the present study. From the point of view of thermodynamic efficiency, the three examined concepts seem to be rather similar with respect to plant efficiency.

When comparing the efficiency for the Graz-cycle with the Water-cycle one should compare the same condenser pressure. Assuming 0.15 bar condenser pressure, the Graz-cycle gives an efficiency of 49%, while the Water-cycle may give up to 53% when assuming very high combustion pressure and temperature. However, assuming the same level of technology, the Graz-cycle and the Water-cycle are similar in terms of efficiency.

The Graz-cycle is an interesting option as an oxy-fuel concept. The high-temperature/pressure loop closely resembles an oxy-fuel gas turbine, with the same challenges related to compressor, combustor and turbine. In order to reduce the gas turbine size, it has previously been proposed that the lower pressure-level of the gas turbine could be increased somewhat above atmospheric conditions. The Graz-cycle represents a very good solution to the problem of increased cycle heat rejection temperature, when increasing the lower pressure of the gas turbine cycle. Adding steam to the combustor may also help to reduce CO concentration for the stoichiometric combustion.

The efficiency of the Graz-cycle is slightly higher than for an oxy-fuel Combined Cycle ([16]), like the CC-Matiant-cycle. The main reasons for this are the following. The CO_2 that is produced in the combustor is expanded to a very low pressure (0.06 bar) compared to that commonly assumed for an oxy-fuel Combined Cycle (typically atmospheric pressure). Expansion of CO_2 has to be counteracted by compression of it to transportation pressure. The compression of CO_2 takes place with intercooling, so that the compression work is significantly less than the expansion work for a given equal pressure ratio in expansion/compression.

Finally, it should be noticed that CO_2 is not completely recovered in power cycles with H_2O condensers due to solubility of CO_2 in water. However, the solubility of CO_2 in the specific systems investigated here is maximum 1% at 1 bar, which corresponds to about 4 g/kWh. This value is acceptable compared to a conventional Combined Cycle, which emits about 400 g/kWh of CO_2. As the solubility is even lower at lower pressure, the Graz cycle seems more favourable regarding this issue.

ACKNOWLEDGEMENT

This study was funded by the CO_2 Capture Project (CCP), a joint programme of 9 leading energy companies aimed at developing technology to reduce the cost of CO_2 capture and geological storage.

REFERENCES

[1] Bolland O. and Undrum H., "Removal of CO_2 From Natural Gas Fired Combined Cycle Plants", presented at the PowerGen '99, Frankfurt, Germany, June 1999

[2] Anderson R., Brandt H., Mueggenburg H., Taylor J. and Viteri F., "A Power Plant Concept Which Minimizes The Cost Of Carbon Dioxide Sequestration And Eliminates The Emission Of Atmospheric Pollutants", Proceedings 4th International Conference on Greenhouse Gas Control Technologies, Interlaken, Switzerland, Sept. 1998.

[3] Anderson R., Brandt H., Doyle S.E., Mueggenburg H., Taylor J., Viteri F., "A Unique Process for Production of Environmentally Clean Electric Power Using Fossil Fuels. 8th International Symposium on Transport Phenomena and Dynamics Rotating Machinery. (ISROMAC-8), Honolulu, Hawaii, March 2000.

[4] Smith J.R., Surles T., Marais B., Brandt H., and Viteri F., "Power Production with Zero Atmospheric Emissions for the 21st Century", Presented to the 5th International Conference on Greenhouse Gas Control Technologies, Cairns, Queensland, Australia, August 13-16, 2000

[5] Bilger R.D., "Zero Release Combustion Technologies and the Oxygen Economy", Fifth International Conference Technologies and Combustion for a Clean Environment, Portugal, 1999

[6] Jericha H., Fesharaki M., "The Graz Cycle-1500°C Max Temperature Potential H2-O2 Fired CO2 Capture with CH4-O2 Firing", ASME paper 95-CTP-79, ASME Cogen-Turbo Power Conference, Vienna, 1995

[7] Jericha H., Sanz W., Woisetschläger J., Fesharaki M., "CO2-Retention Capability of CH4/O2-Fired Graz Cycle", CIMAC Paper G07, CIMAC Conference Interlaken, Switzerland, 1995

[8] Gochstein D.P., "Carbon dioxide power cycle", in Soviet Boiler- and Turbine construction, in Russian, No. 10, pp. 420-423, 1940

[9] Iantovski E., Zvagolsky K., and Gvarilenko V., "Computer exergonomics of power plants without exhaust gases", First International Conference on Carbon Dioxide Removal, Amsterdam, The Netherlands, March 4-6, 1992.

[10] Iantovski E. and Mathieu Ph., "Highly efficient zero emission CO_2-based power plant", Third Intern. Conf. on Carbon Dioxide Removal (ICCDR-3), Boston, 1996

[11] Mathieu, Ph., Nihart R., "Zero-emission MATIANT cycle", published in the ASME Journal of Engineering of gas turbines and power, Vol. 121, pp. 116-120, January 1999.

[12] Mathieu, Ph., Dubuisson, R., Houyou, S., Nihart R., "New concept of CO_2 removal technologies in power generation, combined with fossil fuel recovery and long term CO_2 sequestration", presented at the ASME Turbo Expo, Munich, May 2000

[13] Houyou, S., Mathieu, Ph, Nihart, R., "Techno-economic comparison of different options of very low CO_2 emission technologies", presented at the Fifth International Conference on Greenhouse Gas Control Technologies, Cairns, Australia, August 2000.

[14] De Ruyck J., 1992, "Efficient CO_2 Capture through a Combined Steam and CO_2 Gas Turbine Cycle", First Int. Conf. on CO_2 Removal, Amsterdam, 1992

[15] Dechamps P.J., Distelmans M., Mathieu Ph. and Pirard N., "Performances of Combined Cycles Power Plants Using CO_2 Gas Turbines", Flowers'94 Conference Proceedings, Florence (Italy) pp 671-682, 1994

[16] Bolland, O. and Mathieu, P., "Comparison of Two CO2 Removal Options in Combined Cycle Power Plants, Energy Conversion and Management, 39 (16-18), p.1653-1663, 1998; presented at the FLOWERS'97 conference July 30-August 1, 1997

[17] Bolland O. and Saether S., "New Concepts for Natural Gas Fired Power Plants which Simplify the Recovery of Carbon Dioxide", First International Conference on Carbon Dioxide Removal, Amsterdam, The Netherlands, March 4-6, 1992, published in *Energy Convers. Mgmt* vol.33, No. 5-8, pp.467-475, 1992