Proceedings of GT2005
ASME Turbo Expo 2005: Power for Land, Sea and Air
June 6-9, 2005, Reno-Tahoe, Nevada, USA

GT2005-68640

OPTIMIZATION OF THERMODYNAMICALLY EFFICIENT NOMINAL 40 MW ZERO EMISSION PILOT AND DEMONSTRATION POWER PLANT IN NORWAY

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ABSTRACT

In Aug 2004 the ***Zero Emission Norwegian Gas (ZENG)*** project team completed *Phase-1: Concept and Feasibility Study* for a 40 MW Pilot & Demonstration (P&D) Plant, that is proposed will be located at the Energy Park, Risavika, near Stavanger in South Norway during 2008.

The power plant cycle is based upon implementation of the natural gas (NG) and oxygen fueled Gas Generator (GG) (1500 ºF / 1500 psi) successfully demonstrated by *Clean Energy Systems (CES) Inc.* The GG operations was originally tested in Feb 2003 and is currently (Feb 2005) undergoing extensive commissioning at the CES 5MW Kimberlina Test Plant, near Bakersfield, California.

The ZENG P&D Plant will be an important next step in an accelerating path towards demonstrating large-scale (+200 MW) commercial implementation of zero-emission power plants before the end of this decade. However, development work also entails having a detailed commercial understanding of the techno-economic potential for such power plant cycles: specifically in an environment where the future penalty for carbon dioxide (CO2) emissions remains uncertain.

Work done in dialogue with suppliers during ZENG Project Phase-1 has cost-estimated all major plant components to a level commensurate with engineering pre-screening. The study has also identified several features of the proposed power plant that has enabled improvements in thermodynamic efficiency from 37% up to present level of 44–46% without compromising the criteria of implementation using "near-term" available technology. The work has investigated;

(i) Integration between the cryogenic air separation unit (ASU) and the power plant.
(ii) Use of gas turbine technology for the intermediate pressure (IP) steam turbine.
(iii) Optimal use of turbo-expanders and heat-exchangers to mitigate the power consumption incurred for oxygen production.
(iv) Improved condenser design for more efficient CO2 separation and removal.
(v) Sensitivity of process design criteria to "small" variations in modeling of the physical properties for CO2 / steam working fluid near saturation.
(vi) Use of a second "conventional" pure steam Rankine bottoming cycle.

In future analysis, not all these improvements need necessarily be seen to be cost-effective when taking into account total P&D program objectives; thermodynamic efficiency, power plant investment, operations and maintenance cost. However, they do represent important considerations towards "total" optimization when designing the P&D Plant.

We observe that *Project Phase-2: Pre-Engineering & Qualification* should focus on optimization of plant size with respect to total capital investment (CAPEX); and identification of further opportunities for extended technology migration from gas turbine environment that could also permit raised turbine inlet temperatures (TIT).

INTRODUCTION

The purpose of the ZENG Project Phase-1 was to gather information and propose a "Base Case" zero-emission plant that is appropriate for a pilot and demonstration phase of technology development. A main criterion has been to use components that are compatible with an investment decision being made in 3Q-2006 and plant commissioning in 2008.

Furthermore emphasis has throughout been placed on ensuring that such a P&D Plant would provide the necessary knowledge and experience to permit construction for "commercial" power plants of 240–400 MW_e (net export) in the 2010–2014 timeframe.

Such a goal for commercialization in the medium-term necessitates attaining power plant efficiency above 50% and ensuring that specific CAPEX is significantly reduced compared with what we estimate for the initial proposed nominal 40 MW P&D Plant.

There still remains considerable scope for optimizing and integrating the CES Gas Generator (GG) within a total balance of plant concept: the "Base Case" described extensively in the Phase-1 Report (Hustad *et al.*, 2004) has already been further developed and improved with respect to thermodynamic efficiency, as is described in this paper.

We are also confident that a focussed effort in Project Phase-2 will enable reduction in CAPEX as we continue to optimize plant integration and work more closely alongside the main equipment suppliers.

Furthermore, there will need to be continued work done regarding integration of CO2-handling, interim storage, transportation and commercial sale of CO2 for enhanced oil recovery (EOR) within the overall power plant design.

DESIGN BASE FOR 40 MW P&D PLANT

Proposed Plant location is on reclaimed "brown field" land made available at the Energy Park, Risavika, shown in Fig. 1.

Fig. 1: *Aerial view of reclaimed land area at the Energy Park, Risavika, nr. Stavanger, South Norway. Highlighted rectangle shows proposed location for the P&D Plant.*

Selection of the P&D Plant nominal design capacity equal to 40 MW_e (net export) corresponds to ~100 MW_t thermal power from the GG. This size of plant was chosen as being a reasonable compromise between development risk, economy of scale, CAPEX and technology status. It also provides a useful next-step on the path to commercialization from the 5 MW_e Kimberlina Test Plant that CES started commissioning near Bakersfield, Ca. during 4Q-2004.

The GG thermal power output scales with cross-sectional area; this therefore entails that the current (20 MW_t) GG diameter increases by a factor of 2.4—whilst length remains the same—and is considered to be within practical limits for empirical scaling of the on-going test experience.

Natural Gas (NG) is made available to the Stavanger region by *Lyse Gass AS* through a newly laid 10-inch diameter sub-sea pipeline from Kårstø with shore landing adjacent to the proposed P&D Plant site as indicated in Fig. 1.

Fuel Composition	Concentration (%-mol)
Methane (CH_4)	88.54
Ethane	7.71
Propane	0.50
i-Butane	0.03
n-Butane	0.04
Nitrogen (N_2)	0.69
Carbon dioxide (CO_2)	2.49

Table 1: *Summary of Fuel Composition for Natural Gas (NG). For the economic analysis we have assumed NG fuel cost to be 85 øre/Nm^3 (3.29 \$/GJ).*

The fuel gas in Table 1 has heat value (LHV) assumed to be 39.8 MJ/Nm^3 (equivalent to 47.7 MJ/kg) with a line pressure in the range 120–180 bar. With the "Base Case" this will be reduced to 94 bar for the GG and 30 bar for reheat (RH) combustion.

PROCESS DESIGN PHILOSOPHY

The process design was based on "current technology" and required that all major equipment items should be commercially available. We utilize a conventional cryogenic air separation unit (ASU) shown in Fig. 2 to supply pure oxygen to the GG—this being the most cost-efficient commercial method available to date.

Fig. 2: *Schematic view of the Air Separation Unit (ASU) adjacent to the main P&D Plant building.*

For the power train we employ a conventional steam turbine coupled to an electric power generator; if necessary through a speed reducer. The high-pressure (HP) turbine inlet steam temperature is restricted to 565 ºC, with an increase to 705 ºC for the intermediary-pressure (IP) turbine (as is acceptable from potential suppliers).

The low-pressure (LP) steam turbine exhaust flows to a vacuum condenser with 0.08 bar pressure. A pressure of 0.04 bar was investigated but this would have considerably increased the condenser size; bearing also in mind that the presence of CO_2 gas in the condensing steam will significantly increase the heat transfer resistance across the condenser compared to a conventional vacuum steam condenser. Furthermore it is advisable to keep the steam conditions upstream of the condenser above saturation level, to avoid corrosion (or erosion-corrosion) on turbine internals.

The mass flow and energy balance data necessary for the selection and dimensioning of the process equipment, fuel feed, utilities consumption etc., are generated by the computer program "Chemcad". The simulator contains comprehensive subroutines for calculation of thermodynamic, physical and transportation properties for the actual mixtures of the fluids involved in the main process, as well as in the utility systems.

However we did experience some variation in the results depending on the simulation subroutine models utilized. These originated from differences in the calculated physical properties for the CO_2 / steam mixtures within the lower pressure and temperature regimes. Subsequent discussions have confirmed that there would appear to be limited reliable data available in this region. This means that process data and equipment parameters in the low-pressure (sub-atmospheric) regime should be treated as preliminary.

Intermediate steam data is based on thermodynamic efficiency specifications obtained from recognized suppliers of steam turbines or state-of-the-art efficiency properties for such equipment as indicated in Table 2. Efficiency factors for gas compressors are based on catalogue values.

Component	Efficiency Factor
HP turbine	0.89
IP turbine	0.90
LP turbine	0.93
Electric Power Generator	0.95

Table 2: *Assumed power train efficiency factors.*

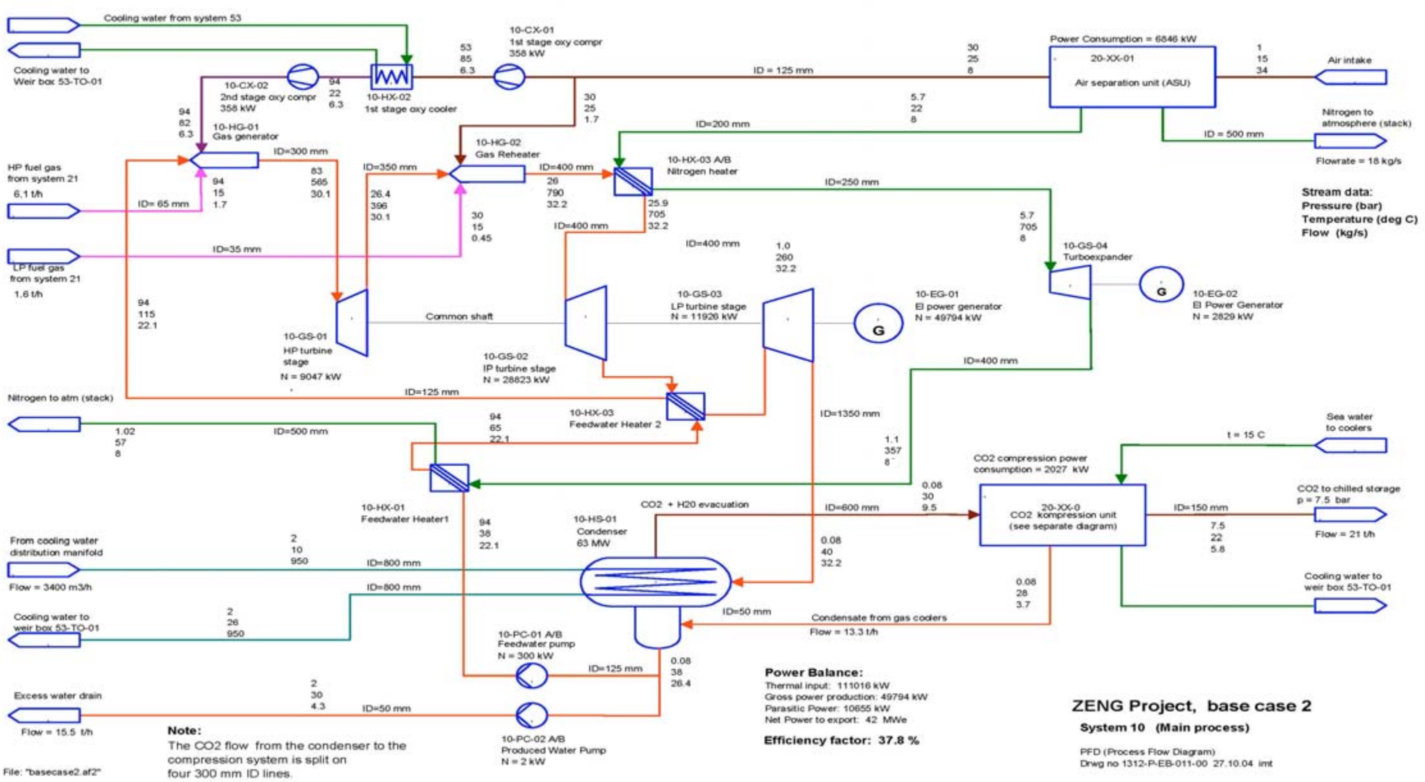

Fig. 3: *Process flow schematic for "Base Case" configuration with 42 MW_e net output and cycle efficiency of 37.8%.*

DESCRIPTION OF BASE CASE PROCESS

The NG fuel is fed at 94 bar to the GG injection nozzles through a filtering and pressure reduction control station (see Fig. 3). The gaseous fuel and pure oxygen are combusted in combination with injection of water in a complex manifold and nozzle system; establishing near ideal conditions for stoichiometric combustion and temperature control within the combuster section of the GG shown in Fig. 4.

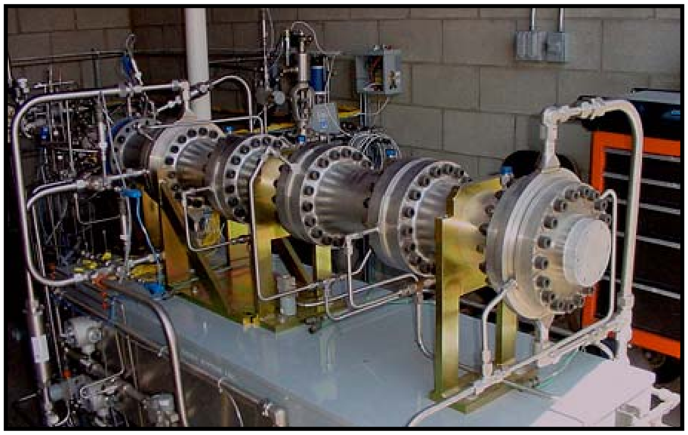

Fig. 4: *The 20 MW_t CES Gas Generator (GG). Combuster section is at far end followed by 4 sequential water-cooldown sections. Closest to observer is the downstream endplate that provided back-pressure during testing 'in lieu' of HP turbine.*

The GG exit pressure is controlled at 83 bar by the rate of fuel and oxygen flow. The gas (CO_2 / steam) temperature is controlled at 565 ºC by the water-injection rate in the cooldown sections. The GG wall temperature is controlled by the flow of water through internal cooling passages within the housing.

The process gas stream is routed through the HP turbine and expanded to the outlet pressure at 26.4 bar and 396 ºC. The HP turbine shaft duty is 9.0 MW.

Next the process gas temperature is raised to 790 ºC using a reheat (RH) combustion chamber operating at 26 bar pressure and fed with NG fuel and oxygen at near stoichiometric ratio [Chorpening *et al.*, 2003]. The process gas stream at the RH outlet comprises a mixture of 17% CO_2 and 83% H_2O (steam) based on %-weights.

Before the IP turbine the process gas passes through a nitrogen gas heater and is cooled to 705 ºC; close to the currently maximum permissible IP turbine inlet temperature (TIT). The IP turbine expands the process gas from 26 to 1 bar and a temperature of 260 ºC. The IP turbine shaft duty is 28.8 MW.

The nitrogen gas (partially taken from the ASU) is expanded in a turbo-expander from 5.7 bar (705 ºC) to 1.1 bar (357 ºC) producing 2.8 MW_e additional power.

Next the process gas is led to the LP turbine where it is expanded to the condenser pressure of 0.08 bar and a temperature of 40 ºC. This is maintained sufficiently above steam saturation temperature, in order to avoid corrosion problems in the steam turbine and exhaust channels. The LP turbine shaft duty is 11.9 MW. The total turbine duty is 49.8 MW, whilst the electric generator efficiency is assumed to be 95%.

The exhaust steam from the LP turbine is condensed in a seawater-cooled condenser. In addition to CO_2 / steam mixture, the flow to the condenser contains a small amount of oxygen and a trace of carbon monoxide. The concentrations of unburned hydrocarbons and NOx are anticipated to be essentially zero. At an absolute pressure of 0.080 bar, partial pressures of the main components are 0.0065 bar for the CO_2 and 0.0735 bar for the steam (at which pressure the condensation temperature is estimated to be 39.9 ºC).

The seawater flow requirement for the condenser is calculated to be ~3,400 m^3/h with assumed cooling-water inlet temperature of 15 ºC at summer conditions.

"Base Case" Cycle Summary Data	
Thermal power input	111.0 MW
Gross power output	49.8 MW
Parasitic power*	10.7 MW
Net power	42.0 MW_e
Overall cycle efficiency	37.3%
Fuel consumption	7 700 kg/h
Oxygen consumption	28 800 kg/h
Cooling water flow (total)	~ 4 300 m^3/h
Excess water production	15.5 m^3/h
HP Turbine inlet pressure	83 bar
HP Turbine inlet temperature	565 ºC
HP Turbine exhaust temperature	~ 396 ºC
IP Turbine inlet pressure	25.9 bar
IP Turbine inlet temperature	705 ºC
LP Turbine inlet temperature	260 ºC
LP Turbine inlet pressure	1.0 bar
LP Turbine exhaust temperature	40 ºC
Condenser pressure	0.08 bar†

Table 3: *Summary Data for "Base Case" Configuration.*

OPTIMIZED PROCESS DESCRIPTION

To date the practical limit for steam temperature from conventional boilers has been around 565 ºC. And no strong

* The "parasitic" power also includes electric energy consumption for the ASU, oxygen and CO2-compressors as well as cooling-water supply pumps.

† The condenser pressure was also increased from 0.08 bar to 0.15 bar due to recommendation from the CO2-compressor suppliers. A higher pressure could significantly decrease the dimensions and costs for both the compressors and intercoolers. This increase in condenser operating pressure reduced the cycle efficiency from 37.8 to 36.3%.

market incentive has existed for the development of steam turbines with higher temperatures. The CES GG however presents new possibilities for cycle improvement with increased steam temperature and process pressure. However, steam turbines will not accommodate significant increase of TIT without introduction of secondary flow and internal blade cooling, together with utilization of sophisticated materials.

But current gas turbine (GT) technology is already operating at significantly higher TIT albeit at comparatively lower pressures: these present an excellent opportunity for inclusion as IP turbines in an "Optimized" process scheme as shown in Fig. 5. In such cycles the IP turbine TIT may potentially be elevated to 1450 ºC, resulting in a very substantial increase in cycle efficiency.

However for practical purpose this would require some—but still limited—redesign of a suitable gas turbine. Such GT development work is considered being "a few years" ahead of the initial demonstration goals of the current P&D Plant. And would require commercial drivers before the equipment suppliers may chose to participate. We have therefore maintained a TIT of ~700 ºC for the time being.

However, to provide an indication of the "short term" potential for improvement, we include here an optional process scheme based on cycle integration using a RR-WR21 recuperated gas turbine (as proposed by Phillips, 2004).

Included in the "Optimized" configuration is also a "double" Rankine steam cycle, together with integration of the air compressor and nitrogen expansion from the ASU. (Nitrogen expansion is here principally the same as for the "Base Case" but now with the total nitrogen flow routed through the expander, thereby contributing significantly to the power production and cycle efficiency.)

The benefit of the double Rankine cycle is that separation of CO2 is achieved at a pressure of 3.0 bar, thus reducing the number of CO2-compressors and dimensions for the CO2-handling equipment. Furthermore the LP "pure" steam Rankine cycle can now have a reduced condensation pressure (0.03 bar) compared with the "Base Case" process (0.08 bar)—this too contributes significantly to overall cycle efficiency.

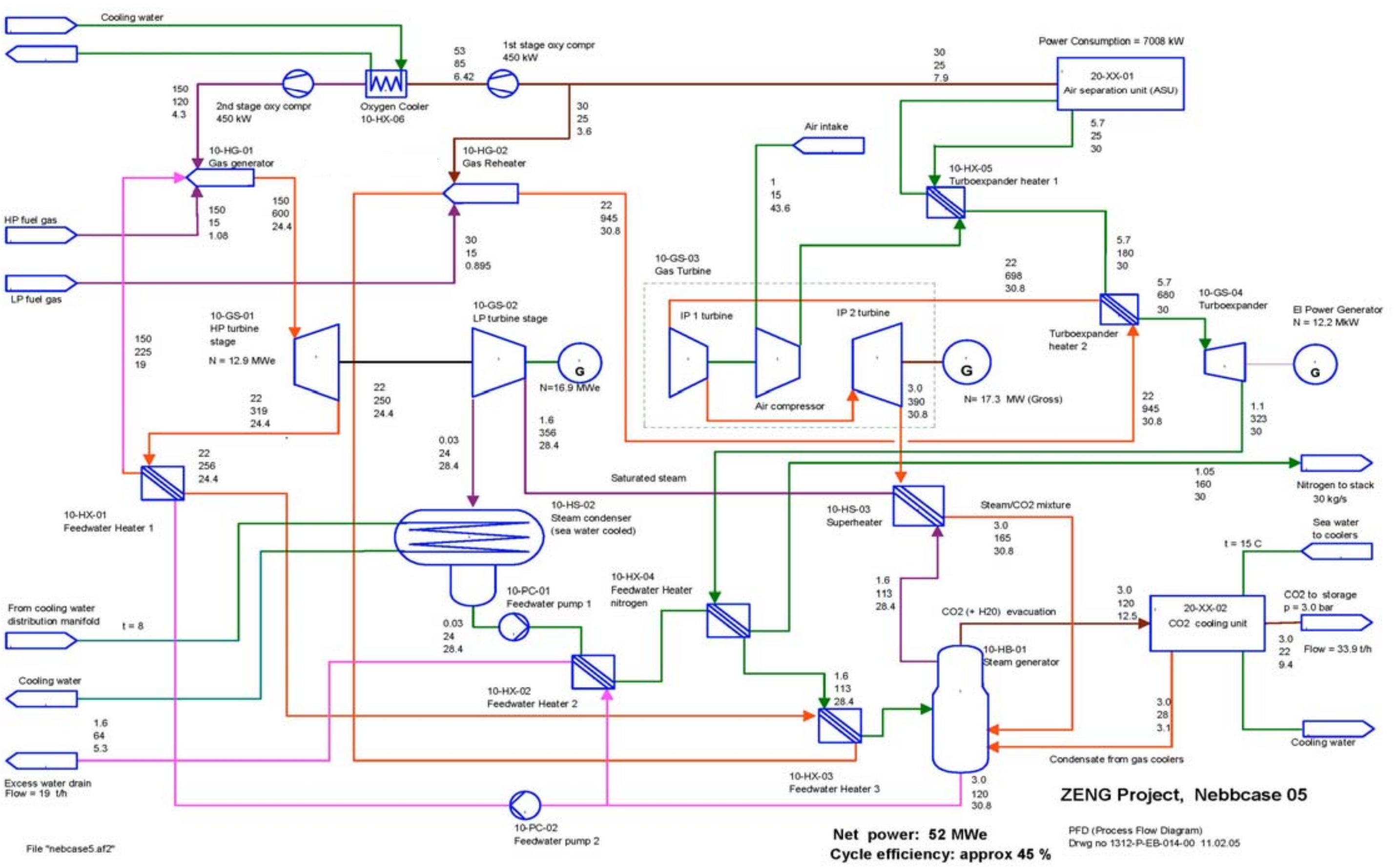

Fig. 5: *Process flow schematic for "Optimized" configuration with 52 MW_e net output and cycle efficiency of 45%.*

The GG, HP turbine, reheater (RH), nitrogen heater, turbo-expander, feed-water heaters and oxygen compressors are principally the same as in the "Base Case". While the LP cycle is a conventional "cogen" condensing steam turbine. The compressor delivers compressed air to the ASU (see Fig. 5).

The GG is operated at 150 bar and therefore a separate pressure reduction station for initial fuel handling should not now be necessary. Total fuel feed to the GG injection nozzle is 1.08 kg/s. The fuel energy supplied is 65 MW and the outlet energy flux is 84 MW. The process gas at the GG exit contains approximately 5.3 %-mol CO_2 while the combustion generates 3.57 kg/s CO_2 and 2.92 kg/s steam.

The process gas at 150 bar and 600 ºC is routed to the HP turbine where it is expanded to 22 bar and ~ 320 ºC. With turbine efficiency maintained throughout as specified in Table 2 the HP turbine stage shaft duty is now 12.9 MW.

The process gas is passed through heat exchanger (10-HX-01) to raise GG feed water temperature from 120 to 225 ºC. The process gas stream also heats feed water to the LP steam generator (10-HB-01) up to vaporization temperature of 113 ºC.

Next the process gas stream is routed to the RH operating at 22 bar and where the process gas temperature is raised to 945 ºC by stoichiometric combustion of fuel gas with oxygen. Fuel consumption is 0.895 kg/s (equivalent to 44.6 MW fuel energy). The RH combustion produces 2.88 kg/s CO_2 and 2.40 kg/s steam; the process gas stream now comprises 6.48 kg/s CO_2 and 24.3 kg/s steam, with the CO_2 concentration being 9.5 %-mol and energy stream flux is 115 MW.

Next the process gas stream flows to the turbo-expander heater (10-HX-05) where 30 kg/s (all available) nitrogen is heated to 680 ºC whilst the process gas temperature is reduced to ~ 700 ºC in order to be compatible with TIT for the IP gas turbine (10-GS-03)

The turbo-expander (10-GS-04) produces 12.2 MW_e power and has an exhaust temperature of 323 ºC; this is heat-exchanged against feed water in the LP steam cycle, reducing temperature of the nitrogen exhausting to atmosphere to ~ 160 ºC. Which is still comparatively high and we should be able to make better use of this with further optimization!

The IP gas turbine (10-GS-03) is based on a modified design derived from a recuperated GT (e.g. Rolls-Royce WR-21) where the recuperator is removed and principally replaced by the gas re-heater. The process fluid expands from 22 to 3.0 bar—through two stages—with temperature decrease from 705 to ~ 390 ºC. Normal exhaust condition for the WR-21 is atmospheric pressure, hence the last turbine stage(s) will need to be modified or removed. The turbine shaft duty is estimated to be 17.3 MW.

The IP exhaust steam is led to the steam superheater (10-HS-03) for the LP steam cycle, where saturated steam from the steam generator (10-HB-01) is heated from 113 to 356 ºC. The steam generator is a conventional unit, as normally utilized for production of clean steam from "unclean" steam sources, where the contaminants may be particles, inert gases, volatile organic components, etc.

The superheater for the produced clean steam is also a conventional freestanding unit, comprising of tube banks in a countercurrent arrangement.

The exhaust steam is routed to the steam generator, where the steam fraction is condensed by heat-exchange against the (boiling) feedwater to the steam generator. The mol-fraction of steam in the process fluid is 0.90.

The superheated steam (at 1.6 bar and 356 ºC) is routed to the LP turbine where the steam is expanded to condenser pressure at 0.03 bar and ~ 24 ºC. The LP turbine stage shaft duty is estimated to be 16.9 MW.

The exhaust steam from the LP turbine is condensed in a seawater-cooled condenser (10-HS-02). At an absolute pressure of 0.03 bar the condensation temperature for the steam is 24.1 ºC.

In this preliminary study we have not to date included recompression of CO_2 from 3.0 bar to 7.5 bar followed by chilling to -45 ºC making it completely ready for interim storage and subsequent ship transportation.

"Optimised" Cycle Summary Data	
Thermal power input	115.6 MW
Gross power output	59.3 MWe
Parasitic power	7.2 MWe
Net power	52.0 MWe
Overall cycle efficiency	45.0 %
Fuel consumption	7 110 kg/h
Oxygen consumption	28 440 kg/h
Cooling water flow (total)	~ 4 300 m^3/h
Excess water production	19.0 m^3/h
HP Turbine inlet pressure	150 bar
HP Turbine inlet temperature	600 ºC
HP Turbine exhaust temperature	~ 320 ºC
IP Turbine inlet pressure	22.0 bar
IP Turbine inlet temperature	698 ºC
IP Turbine exhaust temperature	~ 390 ºC
CO2 / Steam Condenser pressure	3.0 bar
LP Steam Rankine Cycle	
LP Turbine inlet pressure	1.6 bar
LP Turbine inlet temperature	356 ºC
LP Turbine exhaust temperature	24 ºC
Steam Condenser pressure	0.03 bar

Table 4: *Summary Data for "Optimized" Configuration.*

ECONOMIC ANALYSIS

Within the present study we have identified and cost-estimated all major components for the "Base Case" configuration and made comparison with a conventional NG Combined Cycle (NGCC) Power Plant. Included here are also cost-factors based on accumulated project experience in Norway: with high labor costs and strong local currency these can typically lead to estimates that are 25–30% above US Gulf Coast.

The economic model permits input of all main power plant parameters; CAPEX, operating costs, internal rate of return, project duration, efficiency, net power generation, sale of CO2, etc. Model output is derived using annualized cash flow and calculates cost of electricity (CoE) by prescribing a zero net present value. All modeling is pre-tax. We have made the following generalized basic assumptions:

- 10% discount rate and project economic life of 25-yrs.
- Fuel cost is 85 øre/Nm^3 NG (equivalent to 3.29 $/GJ).
- For P&D Plant we assume 60% financed debt at 5% interest. This reflects some "goodwill" from the Norwegian government's willingness to help promote development and demonstration of such "new" power generation technology[1].
- For comparison between the conventional NGCC Reference Plant and a "commercial" ZENG-CES plant we revert to assuming 100% equity financing.
- Assume two years for total investment and construction.
- Assume 6 weeks for commissioning during first year.
- Exchange rate is 6.50 NOK/US$.
- CoE is expressed in mills/kWh (1,000 mills/US$) and in Norwegian currency as øre/kWh (100 øre/NOK).

Using Reference CoE from the NGCC without CO2-capture[2] we can also calculate a CO2-capture cost ($ per ton) for comparison with a pre-requisite sale of CO2 for EOR to a CO2-aggregator / transporter / oilfield operator.

For the "Base Case" 42 MW_e P&D Plant (inclusive of the ASU) we have total CAPEX of $93 million (equivalent to 605 MNOK). With further focus on cost optimization in Project Phase-2 and with economies of scale, we believe there is considerable opportunities for reducing this CAPEX.

The incremental CoE for the "Base Case" is estimated to be +25.4 mills/kWh compared with the 400 MW Reference CoE. Alternatively, the plant would need to recover a CO2-capture cost of $27.3 /ton (at perimeter fence) in order to be competitive with electricity from the Reference Plant[3].

For the "Optimized" Configuration we have estimated total CAPEX to be $106.8 million (equivalent to 694 MNOK). Net export power is 52 MW_e resulting in incremental CoE of +17.6 mills/kWh compared with the Reference CoE. Alternatively the "Optimized" P&D Plant would need to sell it's CO2 at a price of 16.7 $/ton (at perimeter fence).

The CO2-liquefaction plant (with storage facilities) and transportation to offshore platform were outside Scope of Work for the Phase-1 Study. However, one may <u>conservative</u>ly account for these additional cost in CO2-handling by assuming an additional ~ $13 /ton (see Hustad and Austell, 2004) in order to indicate that delivered cost should be ~ $30 /tCO2.

Recent alternative studies have indicated delivered cost for CO2 on North Sea platform to be in the range from $35 /tCO2 as proposed by Elsam / Kinder Morgan, CENS Project (Markussen *et al.*, 2002). Alternatively up to $48 /tCO2 as presented by Statoil for proposed CO2-flooding at Gullfaks.

In the medium-term (2010-2012) we have identified cycle optimization opportunities that could ensure plant efficiency of ~ 51%. Furthermore, we assume cost-optimization and economies of scale can jointly contribute to ensure an additional one-third reduction in specific CAPEX. This would entail that a "100% equity financed" commercial 240 MW ZENG-CES Power Plant could have a CO2-capture cost (at perimeter fence) of $21.1 /tCO2 whilst delivering 0.77 mtCO2/yr for EOR.

In this context the key economic variable is the market price of crude oil which determines the sales value of CO2 for EOR. Again we may assume using larger volumes that delivered cost of CO2 at the offshore platform from such a 240 MW ZENG-CES Power Plant could be ~ $30 /tCO2. Thus, even with the current fiscal regime in the North Sea—which is not yet optimized to create incentives for CO2-EOR—the pre-requisite crude oil price needed to sustain project economics would be in the range $26 – $29 /bbl (see Hustad and Austell, 2004).

In the longer-term (2012–2014) we foresee technology improvements[4] and economy of scale that should permit a 400 MW ZENG-CES Power Plant to operate with ~ 54% efficiency and have specific investment cost below 1,400 $/kW. Economic modeling for such a plant suggest it would have a CO2-capture price of $11.4 /tCO2 whilst delivering 1.21 mtCO2/yr.

[1] In 2004 the Norwegian government specifically set aside a fund of $310 million to promote P&D Power Plants with CO2-capture & storage (CCS). They have also indicated that as cost-effective technologies emerge, then they shall be willing to further add to this level of support if necessary.

[2] The Reference Plant assumes a new build on the West Coast of Norway with specific CAPEX of 745 $/kW installed. We obtain CoE at 35.2 mills/kWh (22.9 øre/kWh) exclusive of CO2-emissions. We assume that the Reference Plant will need to purchase CO2-credits for an additional cost of $12 /tCO2 starting in 2008 and rising linearly to $24 /tCO2 at end of project economic lifetime. With these assumptions we derive a Reference CoE equal to 40.7 mills/kWh (26.5 øre/kWh). For further details see Hustad *et al.* (2004).

[3] Here we assume "Base Case" P&D Plant will capture 100% of its CO2-emissions at a pressure of 7.5 bar. This will subsequently need to be dried and cooled to -45 ºC (near triple point) for liquefied storage and ship transportation.

[4] Specifically we here foresee commercial introduction of; (i) oxygen membrane technology, and (ii) blade-cooling to permit increased TIT for the HP and—in particular—the IP turbine expansion stages.

Fig. 6: *Schematic sketch of the proposed 40 MW (nominal) Pilot & Demonstration Power Plant at the Energy Park in Risavika.*

In this timeframe we may also assume that cost of CO2 transportation from the power plant perimeter fence out to oilfield will have become aggregated and handled in a more cost-effective manner within a dedicate CO2-infrastructure.

We therefore assume a future delivered price for CO2 to be ~ $17 /ton. And in which case the long term sustaining price of crude oil would need to be above ~ $22 /bbl.

The 10-Year Forward price for market crude oil is currently above $32 /bbl. Thereby indicating that there may be a substantial commercial upside in developing zero-emission power plants solely on the basis of enhanced oil recovery. Needless to say additional benefit may also be had when also taking account of greenhouse gas mitigation.

CONCLUSIONS

Results suggest that a ZENG-CES Power Plant, in combination with sale of CO2 for enhanced oil recovery (EOR) could provide up to 3.2 TWh of base load (+8,000 hours per year) zero-emission electricity by 2011. And in a carbon constrained market could during project economic lifetime be more cost-effective than a conventional power plant.

Furthermore, such zero-emission Power Plants in combination with recognized CO2-EOR opportunities, represents a competitive business opportunity providing an important contribution to the use of natural gas in Norway as well as possible life-extension for the mature oil reservoirs on the Norwegian Continental Shelf.

ACKNOWLEDGMENTS

The following have participated and contributed to the ZENG Project Phase-1 Study Report: ***Sweco Grøner AS*** (a Swedish / Norwegian engineering consultancy), Steen Jakobsen, Tommy Johnsen, Brita Schaulund, Knut Skjæveland and Øystein Østerhus; ***Nebb Engineering AS*** (a Norwegian power and process engineering consultancy), Inge Trondstad; ***Det Norske Veritas*** (the Norwegian safety, risk and certification company), Håvard Thevik; ***Clean Energy Systems Inc.*** (California-based technology developer), Roger Anderson, Ron Bischoff, Larry Hoffman, Keith Pronske and Vic Viteri; ***CO2-Norway AS*** (a Norwegian project developer), J. Michael Austell, Carl-W. Hustad and Steffanie Siemes; ***Lyse Energi AS*** (the regional utility company), Jan Arvid Øvestad; ***Energiparken AS*** (Manager, Energy Park), Per Møller-Pedersen.

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