Alternatives for power supply to natural-gas export compressors combined with heat production evaluated with respect to exergy utilization and CO_2 emissions

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Abstract The supply of process steam in combination with power for natural gas export compressors was investigated using exergy analysis. The existing system with three 12.32 MW direct drive gas turbines each with a HRSG delivering 19.2 kg/s high-pressure steam was compared with an alternative where the gas turbines were replaced with new turbines. The exergy efficiencies were 46.7% and 48.6%, respectively, for the two cases. A second alternative with electric motors and a new CHP was investigated in three variants, all with some surplus electricity production. All variants gave higher exergy efficiencies than the other alternatives, from 51.5% to 53.6%. A third alternative with electric motors, stand-alone boilers and purchase of electricity was also analyzed, considering different origins of the electricity. This alternative gave the lowest exergy efficiencies, from 37.1% to 41.4% for different variants. In accordance with the exergy utilization, the CO_2 emissions per unit of exergy delivered were the lowest for the second alternative, while the total emissions were the highest for the third alternative. However, the domestic emissions, important in relation to international CO_2 agreements, were shown to be the lowest for the stand-alone boiler in combination with imported electricity.

Key-words: Combined heat and power, cogeneration, exergy, efficiency, CO_2 emissions, electricity

1 Introduction

The Kårstø gas processing plant at the south-western coast of Norway was built in the early 1980s to receive natural gas from the northern part of the North Sea. The plant was extended in 1993 to receive condensate from the Sleipner field and in 2000 to receive gas from the Åsgard field in the Norwegian Sea. Moderate extensions were also made in 2003 and 2005. The plant distills raw natural gas and condensate into methane-rich sales gas and ethane, propane, iso- and normal-butane, naphta and condensate. The sales gas is

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compressed and exported through subsea pipelines. The other fractions are delivered in liquid state by boat.

The Kårstø plant has a nominal capacity to handle 88 million standard cubic meter rich gas per day [1], which is one-third of the Norwegian natural gas production. This corresponds to approximately 1.2 EJ annually, which is 1.5 times the domestic Norwegian end use of energy (excluding the oil and gas sector). The dry-gas fraction, pipeline sales gas, from the oldest part of the plant (Statpipe) is compressed in three parallel compressors each powered by a Rolls-Royce Avon gas turbine (GT). The gas is further compressed by three electrically driven booster compressors. As this equipment has been in operation since 1985, one is discussing possibilities for replacement or upgrading.

The operation of a processing plant like Kårstø consumes considerable amounts of energy. The plant "feeds" on the hydrocarbon flow, and saved energy can be sold to the customers. With increasing energy prices, the economic potential for improvements is increasing. Moreover, the oil and gas industry contributes a considerable share of the Norwegian CO_2 emissions. Thus, efficiency improvements in this industry are likely to be required to comply with the obligations of the Kyoto protocol on greenhouse-gas emissions.

This study focuses on the drivers of the sales gas compressors. Three main alternatives to the existing GTs and heat-recovery steam generators (HRSGs) are studied with respect to energy and exergy utilization. The alternatives are to replace the existing turbines with new and retain the HRSGs, to electrify the compressors and build an on-site combined heat and power (CHP) plant, or to purchase electricity from outside.

The alternatives will result in different types and quantities of input and delivery: fuels, mechanical work, electric energy and steam. Hence, a common metric for comparison is needed. Exergy is regarded as such a metric, as it accounts for the "quality" of the energy. Hence, the delivered exergy as a fraction of input exergy can be used for comparing the alternatives.

Emissions of CO_2 are considered for the alternatives. The cases can be compared in terms of emissions per unit of delivered exergy. Furthermore, the different cases can lead to different consequences with respect to international agreements (e.g. Kyoto protocol) on emissions reduction. For instance, use of purchased electricity may cause emissions outside the borders of the country and hence, be assigned the CO_2 "account" of another country.

The existing plant was originally designed with respect to economical conditions as they were known at that time. Since then, the regular operation and all suggested modifications have been evaluated by economic analysis. In recent years, increasing energy prices have changed the limitations of such plants. In near future, possibilities for pricing of CO_2 (tax, tradable quotas, national limits) may cause even greater changes in the economics of the plant. The objective of the study was to take a step back, and compare the alternatives on a thermodynamic basis while economical considerations are left out.

The process will be described in Sec. 2, theory and methods for the exergy analysis in Sec. 3. Subsequently, the details of the predictions and results will be presented and discussed. The latter includes a discussion of how to account for the electric energy that has to be purchased in the third alternative. Further details of the investigations can be found in [2]. Some considerations on CO_2 emissions within or outside country borders are also given.

2 Process description

The methane-rich sales gas is compressed and delivered to the export gas pipelines. The purpose of the process studied is to provide mechanical power to the compressors and to produce high-pressure (HP) steam for the process plant. The alternatives to the existing process, GTs with HRSGs (Base Case) are as follows:
- Alternative 1: New gas turbines, remaining system unchanged.
- Alternative 2: Replace the GTs and HRSGs with electric motors and a new on-site CHP plant.
- Alternative 3: Replace the GTs with electric motors and purchase electricity, while the HRSGs are retained as directly fired steam generators.

In the existing process, each of the three compressors is powered by a Rolls-Royce Avon gas turbine. Since they were new in 1985, the turbines have been regularly maintained. The flue gas from each GT is ducted into a Foster-Wheeler HRSG with supplementary firing (SF). Only high-pressure (HP) steam is delivered. The HRSG consists of a pump, an economizer, a boiler and two superheater sections. Compressed water is fed into the economizer and also mixed into the steam between the two superheaters. The boiler has a continuous blowdown to remove impurities. The blowdown mass flow is taken care of, and a part of it is utilized as low-pressure steam. The HP steam is used for expansion and heat transfer and then returned to the HRSG.

The flowsheet for all cases is shown in Fig. 1. Air (stream 1) is compressed (streams 2-3) for the combustor where fuel (5) is supplied. Some of the air flow (4) from the compressor is ducted directly to the turbine for blade cooling. After expansion, the turbine exhaust flow (8) is ducted into the HRSG, where some more fuel (9) is burnt in supplementary duct firing (SF) before the flue gas (10) exchanges heat with the steam and is released (11) to the stack. Recirculation, stream 12, is considered for alternative 3 only (see below).

Alternative 1 is simply to replace the old turbines with new turbines. A GE LM1600 with a design load of 14.3 MW [3] is chosen for this study. Provided that the physical shape and extension of the new turbines are similar to the old ones, replacement should be relatively simple to accomplish. This could e.g. be done during a planned shutdown for maintenance. The existing Foster-Wheeler HRSGs are retained for further use. Higher

GT efficiency is expected for a new turbine. On the other hand, this may lead to a lower temperature or lower flue-gas mass flow to the HRSGs. More supplementary firing will then be required to maintain the steam production.

In alternative 2, the GTs driving the compressors are replaced by electric motors. A new CHP plant is built on the site to deliver both the electric energy to the compressor drivers and the required amount of HP steam to the process plant. In this alternative, the existing HRSGs can be retained for back-up but will not be used in the normal operational mode.

This alternative removes the direct relation between the power required by the compressor and that produced by the GT. It opens for a possibility to produce additional electricity for the booster compressors and other electric equipment. Currently, this electricity is purchased from outside. The alternative also enables continuous operation of the compressors during maintenance of the GTs.

In this study, three different variants for the CHP plant of this alternative are considered:

- 2a: Three GE LM2500+ with a design power load of 34.57 MW [3] each.
- 2b: Four GE LM2500+ of said load.
- 2c: Three GE LM2500 PE with a design load of 23.24 MW [3] each.

For all alternatives, each GT has a separate HRSG to produce steam. In alternative 2b, the four HRSGs produce the same total amount of steam as the three HRSGs in the other alternatives.

In alternative 3, the existing HRSGs have to be operated as stand-alone steam generators to produce the required steam. Hence, only streams 8 to 12 in Fig. 1 are relevant. Here, stream 8 is air taken from the atmosphere and blown by a fan. The effects of flue gas recirculation are investigated. A fraction (stream 12) of the flue gas is then recirculated from the tail of the flue gas duct and blown by a fan into the firing section. This is expected to reduce the required excess air and the stack loss without reducing the volumetric flow and temperatures in the heat exchangers.

All alternatives involve exchange of electricity with the outside and three different origins for the electric energy are considered: Natural gas-fired power plants within Norway, natural gas-fired power plants abroad and coal-fired power plants abroad.

3 Theory and method

The analyses of the overall system and the subsystems were based on the steady-state rate balances of mass, amounts of species or elements, energy and exergy [4,5]. Numerous exergy analyses of GT systems are presented in journal literature and textbooks, e.g. [4,6-7].

The species, total mass and energy balances were solved using the commercially available program PRO/II (ver. 7.1) [8]. This program provided enthalpy and entropy differences of the flows and units (subsystems) using a Soave-Redlich-Kwong equation of state with a mixing model extension. The corresponding exergy differences were then calculated from these differences and balanced in a spreadsheet. Thus, the exergy calculator of PRO/II was not used.

The thermal enthalpy was determined as the enthalpy at the actual state relative to the chosen ambient temperature and pressure (T_0, p_0),

$$h_{th} = h - h_0 = h(T, p) - h(T_0, p_0). \qquad (1)$$

The total enthalpy was determined as the sum of the thermal enthalpy and the lower heating value (LHV) of the substance. LHVs were obtained from [4]. The fuel, air and flue gas mixtures at ambient pressure were regarded as ideal mixtures and, accordingly, enthalpies were calculated as weighed sums of component enthalpies.

The flow exergy can be split into thermo-mechanical and chemical exergy: $\varepsilon = \varepsilon_{\text{tm}} + \varepsilon_{\text{ch}}$. The thermo-mechanical exergy is determined from

$$\varepsilon_{\text{tm}} = h - h_0 - T_0(s - s_0), \qquad (2)$$

where $h_0 = h(T_0, p_0)$ and $s_0 = s(T_0, p_0)$ for the relevant flow (mixture).

For a single, gaseous component present in the atmosphere, the molar chemical exergy was determined as

$$\bar{\varepsilon}_{\text{ch},i} = \bar{R}T_0 \ln(p_0 / p_i^e) = -\bar{R}T_0 \ln(x_i^e), \quad (3)$$

where the overbar denotes molar quantities, $\bar{R}$ is the universal gas constant, x_i^e is the mole fraction of the species *i* in the atmosphere and p_i^e is the corresponding partial pressure. For other species, data for chemical exergy were obtained from Kotas [4]. These data are given at a reference state of 1 atm, 25 ºC and 28% relative humidity (RH), and were corrected for deviating ambient conditions as [9]

$$\bar{\varepsilon}_{\text{ch},i} = \bar{\varepsilon}_i^{\text{ch,ref}} \frac{T_0}{T^{\text{ref}}} + \bar{h}_{\text{LHV},i}^{\text{ref}} \frac{T^{\text{ref}} - T_0}{T^{\text{ref}}} + T_0 \bar{R} \sum_{j \neq i} \nu_j \ln \frac{x_j^{\text{ref}}}{x_j^e}. \qquad (4)$$

Here, $\bar{\varepsilon}_i^{\text{ch,ref}}$ and $\bar{h}_{\text{LHV},i}^{\text{ref}}$ are the chemical exergy and LHV, respectively, of the species at the reference state of the table, T^{ref} is the reference temperature (25 ºC), x_j^{ref} is the mole fractions of the co-reactant (here: O_2) and products (here: CO_2 and H_2O) in complete combustion of the species at the reference state, while ν_j denotes the corresponding stoichiometric coefficients of these substances. The LHV varies much less with atmospheric conditions [9], and the variation was neglected.

The chemical exergy of a mixture was determined from

$$\bar{\varepsilon}_{\text{ch,mix}} = \sum_i x_i \bar{\varepsilon}_{\text{ch},i} + \bar{R}T_0 \sum_i x_i \ln x_i, \qquad (5)$$

where x_i is the actual mole fraction of a species in the mixture. The last term represents the reduced exergy due to the mixing of the components.

For presentation purposes, efficiencies and performance indicators are often used. The total energy efficiency and the exergy efficiency are two of these.

The total energy efficiency is the ratio of delivered usable energy to the energy input. For a system producing heat and mechanical energy, this can be expressed as

$$\eta_{\mathrm{tot}} = (W + Q) / H = \eta_{\mathrm{el}} + \eta_{\mathrm{Q}} \,. \tag{6}$$

Here, W is the mechanical or electric energy that is produced, $\eta_{\mathrm{el}} = W / H$ is the electric efficiency (or work efficiency for direct mechanical drive), Q is the thermal energy in delivered steam, $\eta_{\mathrm{Q}} = Q / H$ is the heat efficiency and H is the input energy, usually the lower heating value (LHV) of the fuel. The efficiency of Eq. (6) is known under a variety of names such as CHP efficiency, overall efficiency and energy utilization factor, depending on the writer and the context.

Correspondingly, the exergy efficiency is the ratio of delivered exergy to input exergy

$$\eta_{\mathrm{ex}} = (W + E_Q) / E_F \,. \tag{7}$$

Here, E_Q is the exergy of the delivered thermal energy, E_F is the input exergy, usually the fuel exergy, while the exergy of electricity or mechanical energy is equal to its energy.

Furthermore, for evaluation combined production is often compared with separate production in specified reference plants. The equivalent electric efficiency (EEE), can be expressed [10,11] as

$$\eta_{eel} = W / (H - Q / \eta_{Q,\mathrm{ref}}) \,. \tag{8}$$

Here, the energy in the fuel supplied to the CHP plant is supposed to be reduced by the fuel needed to produce the heat in a separate boiler with efficiency $\eta_{Q,\mathrm{ref}}$. The notion implied by this expression is that the heat is produced, in any case, with a certain efficiency. The remaining fuel is then attributed to electricity production, with the efficiency equal to the EEE. The rationale is that a CHP is operated according to the required heat production, while the electricity generation is more freely variable.

The relative primary energy savings (RPES) or relative fuel energy savings [10,11] is the savings achieved by combined production divided by the fuel energy that has to be used to generate the same quantities of heat and electricity in the separate reference devices,

$$\mathrm{RPES} = 1 - H / H_{\mathrm{ref}} = 1 - (\eta_{\mathrm{el}} / \eta_{\mathrm{el,ref}} - \eta_{\mathrm{Q}} / \eta_{\mathrm{Q,ref}})^{-1} \tag{9}$$

Here, $\eta_{\mathrm{el,ref}}$ is the efficiency of a defined reference plant for separate electricity production, and H_{ref} is the amount of fuel required to produce the same electricity and heat separately in the reference plants.

Both the EEE and the RPES are based on 1st law (energy) considerations. The relative avoided irreversibility, RAI, is an indicator [12] similar to RPES but based on the 2nd law. It is defined

$$\mathrm{RAI} = 1 - E_{\mathrm{F}} / E_{\mathrm{F,ref}} = 1 - (\eta_{\mathrm{el}} / \eta_{\mathrm{el,ref}} - \alpha_{\mathrm{Q}}\eta_{\mathrm{Q}} / (\alpha_{\mathrm{Q,ref}}\eta_{\mathrm{Q,ref}}))^{-1}, \tag{10}$$

which is the irreversibility (exergy destruction) that is avoided by choosing the CHP instead of separate production of heat and work. Here, $\alpha_{\mathrm{Q}} = E_{\mathrm{Q}} / Q$ is the exergy to energy ratio of the heat, $\alpha_{\mathrm{Q,ref}}$ is this ratio for a reference plant (which may be a boiler or a heat pump) and $E_{\mathrm{F,ref}}$ is the fuel exergy required to produce the same electricity and heat separately in the reference plants.

4 Present predictions

The existing process and alternative 1 are simulated in one case each, labeled Base Case and Case1, respectively. Alternative 2 is simulated in three variants, Cases 2a-2c, and alternative 3 in five variants, Cases 3a-3e.

The following operational conditions were assumed in the simulations:

- The power to each of the three compressors was 12.32 MW.
- Steam was delivered at 59 bar (abs) and 420 °C at a mass flow rate of 69.12 tonnes per hour (t/h), i.e. 19.20 kg/s, from each of the three HRSGs.
- Return water was received at 2.116 bar and 122.0 °C
- The atmospheric air had temperature 15°C and pressure 1.013 bar (1 atm).
- The air composition was assumed to be 77.09% N_2, 20.69% O_2, 0.93% Ar, 0.03% CO_2 and 1.26% H_2O. At 15 °C and 1 atm, this corresponds to a relative humidity of 75%. From meteorological data for a nearby location (Haugesund Airport) [13], this appeared to be a representative atmospheric state in the summer.
- The fuel for gas turbines was available at 31.0 bar and 40 °C and consisted of 94.85% methane, 2.54% ethane, 0.73% propane, 0.28% butane, 0.10% pentane, 0.03% hexane, 0.93% N_2 and 0.54% CO_2.
- The fuel for supplementary firing in the HRSGs was available at 4.5 bar and 28 °C and consisted of 92.96% methane, 4.79% ethane, 0.73% propane, 0.07% butane, small amounts of pentane and hexane, 0.99% N_2 and 0.46% CO_2.
- The minimum temperature difference between flue gas and water/steam in the heat exchangers was 20 °C.
- Power required for water pumping and fans for air and flue gas recirculation (when relevant) was calculated by the model, whereas other auxiliary power was neglected.

- All units were assumed adiabatic.
- Possible leakages were neglected.
- Pressure losses in the combustion chambers were neglected, except for throttling of fuel in the nozzles. Pressure losses were included in the models of all other units. The air intake filter was modeled by assuming a pressure loss of 0.010 bar before the compressor inflow.

For the two flows of fuel, it was assumed that the amounts can be changed without changes in the composition or state. Surplus fuel compared to the base case can be absorbed by the processing plant, and additional fuel can be drained from the plant.

The GT and the HRSG were simulated in PRO/II [8]. The parameters of the GTs were adapted to reproduce data obtained from [3] and the corresponding models in the commercial simulator GTPro [14]. For the Base Case and Case 1, the GTs were first modeled with their design power loads of, respectively, 13.34 MW [14] and 14.25 MW [3]. Then, the power was scaled down to the desired value of 12.32 MW by reducing the fuel flow rate and increasing the air flow rate while maintaining the total flow rate, the pressure ratio and the adiabatic efficiencies. An alternative approach of reducing the fuel flow rate while maintaining the air flow rate was also tested with virtually identical results.

For alternative 3, only the steam generator (SG) was used (streams 8-11), and all the heat was provided by the fuel burnt therein. This configuration was simulated with recirculated mass flows (stream 12) equal to (Case 3a:) 0 kg/s, (3b:) 10 kg/s, (3c:) 20 kg/s, (3d:) 35.934 kg/s, (3e:) 50 kg/s. In Case 3d, 50% of the flue gas was recirculated.

In all cases but Case 2b, 18.30 kg/s water at 2.116 bar, 122.0 °C was pumped to 69.92 bar and heated in the economizer, boiler and two superheaters. Blowdown in the boiler amounted to 0.181 kg/s water, while 1.08 kg/s of water was added between the superheaters. The resulting flow was 19.20 kg/s of steam. For simplicity, it was assumed that the fraction of the blowdown utilized as low-pressure (LP) steam was 50%. The state of the LP steam was 7 bar and 200 °C. The need for make-up water is small and its treatment and preheating was neglected in this study.

In Case 2b the steam was produced in four units instead of three. Therefore all water/steam mass flow rates for each unit were three-fourth of those given above, while temperature and pressure were maintained for all streams.

For alternative 3, the three different assumptions made for the origin of the purchased electricity were as follows:
- Natural-gas (NG) fired power plants in the same region; net efficiency 58%.
- Imported electricity from NG-fired power plants abroad, net efficiency 54%.
- Imported electricity from coal-fired power plant abroad, net efficiency 40%.

Here, net efficiency is the ratio of electric energy used at the Kårstø plant to the LHV of the fuel consumed in the power plant. The difference between regional and abroad NG net efficiency reflects the fact that the local plant will be a new plant (the first Norwegian large-scale thermal power plant is being built at Kårstø next to the processing plant), while imported electricity will be a mixture of new and old plants. Imported electricity also means larger transport losses (i.e. reduced net efficiency). The CO_2 emissions were assumed to 204 kg per MWh of LHV for natural gas and to 334 kg of CO_2 per MWh of LHV for coal.

The electricity for the water pump of the HRSG of the Base Case and alternative 1 is produced outside the system. For alternative 2, electricity for the pumps is provided from the CHP and for alternative 3, electricity for pumps and fans is purchased together with the electricity for the NG export compressors.

When calculating the CHP performance indicators of Eqs. (8) to (10), reference plants for separate production have to be specified. Here, reference efficiencies of 0.55 for electricity and 0.90 for heat were assumed [11]. The reference exergy to energy ratio of thermal energy was set to 0.28 [12].

5 Results

5.1 Fuel, mass flow rates, temperature and pressure

For the GT fuel, an LHV of 48.303 MJ/kg and an exergy of 50.680 MJ/kg were calculated. This exergy figure includes a reduction of the chemical exergy by 0.34% due to the chosen ambient conditions, Eq. (4), and a thermomechanical exergy of 0.474 MJ/kg, Eq.(2). The CO_2 emissions were 2.70 kg per kg of fuel. For the fuel used in the boiler, the LHV was 48.332 MJ/kg and the exergy 50.445 MJ/kg, including 0.208 MJ/kg of thermomechanical exergy. The CO_2 emissions were 2.71 kg per kg of the latter fuel.

The mass flow rates, temperatures and pressures for the Base Case and Cases 1 and 2a are shown in Table 1 and for Cases 3a (no recirculation) and 3d (50 % flue gas recirculation) in Table 2. In Cases 3a-3e the pressures in the SG were similar to those of the HRSG and were provided by fans for inlet air and recirculated flue gas. The flow rates, temperatures and pressures of the water/steam process are shown in Table 3. These values are the same for all cases except that the steam mass flow rates in Case 2b (four units) are three-fourth of those given in the table. Some key results for all cases are shown in Table 4.

The increase in turbine inlet temperature from the 20-25 year old existing GT (Base Case) to the modern alternatives is a result of the technological development to increase efficiency and reduce irreversibilites in GTs (see below). Correspondingly, the air-to-fuel mass ratio is reduced.

Regarding Case 1, it can be noted that the temperature of the flue gas after supplementary firing (stream 10) had a very high temperature. In fact, this exceeded the expected maximum temperature of the existing Foster-Wheeler HRSGs. In reality, this means either that the HRSGs have to be replaced together with the GT or that they have to be operated at a lower capacity (i.e. less SF and less steam) if this alternative is chosen.

In alternative 3, the CO_2 content of the emitted flue gas increased from 3.15% (Case 3a) to 5.73% (3d) and 8.98% (3e), while the O_2 content decreased from 13.94% (3a) to 8.36% (3d) and 1.32% (3d). This compares to 3.76% CO_2 and 12.6% O_2 content in the flue gas of the Base Case.

5.2 Energy, exergy and CO_2 emissions

Results for energy and exergy are shown for the Base Case and alternatives 1 and 2 in Table 5. The power delivered to the compressors and the rate of heat transferred to steam were fixed quantities that were common for all cases. The exergy increase in water/steam from pumped return water (Streams S2 and S3) to delivered steam (Stream S10) was 0.46 times the corresponding increase in enthalpy. This is the exergy to energy ratio of the boiler.

When the direct drive cases (Base Case and Case 1) were simulated with design power load, the 1st law GT efficiencies were 28.5% and 36.6%, respectively. Hence, the deviation between design and actual load caused a reduction in efficiency. In alternative 2, the GTs are separated from the NG compressors, and the surplus power can be used for other purposes.

It can be noted from Table 4 that the effluent flue gas had a higher temperature in Cases 2a and 2b. This was caused by the pinch-point of the steam generator, which in these cases was found at or near the saturated water state (evaporator inlet). In the other cases, the limiting temperature was the return water temperature or close to this. This caused a larger stack loss and lower energy efficiency of the HRSG for Cases 2a and 2b compared to Case 2c. The associated exergy losses were relatively lower, as the thermal energy was lost at a moderate temperature. The underlying reason for this loss was the choice of a single pressure level of the produced steam. In a dual-pressure boiler, the lower pressure part could have utilized more heat from the flue gas.

In Case 2b, with four GT/HRSG units instead of three, the amount of supplementary firing (SF) was reduced. Hence, a larger fraction of the heat to steam was supplied by the turbine flue gas and, consequently, the irreversibility of the HRSG was lower. In spite of the higher stack losses (see above), the exergy efficiency of the HRSG was maintained.

Table 6 shows results for alternative 3. The main observation was that the exergy efficiency of the stand-alone steam generator (SG) was considerably lower than for the HRSG of the CHPs. It was also seen that, as expected, increased recirculation reduced the stack loss and, consequently, the amount of fuel required to provide the process heat.

The combustors were the main contributors to irreversibility. In the Base Case, the sum of the irreversibilities of the two combustors (GT and SF) was 34 % of the fuel exergy. In alternatives 1 and 2, this was reduced to approximately 29%, while the irreversibility of the combustion chamber was from 47% to 43% of the fuel exergy in Cases 3a to 3e. The evaporator destructed from 2.3% (Case 2b) to 8% (Case 1) of the fuel exergy, while the emitted flue gas caused a loss of approximately 5% in all cases. Each of the other units had only small contributions to the irreversibility. These results are consistent with previous results, e.g. [4,6-7].

The mass flow rates of CO_2 emissions are shown for each case in Tables 5 and 6. These are based on the amounts and compositions of the fuels. Hence, the CO_2 from combustion air was not included.

5.3 Quantities and indicators for comparison of the cases

The exergetic efficiencies gives a straightforward comparison of the cases, as in Tables 5 and 6. This showed that both the HRSG and the overall plant have higher efficiencies than the stand alone boilers of alternative 3.

As the three alternative configurations involve different types of energy exchange with the surroundings, some care has to be taken in the comparison. In this instance, the principles behind the indicators referred in Eqs. (8) to (10) can be helpful. These are based on a comparison with separate production of work (or electricity) and thermal energy. The notion behind the equivalent electric efficiency (EEE), Eq. (8), is that thermal energy has to be produced close to the user. Thus, based on a specified reference boiler, a certain amount of fuel is assigned to the thermal energy production in a CHP. The remaining part of the actual fuel consumption is then assigned to the production of work or electricity. Hence, if the EEE is larger than the typical efficiency of a power plant, this shows that the CHP is beneficial to separate production.

The underlying principle of the EEE can be extended to the present cases of localized work and thermal energy production in combination with electricity exchange. It can be assumed that purchased electricity is produced in a powerplant with a certain fuel and efficiency. Similarly, surplus electricity can be associated with the amount of fuel that would have been used in the separate plant to produce this amount of electricity. With these corrections for surplus or purchased electricity, an “equivalent” amount of fuel can be associated to the desired delivery of work for the NG compressors and heat to the process steam. This amount is shown in Fig. 2 for all cases, each with the three options for separate electricity production (NG-fired in Norway, NG-fired abroad and coal-fired abroad).

For the CHP cases, the EEE, RPES and RAI, Eqs. (8) to (10), are shown in Fig. 3. In Cases 2a-2c, the NG compressor work and surplus electricity was added on an equal basis into the “electric efficiency”, as is customary in the legislation of countries where such indicators are used [11]. In the Base Case and Case 1, the small amount of electricity was subtracted from the NG compressor work. The EEE assigns the benefit of combined production to the electricity production, while the RPES puts the focus on the total fuel savings. Both these indicators are based on 1st law evaluation of heat and work. The RAI is similar to RPES but is based on a 2nd law evaluation.

The CO_2 emissions can be treated similar to the energy above. Figure 4 shows the total amount of CO_2 emitted locally and remotely per unit of exergy in the work, steam and electricity delivered from the plant. The amount of CO_2 emissions that could be assigned to the desired delivery of work for the NG compressors and the process steam is shown in Fig. 5. Here, the explanations are similar to those of Fig. 2. A particular comment is required for the negative amount shown in Case 2b. This means that a coal-fired power station producing the surplus electricity of Case 2b will emit more CO_2 due to this production than the total (local) emissions of Case 2b. Therefore, the emissions attributed to the primary delivery of Case 2b were negative. In Fig. 5, also the CO_2 emissions for the plant are shown, that is, the sum of local emissions.

When electricity is exchanged with other countries, the question of national and remote emissions will rise. This primarily affects alternative 3. It appears from Fig. 5 that the total emissions were up to twice as large as the on-site emissions. When the electricity was assumed to be purchased from a coal powerplant abroad, this alternative had the largest emissions. However, only half of the emissions occurred within Norway and this alternative had the lowest domestic emissions.

The exergy efficiencies of the HRSGs (Table 5) were quite similar. For the Base Case it was seen to be slightly lower than Cases 2a, 2b and 2c. Case1 gave a lower value due to the higher maximum temperature and had, consequently, higher irreversibility due to larger temperature differences. The overall exergy efficiency was the highest for Cases 2a and 2b, while that of Case 2c was slightly lower. The figures for alternative 3 were considerably lower and typical of boilers with this quality of steam.

Also when the fuel for surplus electricity production was separated, the fuel consumption (Fig. 2) was the lowest for the Cases 2a-2c. Although there were differences among the cases depending on the assumed origin of separate electric production, all cases of alternative 2 required less fuel for the primary deliveries (work and steam) than any of the other cases. Correspondingly, alternative 3 required the largest amount of fuel. Similar results were seen from the RPES figures, which were positive for all the CHP cases and the largest for alternative 2. However, the indicators EEE and RAI gave differing results, as alternative 1 got the highest values for both. One reason for this may be that this alternative had a large fraction of the total fuel used for SF, and the steam accounted for a large fraction of the total energy and exergy delivery. This case can be regarded as a boiler with some work production. Such systems are known [12] to give marginal production of work (or electricity) at a high efficiency (EEE) and also a higher RAI.

5.4 Uncertainties, errors and influence of specific choices in the analyses

The least accurate calculations appeared to be for the combustors. Figures for LHV of light hydrocarbons have an uncertainty of 0.05% when the best available data are used [9]. All mass flows rates were either specified values or simple sums of such quantities. However, the elemental flow rates into and out of the GT combustor showed a deviation of 0.1% for carbon and up to 0.05% for hydrogen. For the SF combustors, the deviations were up to 0.05% and 0.1%, respectively, for carbon and hydrogen. The energy balances showed deviations between inflow and outflow of up to 0.5% for the combustors. Hence, this indicates the errors of enthalpy calculations. It appeared that the "convergence criteria" that can be specified in the program did not affect the accuracy of the elemental and energy balances of the calculations.

For the units without reactions, the elemental balances were satisfied, while the relative deviations of the energy balances were 10^{-5} or less. In the compressor and turbine units of the program, the work is calculated from a difference between inflow and outflow enthalpies. Hence, for these units the energy balance is satisfied, while the errors have to be estimated from the enthalpies.

The accuracy of the exergy and irreversibility calculations are harder to estimate than that of the energy balances. The chemical exergy of methane can be calculated with an uncertainty of 0.08% [9]. The uncertainty of the entropy part of thermomechanical exergy due to deviations in temperature is (T_0 / T) times that of the enthalpy part and, hence, less or equal for the cases investigated. The uncertainty due to the deviations in the elemental balance should be of the same order.

Heating and pumping of make-up water to mix with the return water was neglected in the computations. The error of this assumption is assumed to be small. First, the amount is a small fraction of the return water. The steam is used for heat transfer and not as an input substance to the process. Second, the heat required for the make-up water is at a low temperature, and heat not utilizable otherwise can be used.

The choices of gas turbine in alternatives 1 and 2 can affect the performance of the investigated system. This is actually seen in the differences between Cases 2a and 2c. However, the differences observed are modest. Performance of GTs relies to a large extent on the turbine inlet temperatures, and these temperatures are similar for all modern GTs.

6 Discussion

The investigated cases had different combinations of thermal, work and electric deliveries. Therefore, exergy was used as a common metric for the utilization of the fuel. The differences seen between the alternatives were significant, that is, larger than the estimated errors of the calculations. An exception is Cases 2a and 2b, which should be regarded at equal level in exergy utilization.

The direct drive of the NG compressors leads to non-optimal operation of the GT. Moreover, the number of GTs available that fits into the requirements of the case is limited and the GT chosen in alternative 1 had a lower efficiency than those of alternative 2. The possibility of producing surplus electricity allows more optimal operation and a greater flexibility of choice. The benefits of direct drive, that is, to avoid conversion of work to electric energy and then back to work, seems not to compensate for the drawbacks.

The stand-alone boilers had lower exergy efficiencies than the CHP plants, and also lower than the HRSGs of the CHP plants. The reason for this is that the HRSGs utilize the medium-temperature thermal energy from the GT. As expected, recirculation of flue gas reduced the required amount of fuel for the boiler, and hence increased the energy and exergy efficiencies of it.

An energy analysis does not consider different “qualities” of the energy flows. As work is more “demanding” both thermodynamically and technically, the energy efficiencies can not be used for comparison of these cases. In the present analyses, both the thermal energy delivery and the work delivery were held constant. Hence, the fuel consumption and the exchange of electricity were the varying quantities. The amount of fuel for the work and steam, Fig. 2, showed mainly the same picture as the exergy efficiencies: Alternative 2 is to be preferred, and alternative 3 had the poorest performance.

The indicators shown in Fig. 3, RPES, EEE and RAI, gave a slightly different picture. These indicators are defined to compare CHP plants with separate production of the same amounts of work (or electricity) and thermal energy (steam). Before discussing these results, two points should be made: First, the ratios of work and electricity production to thermal production were quite different in the five CHP cases. In the Base Case and Case 1, the thermal energy delivery was 4.4 times the work delivery while in Case 2b, this ratio was 1.2. Second, the RPES and EEE are based on the 1st law, while the RAI is based on both 1st and 2nd law considerations.

The primary result of Fig. 3 is that for all cases, RPES and RAI were positive and EEE was larger than the reference electric efficiency. This means that all the CHPs have a benefit compared with separate production. Similar to the exergy efficiency and the net fuel for work and steam, the RPES showed the largest values for alternative 2. However, the EEE and RAI both gave larger values for alternative 1. Furthermore, while the exergy efficiency and fuel amount in Fig. 2 favored Cases 2 and 2b to Case 2c, the EEE and RAI favored Case 2c to Cases 2a and 2b. The results of EEE can be explained by the varying ratio of work and electricity to thermal energy. This indicator assigns all the benefits of a CHP to the work (or electricity) produced. The relative amount of thermal energy increases from Cases 2a to 2c, and to Case 1. Hence, when the amount of work (electricity) to share the benefit is reduced, the value of EEE will increase. Seen the other way, the limiting case of increasing electricity production is a separate power plant, which has an efficiency equal to the reference efficiency.

Also the higher RAI for Case 1 can be explained by the high fraction of thermal energy of the total delivery for this case and by a relatively high exergy to thermal energy ratio for the steam delivery of this system. As the RAI compares the CHP with a reference production of steam with lower exergy to energy ratio, a large fraction of thermal energy delivery is favored by this indicator as well. Hence, the better results of Cases 1 and 2c.

The surplus electric energy divided by the fuel that is additionally used compared with direct drive is another quantity that can be calculated in order to evaluate the efficiency of electricity production. That is,

$$E_{\mathrm{el},2}/(H_2 - H_1), \tag{11}$$

where H_2 and H_1 are the rates of fuel energy consumed in alternatives 2 and 1, respectively, and $E_{\mathrm{el},2}$ is the rate of surplus electric energy delivered by alternative 2. This quantity was 67.9%, 63.6% and 69.0%, respectively, for Cases 2a, 2b and 2c. This can be regarded as the net efficiency of the electricity production as compared with Case 1. The corresponding results for comparison with the Base Case gave 74.8%, 67.4% and 86.7%, respectively, for Cases 2a, 2b and 2c. These results are in line with the EEE calculations above.

The choices and assumptions of alternative electric supply require some words of discussion: A natural gas processing plant is a typical baseload electricity customer. Although the production, and hence the consumption, varies from hour to hour, the variations are small compared both to the total load and compared to consumption variations in e.g. households, commerce or transportation. Hence, it is reasonable to assign the external electric production to plants with relatively high efficiencies. The domestic NG fired plant and the coal fired plant abroad serves as minimum and maximum limiting cases for fuel consumption and CO_2 emissions. Although there is being built a new NG powerplant next to the gas processing plant, this electricity is requested by other users as well. Driving the three NG compressors of this study would take one-tenth of the capacity of the new plant.

Both Fig. 4 (CO_2 per unit of exergy delivery) and Fig. 5 (CO_2 emissions for assigned to work and steam) showed the lowest levels for alternative 2 and the highest for alternative 3. Hence, when focusing on total CO_2 emissions, alternative 2 should be preferred. There is, however, one quantity favoring alternative 3: If the purchased electricity was imported from another country, this alternative would give the lowest contribution to Norway's emissions. Technically, this seems to be a weird consideration. However, politically, it may make sense. Norway has to reduce the emissions considerably to meet its Kyoto obligations without buying quotas abroad. At the time of publication, agreements of quota trade across borders are not ready. Such treaties may include statements of how much each country has to "clean at home". Importing electricity is then a means of transferring CO_2 emissions to the account of someone else. Keeping in mind that the compressors are delivering gas to other countries, the system of this study clearly shows the international character of CO_2 emissions.

7 Conclusions

The existing CHP plant for the Statpipe natural gas export compressor drivers and steam production was analyzed by an exergy analysis. Alternatives to the present plant were analyzed as well: Alternative 1, a new GT with the old HRSGs remaining; alternative 2, electric motors and a new CHP with surplus electricity delivery, and alternative 3, electric motors with purchased electricity and steam generated in the HRSGs operated as stand-alone boilers.

The comparison between the three alternatives showed that alternative 2, electrification and a new CHP plant, had the highest exergy efficiency. Although different assumptions within each alternative gave different results, all three cases of this alternative gave significantly higher exergy efficiency, from 51.5% to 53.6%, than the direct-drive alternative. The new GT (alternative 1) gave better performance, 48.6% exergy efficiency, than the existing Base Case (46.7%). All cases of the stand-alone boiler (alternative 3) gave lower exergy efficiencies, from 37.1% to 41.4%.

Alternative 1, new GTs, appeared to require new HRSGs as well or, alternatively, it had to be operated at a lower capacity. This was due to the high temperature after supplementary firing required to produce the specified amount of high-pressure steam. In Cases 2a and 2b, it appeared that the thermal energy of the flue gas could not be fully utilized in the HRSGs. This was due to the high pressure and high temperature of the delivered steam, and steam production at two pressure levels could have increased the utilization.

Different quantities defined for comparing CHP with separate production showed that all CHP cases, including the present, were favorable compared to separate production.

The CO_2 emissions per unit of exergy delivered were lower for alternative 2 than for alternative 1, and lower for all CHP cases than for alternative 3, the stand alone boilers. This result was obtained regardless of the choices for external electricity production. However, the local – and hence domestic – emissions were the lowest when electricity for the electric motor drive of the NG export compressors was assumed to be imported from abroad.

In summary, a new CHP plant with electric drive of the compressors and surplus electricity production clearly showed to be the best thermodynamic solution. This alternative also showed the lowest overall CO_2 emissions.

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Tables

Table 1 Mass flow rate, temperature and pressure in the flows of the GT and HRSG models

	Base Case			Case 1			Case 2a		
Flow No.	Mass (kg/s)	Temp. (ºC)	Press. (bar)	Mass (kg/s)	Temp. (ºC)	Press. (bar)	Mass (kg/s)	Temp. (ºC)	Press. (bar)
1	70.33	15.0	1.003	46.59	15.0	1.003	87.59	15.0	1.003
2	70.33	291.0	8.369	46.59	443.0	20.261	87.59	458.4	22.969
3	65.97	291.0	8.369	43.70	443.0	20.261	82.16	458.4	22.969
4	4.35	291.0	8.369	2.89	443.0	20.261	5.43	458.4	22.969
5	0.909	40.0	31.013	0.717	40.0	31.013	1.677	40.0	31.013
6	66.87	847.7	8.034	44.42	1272.5	20.260	83.84	1217.9	22.969
7	71.23	816.2	8.034	47.31	1037.2	20.260	89.27	1176.0	22.969
8	71.23	427.1	1.046	47.31	448.7	1.046	89.27	486.5	1.046
9	0.633	28.0	4.513	0.768	28.0	4.513	0.433	28.0	4.513
10	71.87	779.9	1.046	48.08	1055.5	1.046	89.70	676.8	1.046
11	71.87	166.3	1.018	48.08	144.7	1.018	89.70	162.2	1.018

Table 2 Mass flow rates and temperature at the flue-gas side of the steam generator of alternative 3

	Case 3a (no recirc)		Case 3d (50% recirc.)	
Flow No.	Mass (kg/s)	Temp. (ºC)	Mass (kg/s)	Temp. (ºC)
8	70.58	15.0	34.75	15.0
9	1.284	28.0	1.186	28.0
10	71.86	788.7	71.87	769.9
11	71.86	145.1	35.93	146.0
12	--	--	35.93	146.0

Table 3 Mass flow rate, temperature and pressure in flows of the steam side of the HRSG

Flow No.	Mass (kg/s)	Temp. (ºC)	Press. (bar)
S1	19.38	122	2.116
S2	18.30	124.7	69.92
S3	1.08	124.7	66.77
S4	18.30	243.0	67.88
S5	18.12	283.7	67.88
S6	0.181	283.7	67.88
S7	18.12	392.0	66.58
S8	19.20	392.0	66.58
S9	19.20	424.4	65.27
S10	19.20	420.0	59.00

Table 4 Key data for all cases, figures for one export compressor (one-third of total plant).

Case	Base	1	2a	2b	2c
Fuel, GT (kg/s)	0.909	0.717	1.677	2.236	1.285
Pressure ratio	8.344	20.20	22.90	22.90	19.50
Air-to-fuel mass ratio, GT	77.33	65.00	52.23	52.23	52.58
Turbine inlet temperature (ºC)	847.7	1072.5	1217.9	1217.9	1195.6
Turbine exit temp. (ºC)	427.1	448.7	486.5	486.5	524.9
Fuel, boiler (kg/s)	0.633	0.768	0.433	0.281	0.485
Stack temperature (ºC)	140.5	144.6	162.2	194.9	144.7

Case	3a	3b	3c	3d	3e
Fuel, boiler (kg/s)	1.284	1.257	1.229	1.186	1.148
Stack temperature (ºC)	145.1	145.4	145.4	146.0	147.7

Table 5: Results of energy and exergy and CO_2 emissions for cases with gas turbines, figures for one export compressor (one-third of total plant).

Case	Base	1	2a	2b	2c
Rate of fuel LHV, GT (MW)	43.93	34.63	81.01	108.02	62.08
Rate of fuel exergy, GT (MW)	46.09	36.33	84.99	113.32	65.13
Rate of work, GT (MW)	12.32	12.32	34.6	46.1	23.2
Efficiency of the GT (1st law) (%)	28.0	35.6	42.7	42.7	37.4
Irreversibility rate of GT (MW)	20.4	14.2	30.1	39.9	24.3
Electric power produced (MW)	--	--	33.5	44.7	22.6
Surplus electric power (to outside) (MW)	--	--	20.5	31.7	9.5
Electric power from outside (MW)	0.33	0.33	--	--	--
Rate of thermal energy to HRSG (MW)	35.7	25.6	54.1	72.1	44.7
Rate of fuel LHV, SF (MW)	30.61	37.10	20.95	13.57	23.44
Fraction of total fuel energy used for SF (%)	41.1	51.7	20.5	11.2	27.4
Rate of thermomech. exergy to HRSG (MW)	13.4	9.8	21.8	29.0	18.6
Rate of fuel exergy, SF (MW)	31.94	38.72	21.86	14.16	24.46
Rate of thermal energy lost with flue gas (MW)	17.2	14.2	24.5	35.2	18.4
Rate of exergy lost with flue gas (MW)	3.7	3.5	5.6	8.5	4.2
Irreversibility rate of HRSG (MW)	17.7	21.1	14.0	10.7	14.9
Energy efficiency of HRSG (%)	79.2	83.8	70.8	61.4	77.1
Exergy efficiency of HRSG (%)	53.2	49.7	55.3	55.9	56.0
Total energy delivered (MW)	65.1	65.1	85.7	96.8	74.7
Total exergy delivered (MW)	36.6	36.6	57.1	68.3	46.1
Overall energy efficiency (%)	87.0	90.4	84.0	79.6	87.3
Overall exergy efficiency (%)	46.7	48.6	53.5	53.6	51.5
CO_2 emissions from plant (kg/s)	4.17	4.01	5.70	6.80	4.78

The delivery comprises work to the NG export compressor (12.32 MW), process steam (52.8 MW energy, 24.3 MW exergy) and surplus electric power.

Table 6: Results of energy and exergy and CO_2 emissions for cases of alternative 3, figures for one export compressor (one-third of total plant).

Case	3a	3b	3c	3d	3e
Electric power from outside (MW)	13.46	13.45	13.45	13.45	13.45
Rate of fuel LHV (MW)	62.05	60.75	59.42	57.33	55.48
Rate of fuel exergy (MW)	64.76	63.40	62.01	59.83	57.91
Rate of thermal energy lost with flue gas (MW)	16.1	14.7	13.3	11.1	9.1
Rate of exergy lost with flue gas (MW)	3.3	3.1	2.9	2.8	2.8
Irreversibility rate of SG (MW)	38.3	37.1	36.0	34.0	32.0
Energy efficiency of SG (%)	84.1	85.9	87.8	91.0	93.9
Exergy efficiency of SG (%)	37.1	37.9	38.7	40.1	41.4
CO_2 emissions from plant (kg/s)	3.47	3.40	3.33	3.21	3.11

The delivery comprises work to the NG export compressor (12.32 MW) and process steam (52.8 MW energy, 24.3 MW exergy).

Figures

Fig.1 Flowsheet of gas turbine and heat-recovery steam generator

Fig.2 Net rate of fuel LHV (MW) consumed (on and off-site) to produce work for natural-gas compressors and process steam. Fuel for surplus electricity is not included. Results for three alternatives for separate electricity production.

Fig.3 Equivalent electric efficiency (EEE), relative primary energy savings (RPES), and relative avoided irreversibility (RAI) of the CHP alternatives.

Fig. 4 Total on-site CO_2 emissions per unit of exergy delivered by the plant (kg/MWh).

Fig. 5 Net rate of CO_2 emissions (kg/s) (on- and off-site) to produce work for natural-gas compressors and process steam. CO_2 from fuel for surplus electricity is not included. Results for three alternatives for separate electricity production. Sum of local CO_2 emissions (kg/s) from the plant.

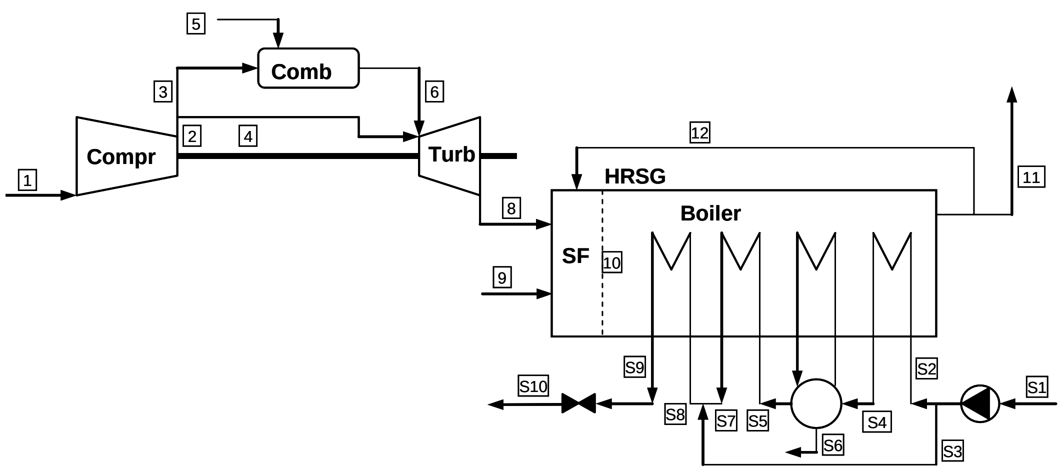
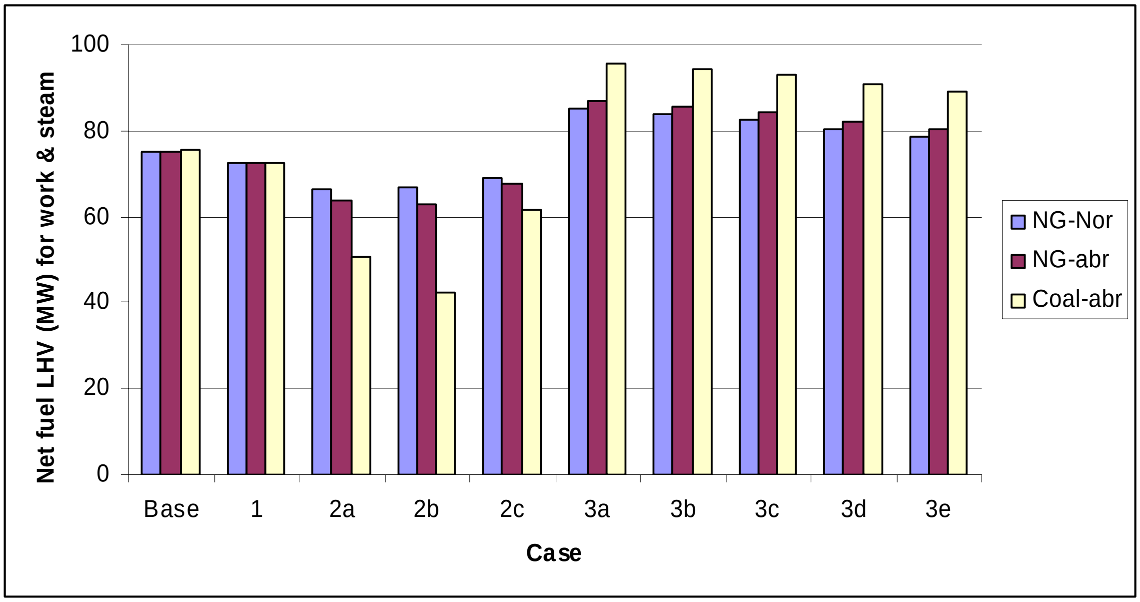
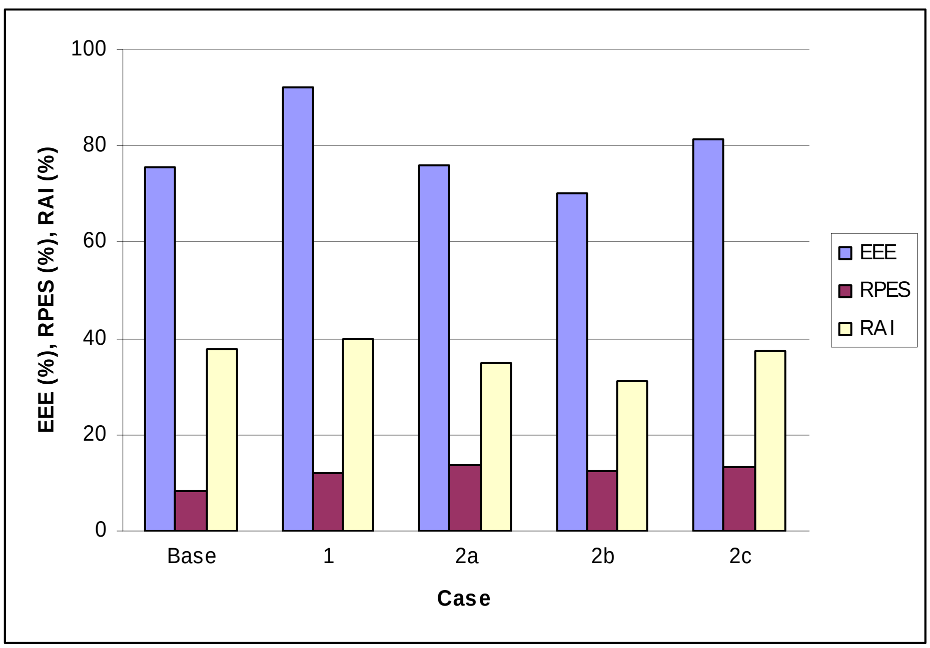


Fig.1



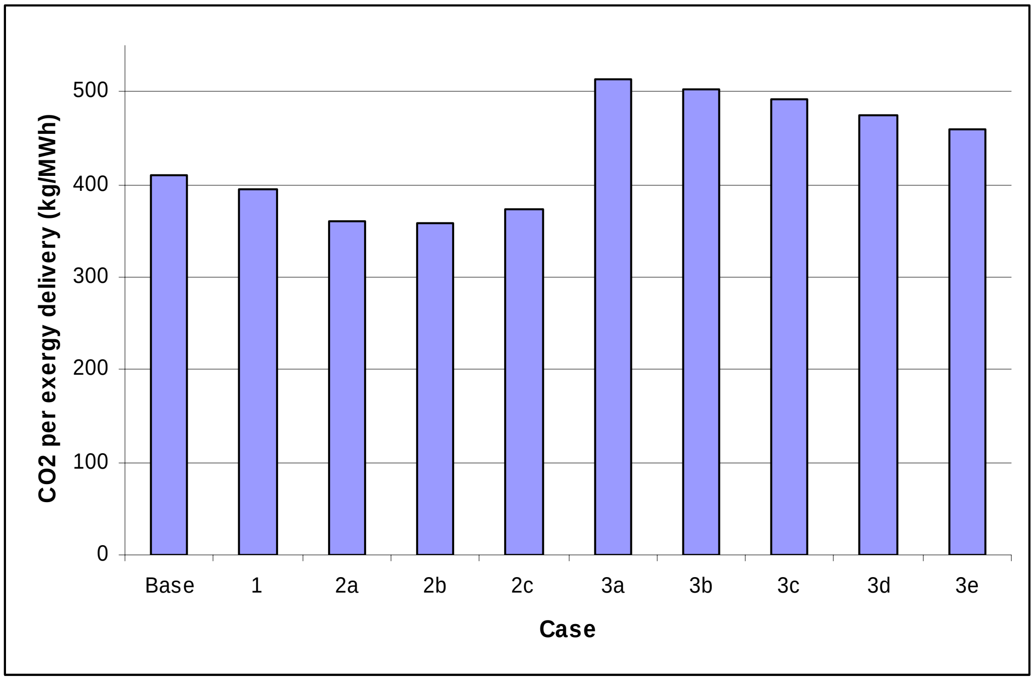
100
80
60
40
20
0
Net fuel LHV (MW) for work & steam
Base
1
2a
2b
2c
3a
3b
3c
3d
3e
Case
NG-Nor
NG-abr
Coal-abr

Fig 2



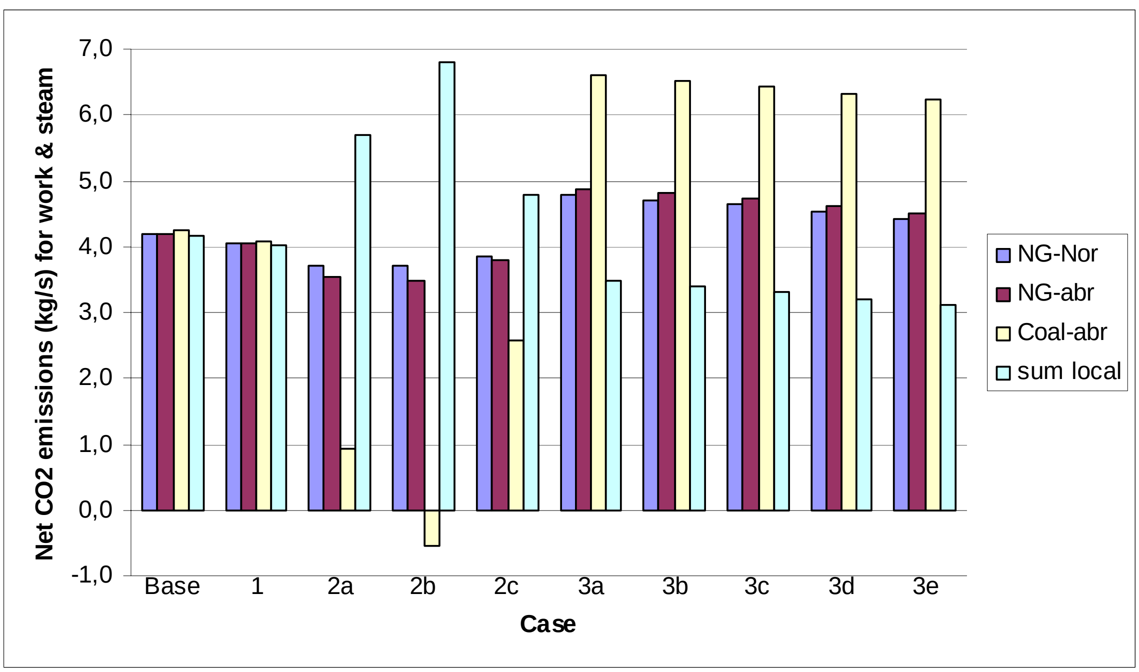
100
80
60
40
20
0
EEE (%), RPES (%), RAI (%)
Base
1
2a
2b
2c
Case
EEE
RPES
RAI

Fig 3



CO2 per exergy delivery (kg/MWh)
500
400
300
200
100
0
Base
1
2a
2b
2c
3a
3b
3c
3d
3e
Case

Fig. 4



Net CO2 emissions (kg/s) for work & steam
7,0
6,0
5,0
4,0
3,0
2,0
1,0
0,0
-1,0
Base
1
2a
2b
2c
3a
3b
3c
3d
3e
Case
NG-Nor
NG-abr
Coal-abr
sum local

Fig. 5