

EXERGY ANALYSIS OF GAS-TURBINE COMBINED CYCLE WITH CO₂ CAPTURE USING PRE-COMBUSTION DECARBONIZATION OF NATURAL GAS

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ABSTRACT

A concept for natural-gas fired power plants with CO₂ capture has been investigated using exergy analysis. The present approach involves decarbonization of the natural gas by autothermal reforming prior to combustion, producing a hydrogen-rich fuel. An important aspect of this type of process is the integration between the combined cycle and the reforming process. The net electric power production was 47.7% of the Lower Heating Value (LHV) or 45.8% of the chemical exergy of the supplied natural-gas. In addition, the chemical exergy of the captured CO₂ and the compression of this CO₂ to 80 bar represented 2.1% and 2.7%, respectively, of the natural-gas chemical exergy. For a corresponding conventional combined cycle without CO₂ capture, the net electric power production was 58.4% of the LHV or 56.1% of the fuel chemical exergy. A detailed breakdown of irreversibility is presented. In the decarbonized natural-gas power plant, the effect of varying supplementary firing (SF) for reformer-feed preheating was investigated. This showed that SF increased the total irreversibility and decreased the net output of the plant. Next, the effects of increased gas-turbine inlet temperature and of gas-turbine pressure ratio were studied. For the conventional plant, higher pressure led to increased efficiency for some cases. In the decarbonized natural-gas process, however, higher pressure ratio led to higher irreversibility and reduced thermal-plant efficiency.

NOMENCLATURE

ABS	Absorber
ATR	Auto-thermal reformer
CC	Combined Cycle
COND	Condenser
FC	Fuel compressor
GT	Gas Turbine

H1-H5	Heat exchangers
HC	Hydrocarbon
HP/MP	High/Medium pressure
HRSG	Heat Recovery Steam Generator
LHV	Lower Heating Value
LTS/HTS	Low/High Temperature Shift reactor
NG	Natural Gas
PRE	Pre-reformer
SF	Supplementary Firing
ST	Steam Turbine
TIT	Turbine Inlet Temperature
WR	Water Removal

INTRODUCTION

In order to reduce the CO₂ emission from natural-gas based power-generation plants, three different main types of concepts have emerged as the most promising.

A) Separation of CO₂ from exhaust gas coming from a standard gas-turbine combined cycle (CC), using chemical absorption by amine solutions [1,2].

B) Oxy-fuel CC with a close-to-stoichiometric combustion with oxygen (97%+ purity) from an air-separation plant as oxidizing agent, producing CO₂ and water vapor as the combustion products [3,4].

C) Decarbonization in which the carbon of the natural gas (NG) is removed prior to combustion and the fuel heating value is transferred to hydrogen. [5-9].

The present work focuses on concept C; decarbonization prior to combustion (pre-combustion). The motivation behind this concept is to utilize well-known technology. In *e.g.* ammonia production, hydrogen is produced by reforming of NG and subsequent removal of the CO₂. In the process described and analyzed in the following, this technology is used to produce

fuel for a conventional combined-cycle power plant, and the two processes are integrated. This work was based on previous results presented in [9]. There, a conceptual variation was performed concerning mainly the pressure level in the reformer part. It was concluded that it is more favorable to operate at a lower pressure in the reformer part than in the gas turbine.

The advance in the present work is a full implementation of the cycle model in PRO/II v.5.5 (SIMSCI Inc.), which makes parameter variations easier to carry out. The results from variations in supplementary firing (SF) for preheating of the reformer streams, gas-turbine pressure ratio and gas-turbine inlet temperature (TIT), are presented in the present work. The intention of the variation of gas-turbine TIT and pressure ratio is to put into perspective the future development in gas-turbine design, taking into account the integration with a pre-combustion CO₂ capturing process. The gas-turbine model in the combined cycle (CC) is based on the GTPRO (Thermoflow Inc.) gas-turbine database for the General Electric (GE) 9351FA technology. This gas turbine represents modern technology of

today, and it is used in a number of plants built in the last few years.

As in the preliminary work reported in [10], a detailed second-law analysis or exergy analysis has been performed in addition to first-order analysis in order to analyze this complex energy system more thoroughly. This is the main advancement compared to previous work. The exergy analysis quantifies and localises the thermodynamic losses (irreversibilities). Such knowledge is useful in explaining the changes observed in a parameter variation and in explaining differences between different processes. A similar exergy analysis was also made for the steam reforming process in [8].

PROCESS DESCRIPTION

The applied flowsheet was based on case 2 as reported in [9], in which the pressure in the reformer part and inlet of the gas turbine is approximately 15 and 20 bar, respectively. Fig. 1 shows the present process configurations.

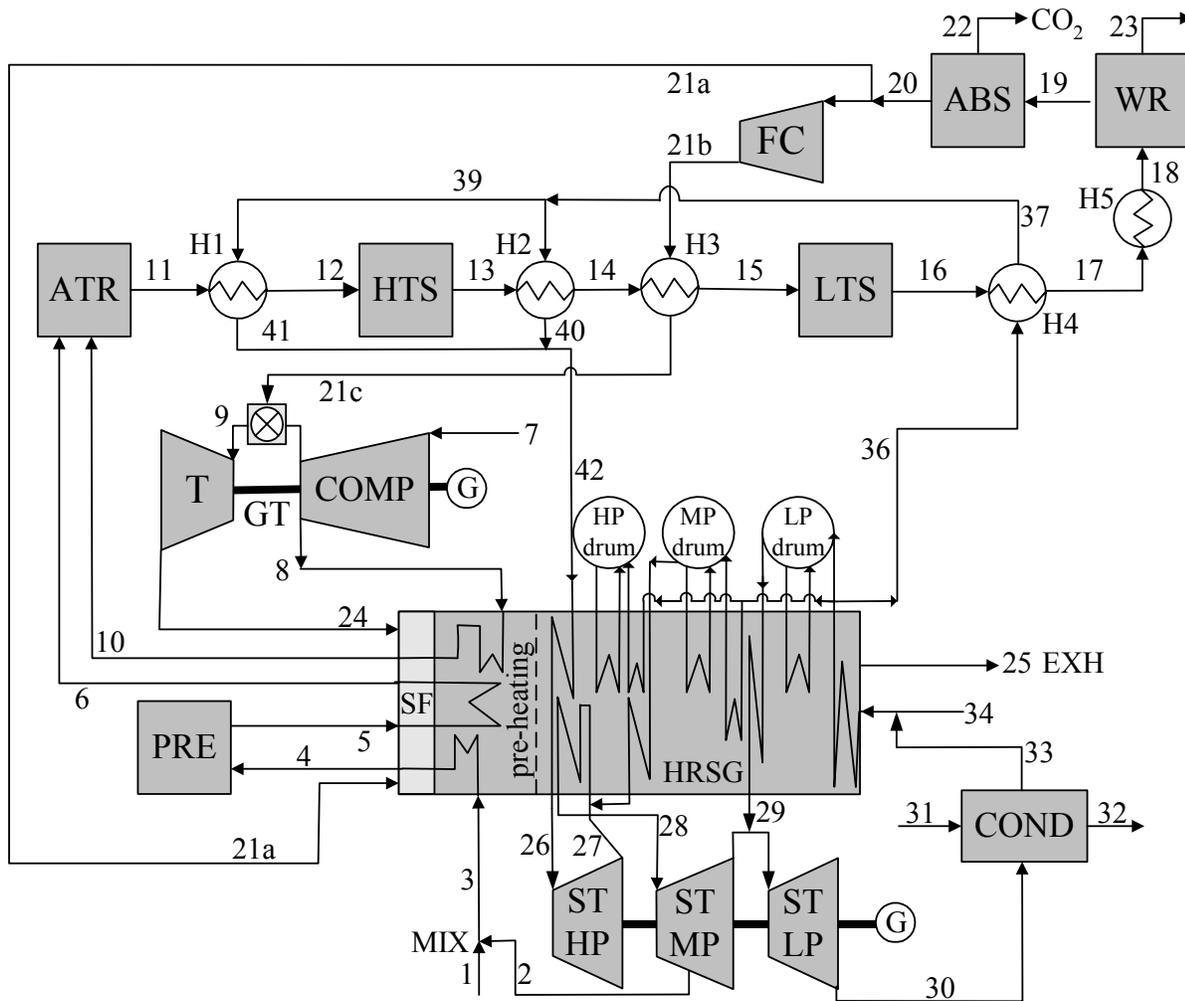


Fig. 1: Process flow diagram

The hydrogen-rich fuel is combusted in a gas turbine (GT), which is integrated with the reforming process. The gas turbine was modeled with units provided among the PRO/II standard unit models (a compressor unit, a Gibbs reactor, and an expander). The gas turbine performance model reflects the GE 9351FA technology. The considered steam cycle; the heat-recovery steam generator (HRSG), the steam turbine (ST), and the seawater-cooled condenser (COND), is an advanced process with three pressure levels and steam reheat.

The reforming process is supplied with high-pressure air (stream 8) and medium-pressure steam (2) from the gas-turbine compressor and the HRSG, respectively. The power plant and the reforming process are integrated with respect to preheating of feed streams for the reformers (auto-thermal reformer, ATR, and pre-reformer, PRE). The heat required in the reforming process is partly supplied by preheating of the reformer feed streams, and partly by the exothermic reaction (combustion) between oxygen and NG in the ATR. The combustion may not only take place in the ATR, but also in front of the preheating section of the HRSG by supplementary firing (SF). For a CC plant, SF reduces the efficiency. However, this is not so obvious for the process given in Fig. 1. The alternative to SF is increased extraction of air from the gas-turbine compressor to the combustion of NG in the ATR, which also decreases the efficiency of the CC. The present work evaluates the optimal method of supplying heat for the reforming process.

The required gas-turbine fuel-nozzle pressure is typically 25 % higher than the pressure of the air extracted from the gas-turbine compressor. Thus, an extra pressurization of the fuel is required. As it is more energy efficient to operate the reforming part at lower pressure, the fuel compressor (FC) is placed downstream this section.

In the reforming part, NG (stream 1) is mixed with medium-pressure steam (2) and preheated to 500 °C in the HRSG unit prior to the pre-reformer (PRE). The steam-to-carbon ratio was set to 2 at the pre-reformer inlet. Air extracted from the gas-turbine compressor (8) and the pre-reformer products (5) are preheated by the exhaust-gas stream upstream the ATR unit. The temperatures of both these streams are 15 °C below the exhaust-gas outlet stream temperature. Both the pre-reformer and the main reformer (ATR) are assumed equilibrium reactors. In the pre-reformer, most of the heavier hydrocarbon components (mainly C₂H₆) are converted to H₂, CO and CO₂, whereas the remaining methane is converted in the ATR unit. The ATR outlet temperature was set to 900 °C. The steam cycle takes advantage of the reforming process by utilizing the cooling process of the reformer products downstream the ATR to generate additional saturated high-pressure steam in the heat exchangers H1, H2, and H4 (streams 36–41). The saturated steam (42) is superheated in the HRSG unit and fed into the HP

steam turbine (26). The temperature of this stream is determined to 15 °C below the exhaust-gas outlet stream temperature, though limited by the maximum temperature of 560 °C. The pressure levels in the triple-pressure reheat steam-turbine cycle are 111 bar, 27 bar, and 4 bar, respectively, which is current design practice for large conventional CC plants.

The CO produced in the reforming process is converted to CO₂ in the high- and low-temperature shift reactors (HTS, LTS). Most of the water (99%+) is removed (stream 23) in the water-removal unit (WR) by condensation at 25 °C.

The main issue of this study was to investigate the described concept and details of the thermal plant, *i.e.* the reforming part and the power generation part and the integration between these. Thus, the CO₂ separator (ABS) was not investigated in detail. However, in the simulations, it was assumed that 90% of the CO₂ content is removed (22) in the chemical absorber unit (ABS). The duty in the heat exchanger H5 was assumed to represent the necessary duty of the stripper boiler in the absorption/desorption section. Therefore, the temperature out of heat exchanger H4 (stream 17) was required to be above 130 °C. This, and the throttling of the CO₂ to 1 atm (flow 22), was assumed to give sufficient exergy for the CO₂ separation. Further, it was assumed that the compression of CO₂ is accomplished using intercooled compression from 1 to 80 bar with 3 intercoolers (15 °C exit temperature) and compressor adiabatic efficiency ranging from 75% (high pressure) to 85% (low pressure).

The resulting fuel mixture (stream 20) contains mainly H₂ and N₂ (see Table 1). It also contains small amounts of CO, CO₂ and hydrocarbons. A fraction of the fuel flow may be used for SF (stream 21a) in the gas-turbine exhaust at the hot end of the HRSG. The remaining fuel (21b) is compressed (FC) to the required gas-turbine fuel-nozzle pressure, heated by the LTS feed stream (14), and then fed to the gas-turbine combustor (21c). Compared to conventional natural-gas fired gas turbines, the air volumetric flow is reduced by the air extraction for the reforming part (stream 8). However, this is compensated by the increased fuel volumetric flow. It is therefore possible to maintain the gas-turbine pressure ratio at approximately the same level as for a natural-gas fired gas turbine without any air extraction.

METHODOLOGY

The process shown in Fig. 1 was modeled in PRO/II v.5.5 (SIMSCI Inc.). The flowsheet simulations provided data for species mass flows and energy flows. It was assumed a pressure drop of 3 % in the pre-reformer, heat exchangers, and shift-reactors, whereas 6 % pressure-drop was assumed in the ATR.

The generator losses, auxiliary power, and work for compressing the removed CO₂ to 80 bar, and hence, the net electric power production, were calculated separately after the thermal-plant simulation.

The flowsheet simulations provided also the necessary data for calculating the physical (thermomechanical) exergy. Based on these data, the chemical exergy was calculated in a separate program according to the theory given in, *e.g.*, [10] and [11]. The chemical exergy of the individual species of this study were taken from [11] and corrected to the ambient temperature of 8 °C according to the procedure given therein. The composition of the dry atmosphere was defined by the molar fractions (%) N₂: 78.03, O₂: 20.99, Ar: 0.94, CO₂: 0.03. For the present simulations, the content of water vapor corresponds to a relative humidity of 82% at 8 °C and 1 atm, which was chosen as the environmental temperature and pressure. This was the state of the air entrained into the system. From this, the exergy of all streams were calculated. The irreversibility was then found

from the exergy balance for each of the individual unit processes.

RESULTS AND DISCUSSION

The process described above was simulated, and an energy and exergy analysis was made. Process data for the main streams are provided in Table 1. A corresponding conventional combined cycle without reforming and without CO₂ capture was also simulated and analyzed for comparison. The fuel in this case was the same as the NG feed (stream 1) shown in Table 1. In both cases, no fuel was used for SF, the turbine inlet temperature (ISO TIT) was 1250 °C, and the gas-turbine pressure ratio was 15.6. In the present computational model, the use of cooling air for the turbine was reflected in the TIT and the turbine efficiency. Some other key information is shown in Table 2 for both cases. Furthermore, some results from the analysis of these two cases are also shown in Table 3 (the first and last columns of results, the other cases shown in this table are described in the following).

Table 1: Stream data for the flowsheet in Fig. 1 (base case: TIT=1250 °C, pressure ratio=15.6, no SF)

STREAM	1	2	4	5	6	7	9	10	11	13	16	19	21c	22	24	25	26	28	29	42
H ₂ (mol%)				7.9	7.9				31.4	38.2	41.2	46.8	55.4							
CO (mol%)				0.09	0.09				10.3	3.4	0.47	0.52	0.6							
CO ₂ (mol%)	5.3		2.0	4.9	4.9	0.03	0.90	0.03	5.53	12.4	15.3	17.4	2.1	100	0.9	0.9				
N ₂ (mol%)	2.7		1.0	0.9	0.9	77.3	73.9	77.3	30.5	30.5	30.5	34.6	41.0		73.9	73.9				
O ₂ (mol%)						20.8	11.1	20.8							11.1	11.1				
CH ₄ (mol%)	83.9		30.8	33.0	33.0				0.2	0.2	0.2	0.21	0.2							
C ₂₊ (mol%)	8.1		3.0																	
H ₂ O (mol%)	0.01	100	63.3	53.3	53.3	0.90	13.3	0.90	21.8	15.0	12.0	0.14	0.2		13.3	13.3	100	100	100	100
Ar (mol%)						0.94	0.8	0.94	0.36	0.36	0.36	0.36	0.45		0.80	0.80				
Temp (°C)	4.0	436	500	440	598	8.0	1250	598	900	429	237	25.0	250	13.6	613	83	501	501	290	325
Pressure (bar)	16.7	16.7	16.1	15.6	15.1	1.01	15.4	15.1	14.2	13.4	12.2	11.5	19.4	1.02	1.02	1.02	111	26.6	4.2	114
Flow (kmol/s)	1.01	1.8	2.8	3.0	3.0	22.7	23.7	2.8	7.1	7.1	7.1	6.2	5.3	1.0	23.7	23.7	5.5	7.0	1.1	4.
MW (kg/kmol)	19.7	18.0	18.6	17.5	17.5	28.9	27.4	28.9	18.6	18.6	18.6	18.7	13.9	44.0	27.4	27.4	18.0	18.0	18.0	18.0

Table 2: Key information for the base case (TIT=1250 °C, pressure ratio=15.6, no SF) together with a conventional modern combined cycle power plant (CC). Stream numbers (#) refer to Fig. 1

	CC	Base
Natural gas LHV Input [MW] (1)	618	834
Total air flow rate GT [kg/s] (7)	656	656
Air extraction to ATR [kg/s] (8)	-	79.8
ATR inlet pressure [bar] (10)	-	15.1
ATR outlet temperature [°C] (11)	-	900
Flow rate PRE inlet [kg/s] (4)	-	52.1
Flow rate ATR inlet [kg/s] (6)+(10)	-	133
Fuel flow GT [kg/s] (21c)	17.8	73.7
Shaft power output GT [MW]	282	272
Shaft power output ST [MW]	160	159

The generator efficiency was assumed equal to 98.6%, and the auxiliary power was assumed equal to 1% of the gross electric power production. Compression of the removed CO₂ from 1 atm to 80 bar for storage (not shown in Fig. 1) was computed separately. This compression consumed some of the gross electric power: approx. 2.8% of the LHV. The net efficiency is the electric power production after subtraction of auxiliary power and power to the CO₂ compression.

The use of a constant percentage for auxiliary power can, of course, be discussed. Then, in the two cases studied here, the most complex plant has a lower efficiency, and thus, with a constant percentage for auxiliary power, turns out with less auxiliary power. On the other hand, the error made by this assumption is marginal. Moreover, the additional losses due to

the greater complexity of the reforming process are mainly caused by the pressure losses and other irreversibilities within the process. These are accounted for in the analysis.

It can be noted that the supplied NG has some exergy, due to pressure, in addition to the chemical exergy. Furthermore, the captured CO₂ represents a significant amount of chemical exergy, which can be considered as utilized exergy. This is not notified in a 1st-law-only analysis (energy analysis). The “Total irreversibility” in Table 1 is the sum of irreversibilities in all the individual units. Some of the supplied exergy is lost in the exhaust gas (stream 25). This is the main contribution to “Exergy in outflows”. The “Total lost exergy” shown in Table 3 is the sum of the total irreversibility, the exhaust exergy, the generator losses, and the auxiliary power. The “Net mechanical power” is the net mechanical power output from the thermal plant. In addition, the chemical exergy of the captured CO₂ is the “utilized” exergy of the thermal plant. Thus, the “Utilized exergy” as a fraction of the total exergy supplied (here,

chemical and thermomechanical exergy of the fuel) is the second-law efficiency (rational efficiency) of the thermal plant. In addition, some of the produced power was used for compressing the CO₂, which was determined in a separate calculation.

Subsequently, the following parametric variations were carried out:

- supplementary firing (SF) for preheating of the reformer streams,
- gas-turbine pressure ratio, and
- gas-turbine inlet temperature (TIT).

The two latter variations were made for the system presented above, and for a corresponding conventional combined cycle without reforming and without CO₂ capture. Both variations were made without SF.

Table 3: Computational results for the different cases in variation of SF. The last column shows the corresponding values for a conventional combined cycle plant without SF ("Exhaust temperature after SF" means for this case the turbine exhaust temperature).

Fuel for SF (% of GT fuel)	0	2,07	5,43	10,3	23,5	39,6	CC
Exhaust temperature after SF (C)	613	630	657	696	796	915	618
HP/MP steam temperature (C)	500	513	533	560	560	560	560
LHV of fuel (MW)	834	845	862	886	951	1030	739
Gross el. power (% of LHV)	51,0	50,9	50,8	50,7	50,1	49,6	59,0
Auxiliary power (% of LHV)	0,51	0,51	0,51	0,51	0,50	0,50	0,59
CO ₂ compression work (% of LHV)	2,83	2,82	2,82	2,81	2,76	2,72	
Net el. power (% of LHV)	47,7	47,6	47,5	47,4	46,9	46,4	58,43
Fuel chemical exergy (MW)	869	880	898	923	991	1073	770
Other exergy inflow (% fchx*)	0,85	0,85	0,85	0,85	0,85	0,85	1,26
Total irreversibility (% fchx)	46,4	46,5	46,6	46,7	47,3	47,9	40,0
Exergy in outflows (% fchx)	2,67	2,66	2,65	2,65	2,61	2,61	3,72
Generator loss and auxiliaries (% fchx)**	1,19	1,19	1,19	1,18	1,17	1,16	1,38
Total lost exergy (% fchx)	50,2	50,3	50,4	50,6	51,1	51,7	45,2
Net mechanical power (% fchx)	49,7	49,6	49,5	49,3	48,8	48,3	57,5
Chemical exergy in CO ₂ (% fchx)	2,14	2,13	2,13	2,12	2,09	2,05	
Utilized exergy (% tx***)	51,4	51,3	51,2	51,1	50,5	50,0	57,0
CO ₂ compression work (% fchx)	2,71	2,71	2,70	2,69	2,53	2,61	
Net electric power (% fchx)	45,8	45,7	45,6	45,5	45,0	44,5	56,1

* % fchx= % of NG chemical exergy

** Generator efficiency: 98.6%

*** % tx = % of total exergy inflow

The first series of simulations was carried out with a TIT of 1250 °C and a pressure ratio of 15.6, and with varying SF. The results from the analysis of these cases are shown in Table 3, together with the two cases described above. The determined efficiencies are also shown in Fig. 2.

As can be seen from Fig. 2, the efficiencies were decreasing with increasing SF. However, the negative effect of increased

combustion irreversibility was to some extent counteracted by the higher HP/MP steam temperature (see Table 3) until the maximum steam temperature was reached (560 °C, streams 26 and 28). This is indicated by the shift in curve gradient. The case with no SF showed an energy efficiency of 47.7% whereas the corresponding value for a natural-gas-fired combined cycle without CO₂ capture was 58.4%.

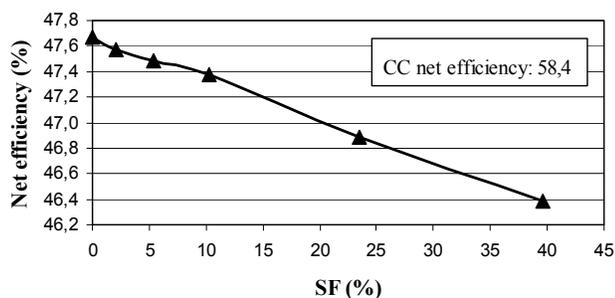


Fig. 2: Net energy efficiency (net electric power) for different cases with variation in SF (SF, as % of GT fuel).

The irreversibility breakdown for varying SF is shown in Table 3. In this table, the preheating of the ATR feed (stream 6 and 10 in Fig. 1), the heat exchangers H1 to H4, and the fuel compressor FC are reckoned under the reforming process together with the reactors PRE, ATR, HTS, and LTS. The “Other Ref.” entry in the table represents the sum of all these units, except the ATR and H1.

Fig. 3 shows those units that exhibited the significant changes in exergy losses. As seen from this figure, the largest contributors to exergy losses are the combustor and ATR units, that is, the main units for chemical reactions. It is also seen that the reduction in the (relative) contribution of these two units was less than the increased contribution from the SF.

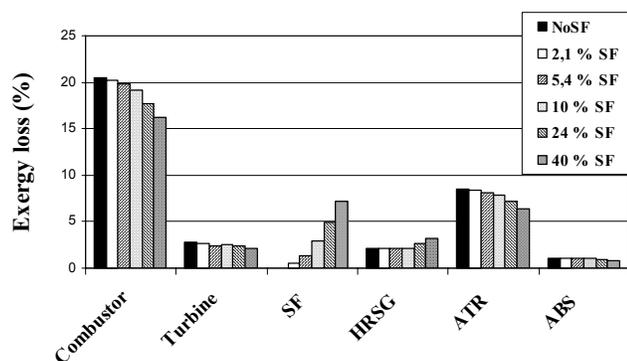


Fig. 3: Irreversibilities of selected units (% of NG chemical exergy) with increasing SF (SF, denoted as % of the GT fuel).

Furthermore, it can be noted that the single heat exchanger H1 had a considerable irreversibility, *e.g.*, greater than that of the HRSG. H1 is a boiler in which high-pressure water is heated and vaporized to saturated steam (approx. 325 °C), and heated with the high-temperature products (900°C) of the ATR. This thermodynamically “unwise” arrangement follows engineering practice and is due to the material problems that would occur in

a high-temperature gas-gas heat exchanger, in particular when H₂ is present (metal dusting). The irreversibility of this unit was 25-30% of the exergy it transferred to the steam. With improved material technology, this may be utilized *e.g.* for superheating of steam.

In Table 4, it is seen that the HRSG irreversibility has a strong increase for the fifth and sixth case. This can be related mainly to the increased temperature difference between exhaust and high-temperature steam, as seen in Table 3. As described above, the CO₂ separation was assumed to be maintained by the duty of heat exchanger H5 and the throttling of CO₂ to 1 atm. The exergy analysis showed that the available exergy in these two sources varies from 2.7 to 2.2 times the exergy required for the separation.

Table 4: Irreversibility breakdown (% of NG chemical exergy). Decarbonized NG power plant with varying amount of fuel to SF. (For abbreviations, see Fig.1; “OtherRef.” represents the units of the reformer part other than H1 and ATR).

Fuel for SF (% of GT fuel)	0	2,07	5,40	10,3	23,5	39,6	CC
AC	1,56	1,54	1,51	1,47	1,37	1,26	1,76
Combustor	20,5	20,2	19,8	19,1	17,7	16,2	28,7
Turbine	2,74	2,71	2,43	2,56	2,37	2,17	3,08
SF	0,00	0,55	1,39	2,99	4,96	7,20	
HRSG	2,12	2,13	2,14	2,16	2,64	3,24	2,83
ST	1,78	1,80	1,84	1,89	2,05	2,20	1,96
COND	1,55	1,56	1,58	1,61	1,71	1,80	1,69
ATR	8,52	8,37	8,16	7,85	7,15	6,44	
H1	2,79	2,79	2,76	2,73	2,62	2,51	
OtherRef.	3,32	3,36	3,39	3,45	3,59	3,77	
WR	0,40	0,40	0,39	0,40	0,37	0,34	
ABS	1,09	1,09	1,04	1,01	0,87	0,78	

Next, the influence of TIT and pressure ratio was investigated. The energy efficiencies for the variations of gas-turbine compressor pressure ratio for 3 different TITs are shown in Fig. 4 both for a decarbonized NG power plant and for a corresponding conventional CC plant.

The TITs (ISO definition), were 1250 °C, 1350 °C, and 1450 °C, respectively, and the pressure ratios were 15.6, 20, 25, 30, and 40. The lowest values (1250 °C, 15.6) reflect data from the GE 9 FA gas turbine. The intention of using the higher values is to look ahead in time, extrapolating the previous development in gas turbine technology. In the computational model, it is assumed that the cooling air penalty is the same in all cases, assuming that maximum surface temperatures and cooling efficiencies will increase accordingly to the increase in TIT. As expected and shown in Fig. 5, the efficiency increased with increasing TIT. Furthermore, for a conventional plant, the general tendency was that the efficiency increased with increased pressure ratio.

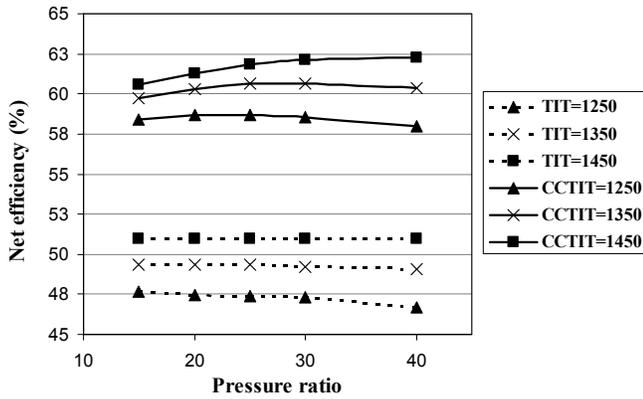


Fig. 4: Net efficiency – pressure ratio (TIT=ISO, °C)

The corresponding irreversibilities are shown in Figs. 5 and 6. Here, the units of the plant are grouped into four groups (two in the conventional CC cases). When details were inspected, it was observed that the changes in efficiency with higher pressure was a result of reduced irreversibility in the combustor and in the steam-cycle system (due to lower temperature of the exhaust inflow to the HRSG), counteracted by increased irreversibility in air compressor and turbine. In the high-pressure cases of the conventional CC, the increased irreversibility outnumbered the reduction, and therefore the upper three curves in Fig. 4 tend to flatten or turn slightly down. It is seen that the irreversibility of the steam cycle is only slightly reduced at the high-pressure ratios, and not enough to compensate for the increased irreversibility in the gas turbine. It may be that the low exhaust-gas temperature at this high pressure-ratio requires changes in the heat- and mass-flow configuration of the HRSG for optimum performance. Some further studies are required to clarify these relations.

These observations were also made for the plant with reforming and CO₂ capture. Here, the changes in irreversibility of turbine and compressor versus combustor more or less counteracted each other. However, the prominent effect in these variations was the irreversibility of the reforming process. In particular, the irreversibility of the ATR increased by approximately 1,0% (of the fuel chemical exergy) for all three TITs (from 8.5%, 8.0%, and 9.0%, respectively) as the pressure ratio was increased from 15 to 40. An important reason for this was the reduced exhaust temperature from the gas turbine. To maintain the ATR outlet temperature at 900°C, more NG was consumed in the ATR for thermal energy and thereby increasing the irreversibility.

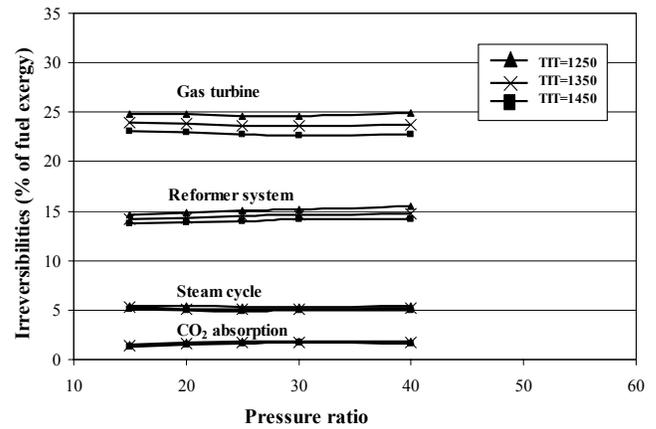


Fig 5: Irreversibilities for all units in groups (% of fuel chemical exergy) for variation in pressure-ratio and temperature (TIT). Decarbonized NG power plant with no SF.

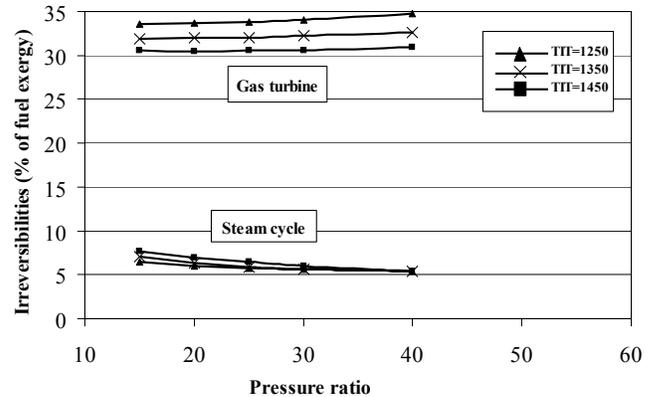


Fig 6: Irreversibilities for all units in groups (% of fuel chemical exergy), for variation in pressure-ratio and temperature (TIT). Conventional combined cycle with no CO₂ capture.

CONCLUSIONS

A power-plant concept with decarbonisation of NG by auto-thermal reforming and precombustion CO₂ removal has been investigated. An energy and exergy analysis has been performed, and the influences of some parameters have been studied. This concept has also been compared to a conventional combined-cycle process.

For the CO₂-reduced plant, with TIT of 1250°C and pressure ratio of 15.6, the net electric power was 47.7% of the LHV of the supplied NG. The exergy analysis showed that the net electric power and the chemical exergy of the captured CO₂

represented 45.8% and 2.1%, respectively, of the chemical exergy of the supplied NG. In addition, 2.7% was used for CO₂ compression to 80 bar. In the corresponding conventional combined cycle with no CO₂ capture, the net electric power was 58.4% of the LHV and 56.1% of the chemical exergy of the fuel.

It was shown that heating the reforming process by oxidation within the auto-thermal reformer is favourable to preheating the reformer feed by supplementary firing (SF) in the gas-turbine exhaust. The exhaust gas can provide enough heat for reformer preheating without SF. Moreover, an increasing amount of fuel to SF increased the total irreversibility, and hence, reduced the net output from the plant. This was due to the higher irreversibility caused by SF compared to that of the two other main reactors, the gas-turbine combustor and the auto-thermal reforming reactor.

Increasing the gas-turbine pressure ratio may improve the performance of conventional combined-cycle processes. This resulted from a reduced steam-cycle irreversibility due to reduced turbine-exhaust temperature. However, this was not the case for the process with NG reforming and CO₂ capture. It was shown that at a certain turbine inlet temperature (TIT), a higher pressure-ratio gave a slight reduction of the net efficiency. This resulted from an increased irreversibility of the reforming process, in particular that of the auto-thermal reforming reactor.

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