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Emission Reduction by the Use of Supercritical CO_2

2014

Emission Reduction by the Use of Supercritical CO_2 Cycles

C. B. Meher-Homji, D. Messersmith, P. Pillai
Bechtel Corporation

K. Masani
ConocoPhillips Company

GASTECH 2014

Seoul, Korea
24-27 March 2014

ABSTRACT
The use of supercritical CO_2 (sCO_2) as a working fluid for waste heat recovery has several advantages and can be deployed in a LNG facility boosting the overall thermal efficiency and reducing emissions. Supercritical CO_2 is inexpensive, nonflammable and easily available. Due to its high working pressure, highly compact systems can be built. The high density and volumetric heat capacity of CO_2 with respect to other working fluids make it very effective in capturing waste heat from gas turbines. This paper examines the feasibility of power generation for a LNG Liquefaction facility and makes a comparison of such a system to traditional simple cycle gas turbines and combined cycle systems. The deployment of this technology for power generation from waste heat can significantly reduce emissions. Steam based combined cycles are thermodynamically weak in performance when coupled with high efficiency aeroderivatives because of their low exhaust temperature and mass flow. This weakness can, to some extent, be offset by the deployment of supplemental firing and multi-pressure level heat recovery, both of which increase complexity. Organic Rankine Cycles have the disadvantage of using a thermodynamically inferior and toxic working fluid. These issues can be overcome by the use of supercritical CO_2. With a supercritical CO_2 system, the power attainable at ISO conditions from a LM6000 is at least 10 MW. Due to the properties of supercritical CO_2, equipment is much smaller, and will cost less than traditional steam HRSG equipment. Plot space is significantly lower than comparable steam systems. The most advanced packaged equipment currently is being tested in the 7-10 MW size.

1.0 INTRODUCTION

The thermal efficiency of a LNG facility depends on numerous factors such as gas composition, inlet pressure, and other factors such as the location of the loading dock relative to the liquefaction process which impacts the heat leakage into the cryogenic system. Gas turbine selection, the use of waste heat recovery, ship vapor recovery, and the power generation configuration, all have a significant effect on the overall thermal efficiency of the LNG process. Detailed discussions of LNG plant thermal efficiency have been made by Yates (2002) and Ransburger (2007) and studies on combined cycle approaches have been made by Avidan et al (2003) and Meher-Homji et al (2012). Market pressures for thermally efficient and environmentally friendly LNG plants coupled with the need for high plant availability have led to the application of high efficiency aeroderivatve gas turbines, the first one being the Darwin LNG facility that implemented steam cogeneration by incorporating heat recovery units on four of the six mechanical drive gas turbines in refrigeration service (Meher-Homji et al, 2007, 2009). This paper focuses on the use of supercritical CO_2 for heat recovery in the ConocoPhillips Optimized Cascade® Process shown in Figure 1.

The Thermal Efficiency [η_{LNG}] of a plant can be defined as:

$$\eta_{LNG\ Pant} = [\text{ LNG BTU + NGL BTU}] / \text{Feed BTU}$$

Another way of looking at it is by means of auto consumption which is defined as

$$\text{Auto consumption} = 1 - \eta_{LNG\ Plant}$$

The Auto consumption is a measure of the fuel utilized by the LNG Facility for

- The refrigeration turbines
- The power generation turbines
- Other fuel use for process heating, such as hot oil heaters, etc.

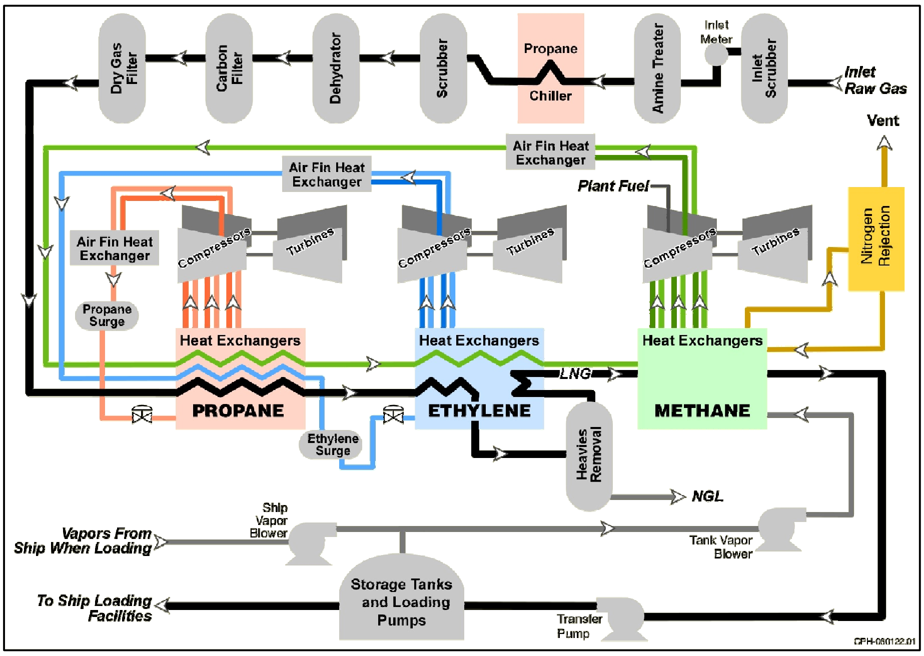

Figure 1. Simplified process flow diagram of the CoP Optimized Cascade® Process.

The minimizing of feed usage (auto consumption) helps in the reduction of CO_2 emissions. Additionally, in gas constrained situations savings in fuel auto consumption of the liquefaction facility can be converted into LNG production. The imposition of a CO_2 taxes will further promote the need for higher energy efficiency and reduction of CO_2. This paper examines the use of advanced heat recovery utilizing supercritical CO_2 as a working fluid as a means to replace the power generation equipment within the ConocoPhillips Optimized Cascade® Process. The system can be deployed on aeroderivative gas turbines with exhaust temperatures in the 500°C range. Aeroderivative engines offer very attractive efficiencies and the deployment of supercritical CO_2 heat recovery can be used to enhance efficiency without the use of complicated steam systems.

In high pressure aeroderivative engines which are optimized for higher efficiency, steam based combined cycles tend to be relatively weak. When steam bottoming cycles have to be used on aeroderivative engines, dual pressure levels often have to be used and supplementary firing also may have to be considered to maximize HRSG efficiency. An aeroderivative engine with a thermal efficiency of 41%, will yield a heat to power ratio of 0.99, while for an industrial gas turbine with a thermal efficiency of 35% the corresponding heat to power ratio would be 1.46. With aeroderivative engines the heat recovery efficiencies tend to be low due to the low exhaust temperature of the exhaust gas.

1.1 CO_2 Reduction

In the overall LNG chain, liquefaction contributes the largest CO_2 footprint (mainly due to the refrigeration drivers and power generation) which accounts for approximately eighty percent of the CO_2 emission in the LNG supply (Rabeau et al, 2007). Consequently, any ability to reduce CO_2 will have a major impact on overall greenhouse gasses. By removing the gas turbine generators and generating plant power required using a supercritical CO_2 cycle, a significant source of greenhouse gas can be eliminated. CO_2 reduction has become a very important component in plant design.

Many major Oil and Gas companies have developed an internal carbon price that they use as a basis for business planning and strategy. As reported by Thinnes (2014) prices used range from \$30-\$60/metric ton of CO_2 and a value of approximately \$40/tonne is a commonly used number. This value has been used in the techno-economic evaluation provided in this paper.

2.0 OVERVIEW OF SUPERCRITICAL CO_2 CYCLES

2.1 Overview of Supercritical CO2

There is currently much interest and focus on the use of supercritical CO2 as a working fluid for power cycles and is considered to be a game changer for improvement. Several large and small companies are active in this area including Dresser-Rand, Echogen, GE, Barber Nichols, Pratt and Whitney and Bechtel in creating these next generation turbines and are placing a significant emphasis on sCO_2 (Robb, 2012 and Wright, 2012).

Supercritical CO_2 has been considered as an alternative working fluid for power cycles for the past 70 years due to its advantage over steam (Dostal, 2004). The density of sCO_2 allows turbomachinery to have a much smaller footprint than comparable steam turbines and reduces the need for two-phase hardware. Due to the operating pressure and highly variable, non-linear fluid properties, suitable hardware for industrial use did not exist until recently. Advancements in compact heat exchangers and turbomachinery coupled with the drive to operate more efficiently has revived interest in sCO_2 power cycles. Additional information may be obtained in the sCO_2 Power Cycle Technology Workshops (http://www.sco2powercyclesymposium.org)

2.2 Properties of sCO_2

Carbon dioxide behaves as a gas in air at normal temperatures. If the temperature and pressure are both increased to be at or above the critical point for carbon dioxide, it adopts properties midway between a gas and a liquid, and behaves as a supercritical fluid. The critical point of CO_2 is low (31 °C and 74 Bara) compared to other fluids, allowing for heat transfer from low temperature (200°C to 500°C) sources to the supercritical state. A supercritical fluid is dense like a liquid but expands to fill a volume similar to a gas. Small changes in temperature at the critical point (31°C) causes a change in density by a factor of 3 or 4 compared to a change of 1000 times for the case of when water changes from liquid to vapor. There is also a large spike in heat capacity near the critical point of CO_2. These properties make sCO_2 an excellent working fluid for power cycles.

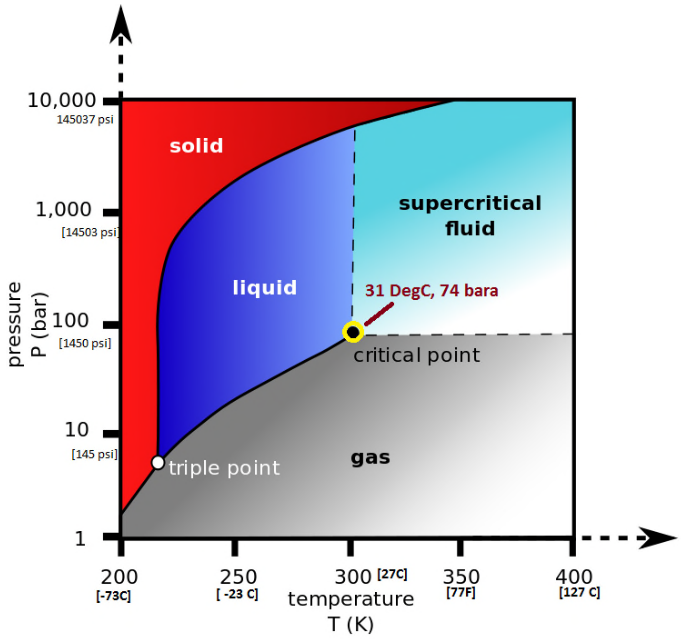

Figure 2. Supercritical CO2 Phase Map Modified from http://en.wikipedia.org/wiki/Supercritical_carbon_dioxide

2.3 Supercritical CO_2 Power Cycles

There are two types of sCO_2 cycles that are currently being studied and researched in industry.

2.3.1 **Rankine Cycle Bottoming Cycles.** This is the cycle that is the focus of this paper, where heat rejection takes place in the subcritical region and the sCO2 turbine exhaust is routed to a condenser where phase changes from vapor to liquid. It is a bottoming cycle similar to a steam cycle used in a combined cycle power plant. This cycle is ideal for heat recovery from an aeroderivative gas turbine. The low pressure ratio of this cycle requires a recuperator. A leading company in this area is Echogen[1] and their cycle is shown schematically in Figure 3.

[1] www.echogen.com

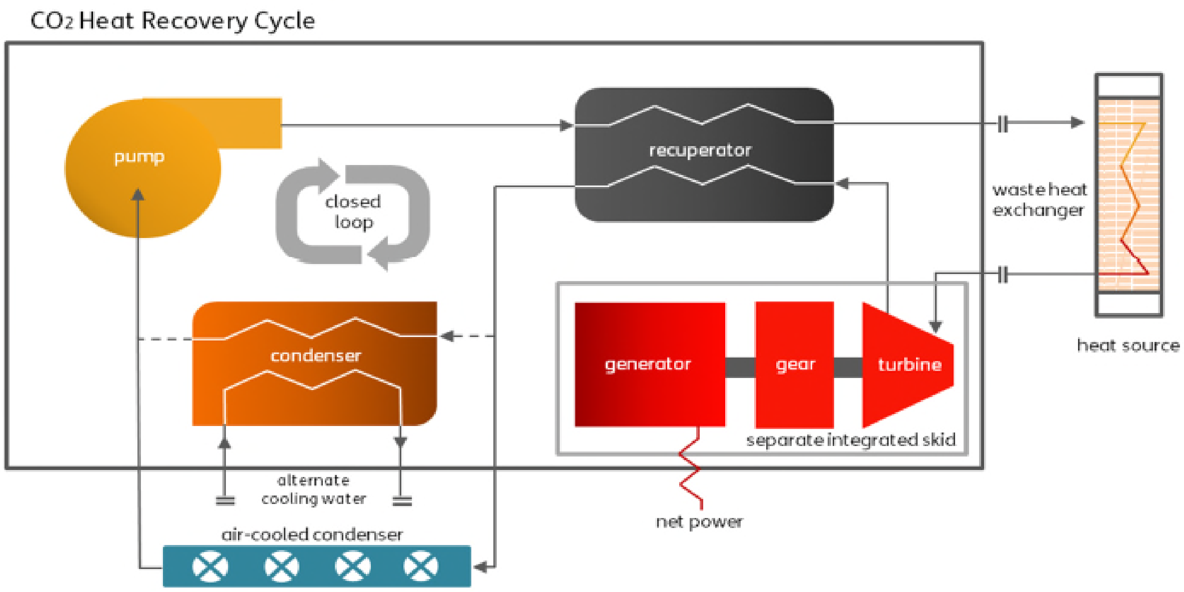

Figure 3. Schematic of Echogen sCO2 Waste Heat Recovery System (Courtesy Echogen Power Systems & Dresser-Rand)

In this cycle, the GT exhaust is used to heat the sCO_2 to a high energy level and it is then expanded in a turbine which drives a generator. The exhaust fluid is then routed to a recuperator where it exchanges heat and is condensed to a liquid at the condenser. The fluid is then pumped into the recuperator (to gain heat) and completes the cycle. The condenser can be either air or water cooled. The high density sCO_2 on both sides of the recuperator allow the use of compact microchannel based heat exchanger technology.

A small 250 kW demonstration system has been built and successfully tested at American Electric Power. A commercial grade 8 MW system has been tested at Olean NY. The process and power skid of this system is shown in Figure 4. Details on this technology may be found in Held et al (2012), and Persichilli et al (2011).

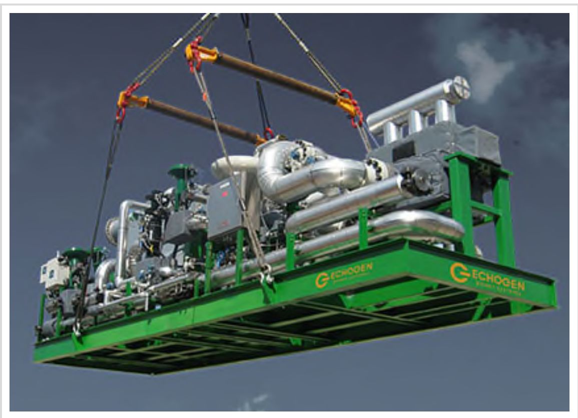

Figure 4. Echogen EPS 100 Process skid (left) and Power Skid Right (Courtesy Echogen Power Systems and Dresser-Rand)

2.3.2 **Brayton Cycle (Gas Turbine Cycles)** – this system is a closed system where there is no phase change in the working fluid in this cycle and the fluid remains in the supercritical state. It is typically a recuperated closed cycle where the turbine exhaust is used to heat the compressed sCO_2 in order to improve efficiency, similar to what is done in recuperated gas turbine cycles. These cycles have low pressure ratios typically below 3:1[2] Details may be found in Moroz et al 2013.

2.4 Advantages of sCO2 Compared to Combined Cycle Steam Systems and ORCs.

Waste heat recovery has traditionally been done by either combined cycles (a Rankine bottoming cycle) utilizing the waste heat from a gas turbine or Organic Rankine Cycles (ORCs) that are closed loop cycles that utilize fluids such as freons or cyclopentane. Of these, steam cycles are much more common.

Supercritical CO_2 is an excellent working fluid for closed loop power generation for the following reasons:

- Low cost: the installed cost would be about 30-40% less than the cost for a dual pressure level steam system (Kacludis, 2012). Much of these savings are in the reduced turbine costs, balance of plant equipment and civil works.
- The working fluid is nontoxic and nonflammable which allows the heat recovery exchanger to be directly mounted within a gas turbine exhaust stack.
- High fluid density of sCO_2 enables very compact turbomachinery and a recuperator heat exchanger.
- Designs allow heat recovery similar to that of a dual pressure heat recovery steam system, but at lower cost and without the complexity of the water cycle.
- Plot plan requirements according to Perscichilli, 2012, are approximately two-thirds of a system HRSG plant.
- Equipment is skid mounted enabling ease of installation.
- O&M costs would be less than that for a comparable steam system, as there would be no water quality monitoring and control costs complexity relating to condensate return. In an LNG facility, the sCO2 plant would operate at a steady load condition most of the time with increased power being required during ship loading.
- Startup times for sCO_2 systems is approximately 20 minutes to full power.

A plot plan for a LM2500 class system is shown in Figure 5. While the size for a LM6000 system will be marginally larger, it will not scale directly. The system shown has a traditional horizontal HRSG, but vertical arrangements can also be used. There are several vendors available who have experience with the development of such heat recovery systems.

One of the issues that occur with steam systems is the pinch point limitation as shown in Figure 6 (Persichilli et al , 2012). In a single pressure level steam system, the pinch point results in a lower temperature. As the sCO2 system does not change phase above its critical point during the heat addition process, there is no pinch limitation and sCO2 provides a better match of the heat source temperature profile compared to situations where boiling takes place[3]. The next section shows the complexity of some thermal designs of steam systems for a LM6000PF engine.

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[2] At a critical pressure of approx. 74 bar, a 3:1 pressure ratio results in a peak pressure of 222 bar which is extremely high.
[3] An actual T-Q diagram for a single pressure level steam system on a LM6000PF engine is shown in Figure 8.

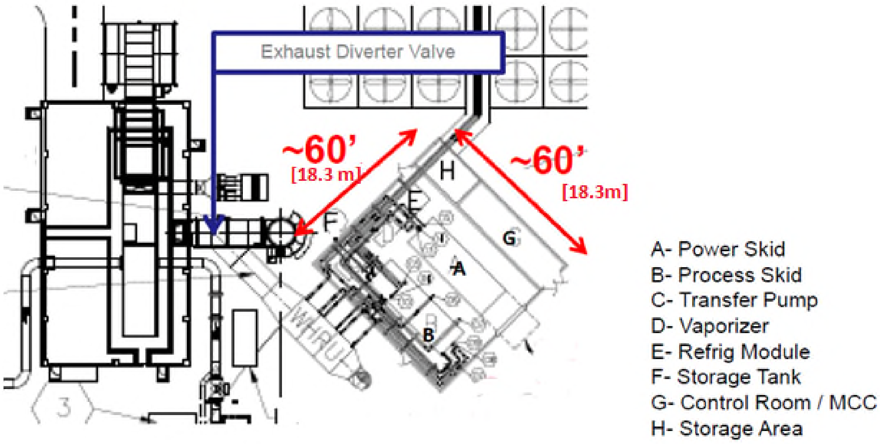

Figure 5. Plot plan for LM2500 class Gas Turbine. (Courtesy Dresser-Rand and Echogen Power Systems)

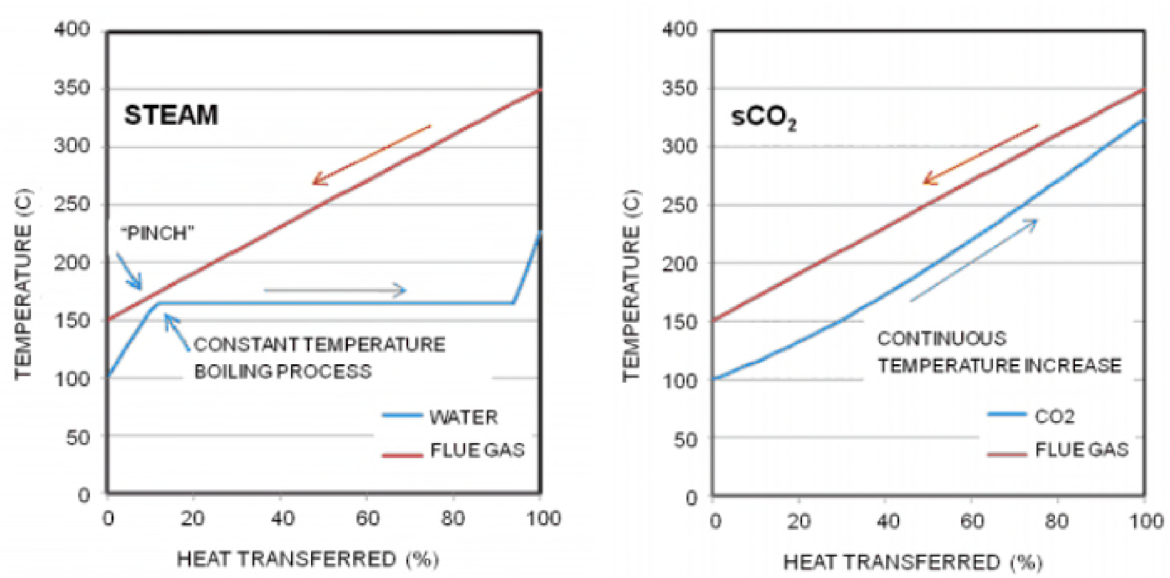

Figure 6. T-Q diagrams showing a two phase situation (left) and a sCO2 situation on right. (Held, 2012)

3.0 STEAM APPLICATION IN LNG LIQUEFACTION

There are several approaches wherein the waste heat from the mechanical drive gas turbines may be used to provide both power and heat for the LNG facility. The full utilization of heat enables a significant improvement of efficiency when low efficiency industrial gas turbine drivers are utilized. A discussion of different options relating to gas turbines and various cogeneration options including ORCs has been studied by Meher-Homji et al (2011). The extent of process heat required is a strong function of the concentration of acid gas contaminants in the feed gas and the process heat required for the AGRU. The required process heat to power ratio for the facility is an important parameter and impacts the type of gas turbine selected, and the heat recovery equipment and approaches used. Aeroderivative engines have higher thermal efficiencies of approximately 40% and this means that the capability to raise heat in an *unfired* HRSG is lower than an industrial gas turbine commonly used in LNG mechanical drive service that have thermal efficiencies around 32-33%. Typical Heat to Power ratios for aeroderivatives are approximately 0.9- 1.0 while those for industrial engines are higher, approximately 1.4-1.5.

To show the thermal design of a steam system, some thermal designs have been created utilizing a LM6000PF engine. The designs include:

- Single Pressure Level – Condensing Steam Turbine
- Dual Pressure Level- Condensing Steam Turbine
- Supplemental fired – Condensing Steam Turbine.

In all cases, the LM6000PF would be the refrigeration compressor driver. It is important to note that if process steam is required, this can be derived from the HRSG itself, but this is not considered in the designs ahead. In all cases sufficient process steam is available with minimal impact on the power cycle.

A thermal design for a combined condensing cycle using a LM6000 gas turbine is presented ahead to show the issues and complexities. The conditions used for these runs include:

- Site ambient conditions: 1.013 bar, 15 °C, 60% RH
- Fuel = $90C_4+10N_2$, supplied @ 25 C
- LHV @ 25C = 41913.78 kJ/kg
- Inlet filter = 10 millibar, Duct & stack = 4.98, HRSG = 17.00 millibar
- Total inlet loss = 10 millibar, Exhaust loss = 21.99 millibar
- G.T. @ 100 % rating
- Air cooled plant

As several companies prefer unfired and single pressure level designs, a thermal design of a LM6000PF at an ambient temperature of 15°C, and using typical fuel is shown in Figure 7. In this single pressure system, approximately 10.5 MW of power can be generated by a condensing steam turbine. This cycle flow schematic shows the key cycle parameters with mass flows M in kg/sec, temperatures T in °C, and Pressures P in Bara. The corresponding T-Q diagram of the heat transfer in the HRSG is shown in Figure 8. The low turbine exhaust gas temperature and the fact that a single pressure level is used results in a low HRSG efficiency and a final stack temperature of 205°C.

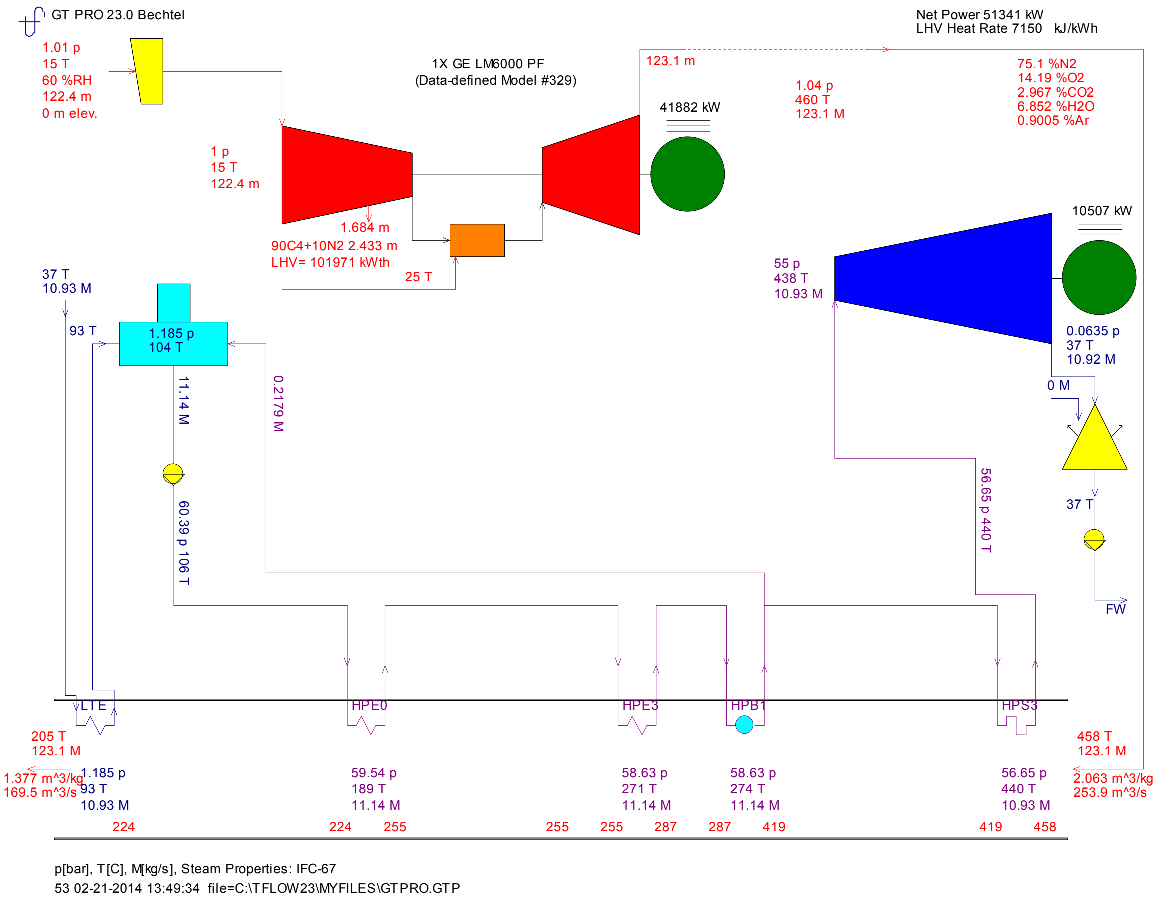

Figure 7. Cycle Flow Schematic - Single Pressure Level Combined Cycle System, with Condensing Steam Turbine.

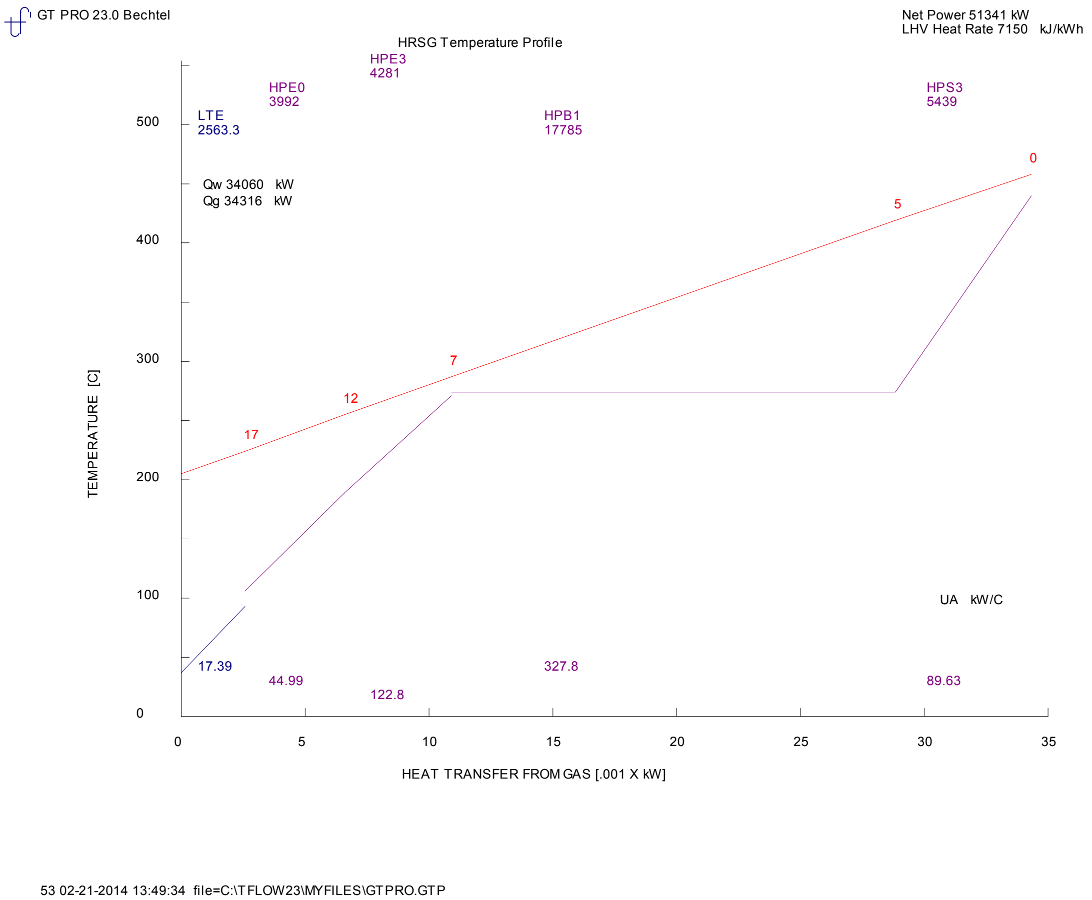

Figure 8. T-Q Diagram for Single Pressure Level Steam System.

A thermal design for a condensing combined cycle with *two* pressure levels is shown in Figure 9. As expected, a greater heat recovery is now attained from the HRSG. The, stack temperature is reduced to 126°C and the steam turbine power increases by approximately 2.2 MW. If supplemental firing to a moderate temperature of 600°C were utilized, the steam turbine power would go up to 21.6 MW, an increase of 11.1 MW. The HRSG stack temperature reduces to 100°C. Relevant cycle parameters are shown in Table 1 below.

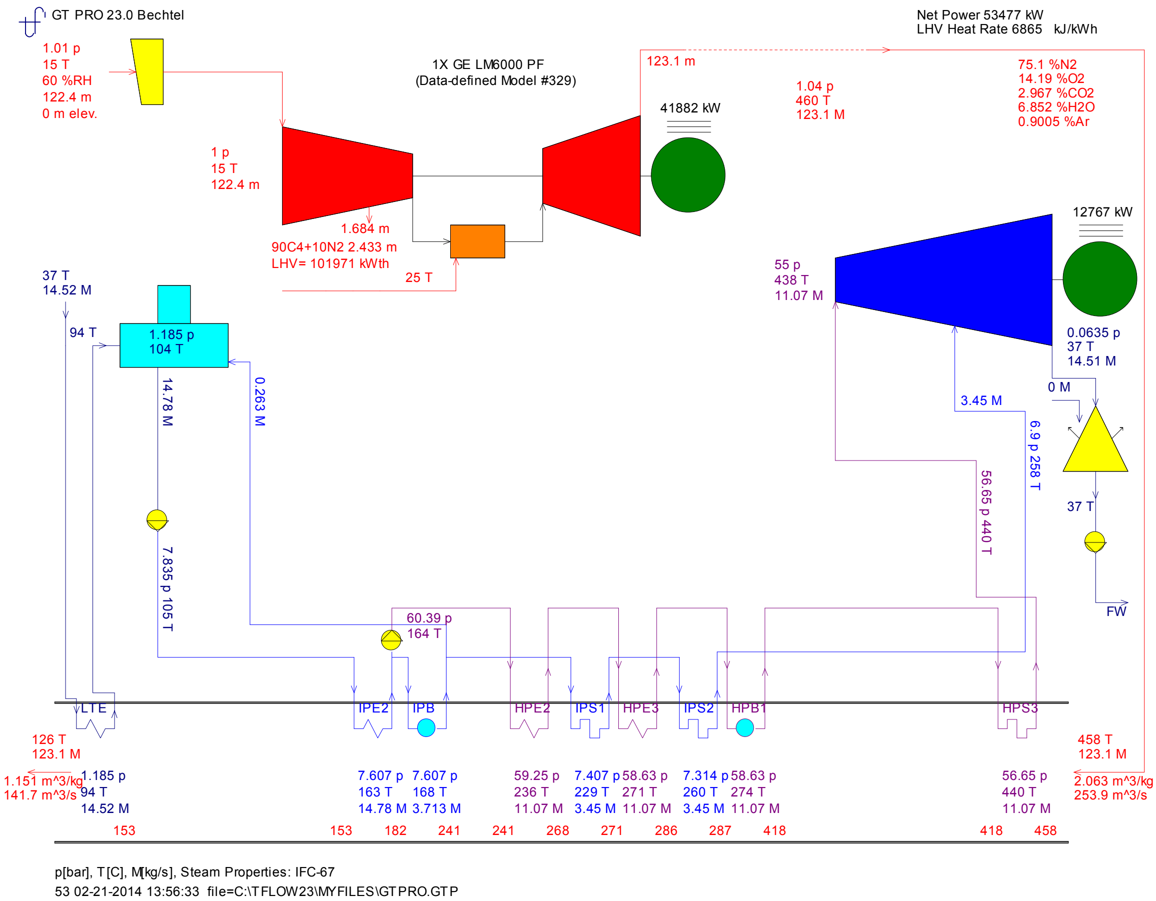

Figure 9. Cycle Flow Schematic - Dual Pressure Level Combined Cycle System, with Condensing Steam Turbine.

	Single Pressure Level Combined Cycle	Dual Pressure Level Combined Cycle	Dual Pressure Level With Supp Fire to 600 C
Net Power, kW	51341	53477	62012
GT Power, kW	41882	41882	41882
ST Power kW	**10507**	**12767**	**21664**
Total Fuel LHV input , kW th	101971	101971	122938[4]
Power Efficiency, net LHV	50.35%	52.44%	50.44
HRSG Efficiency	57.57	74.74	85.29
HRSG stack exit Temp, C	205	126.5	100
NOX emissions, kg/hr	9.617	9.617	21.23
NOx emissions kg/MW-hr	0.1836	0.176	0.3341
CO2 kg/hr	20123	20123	24260
CO2 kg/MW -hr	384.1	368.2	381.8

Table 1. Key Cycle Parameters for Combined Cycle Configurations- unfired SP, unfired DP and fired DP

[4] Supplemental Fuel flow = 23,352 kWth (0.5002 kg/sec)

These designs show the relative complexity of the steam systems compared to a sCO_2 cycle. In a sCO_2 cycle, there would not be a need for a complex HRSG, condensing steam turbine with a large surface condenser, and all the operational complexities relating to water management and chemistry control.

4.0 TECHNO-ECONOMIC STUDY OF sCO_2 POWER GENERATION

4.1 Process Study

A process study was developed for comparing the benefits of using a supercritical CO_2 power generation cycle in place of traditional gas turbine generators to sustain LNG plant power demand.

The basis of the study assumed a feed gas composition with <1% Nitrogen and >95% Methane to the LNG plant. Refrigeration drivers considered were LM6000PF DLE drivers in a 2+2+2 configuration in a cold climate (Average temperature = 11°C). High pressure fuel gas is taken from the HP Methane loop of the COP Optimized Cascade Process for fuelling the main refrigeration turbines as well as the gas turbine generators which provide power to the facility.

Fuel gas that is not burnt in the GTG (when power generation is by sCO_2 cycle) is utilized as LNG production. Various simulation cases were executed to understand the sensitivity of reduced GTG fuel on the LNG plant. Since the fuel rejection from the HP methane loop is also a means of limiting non condensables build up in the methane loop (mainly Nitrogen), the incremental power usage due to the increased nitrogen build up is not realized for lower concentrations of N_2 in feed gas. For significantly higher N_2 concentrations the inherent presence of a Nitrogen Rejection Unit is expected to largely cancel out the inefficiencies associated with lower N_2 purge from the methane loop.

The Basis of the Simulation runs include:

- Driver: LM6000PF DLE (6 Refrigeration Drivers)
- Feed gas N_2: <1%
- Ambient T: 11° C
- Three GTGs (15 MW ISO units) operating at part load and sharing loads representing a N+1 power generation scheme. Total load approximately 22 MW.

Four cases were examined:

- Case 1: Base case- traditional LNG facility with 3 GTGs in operation and GTG Fuel being obtained from the Methane loop. Low N_2 in feed (0.7% N_2)
- Case 2: Base case, no GTGs, utilizing 4 SCO2 cycles each with nominal power generation of 8-10 MW
- Case 3: Base case with sCO_2 power generation, low N_2 (0% N_2)
- Case 4: Base case with sCO_2 power generation, high N_2 (2%N_2)

The reason for examining the different N_2 cases was to examine the impact on LNG production. As less N_2 is rejected into the GTG fuel, it tends to build up in the LNG requiring more power to condense it.

	CASE 1	CASE 2	CASE 3	CASE 4
N_2 Content In FEED, %	0.7	0.7	0	1.94
Normalized Impact on LNG Production	**1.0**	1.007	1.008	1.029

The situation is shown schematically in Figure 10. The benefit to LNG production is dependent on the power margin available within the refrigeration drivers. As can be seen in the figure, if no power margin is available, then LNG production will drop mainly due to the buildup of N_2 and the inability to convert the “saved” fuel into LNG due to power limitations. However, as is the typical case, power margins do exist and this enables some benefit in LNG production. In the case where a NRU exists when N2> 1.5-2% of feed, then the issue of N_2 ceases to exist and benefits in LNG production can be derived.

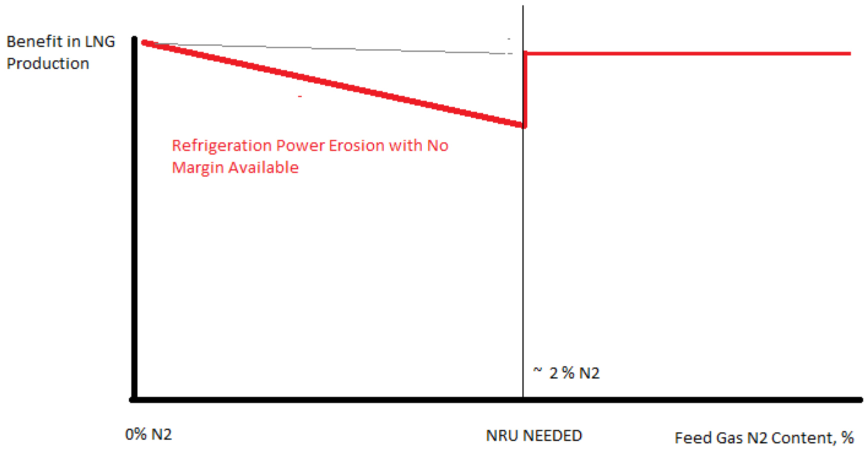

Figure 10. LNG Production Benefits and N_2 Levels.

4.2 Economic Analysis

A NPV type analysis was conducted of the two scenarios-

- Case 1- Traditional power generation
- Case 2- Power generation using sCO2

The following assumptions were used:

- Life 20 years
- Discount Rate 10%
- LNG Cost FOB 8$/MMBTU
- Fuel Cost 3$/MMBTU
- Plant Availability 95%
- CO2 Emission Tax $40/tonne

At an LNG sales price of $8/MMBTU, a fuel cost of $3/MMBTU, and a CO_2 Tax of $40/tonne, the NPV benefit of using a sCO_2 power generation system is of the order of $270 million. The sensitivity of the NPV benefit is shown in Figures 11 through 13 where some parameters have been varied. Clearly the benefit is most sensitive to the LNG price, and is relatively insensitive to the CO_2 tax value. It is also sensitive to fuel cost, as the sCO_2 system will derive power by capturing waste heat thus reducing auto-consumption. For conservativeness the OPEX costs have been considered at parity, even though in reality it estimated to be approximately 5$/MWhr, which is lower than a gas turbine in the size range considered[5].

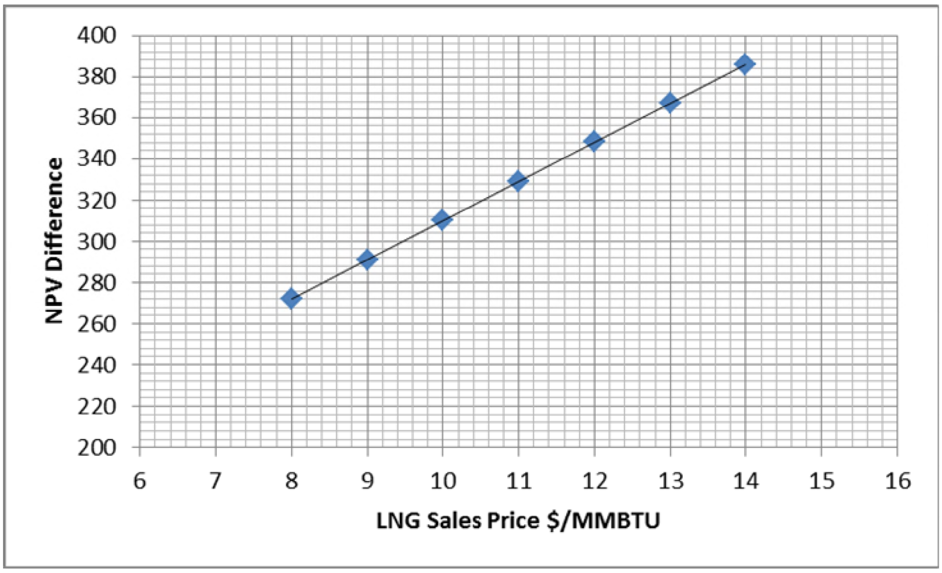

Figure 11. Sensitivity of NPV Differential with LNG Sales Price

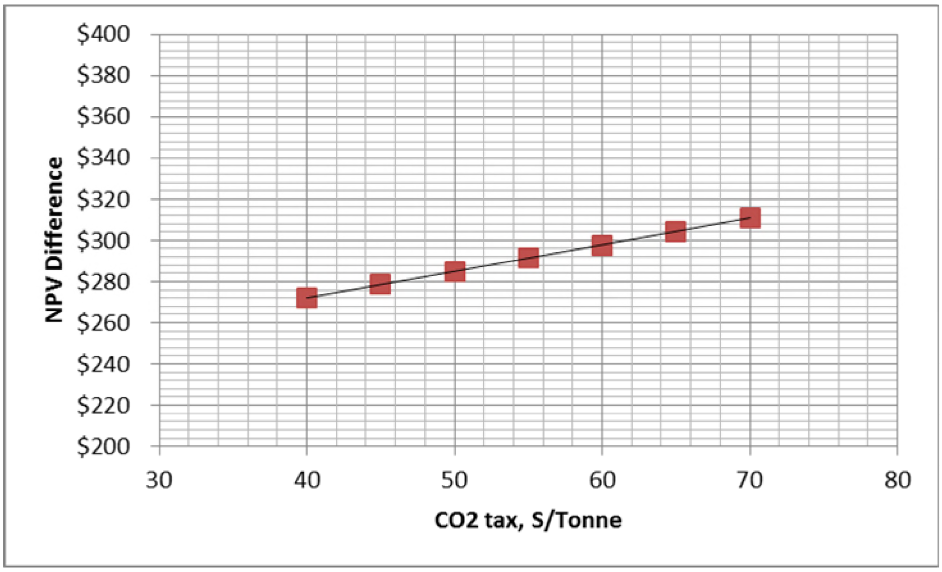

Figure 12. Sensitivity of NPV Differential with CO_2 Tax.

[5] The system is designed for remote operation. It is estimated that compared to a steam system, maintenance costs would be approximately 50%. Compared to a simple cycle gas turbine, the maintenance costs would be approximately 60-75%.

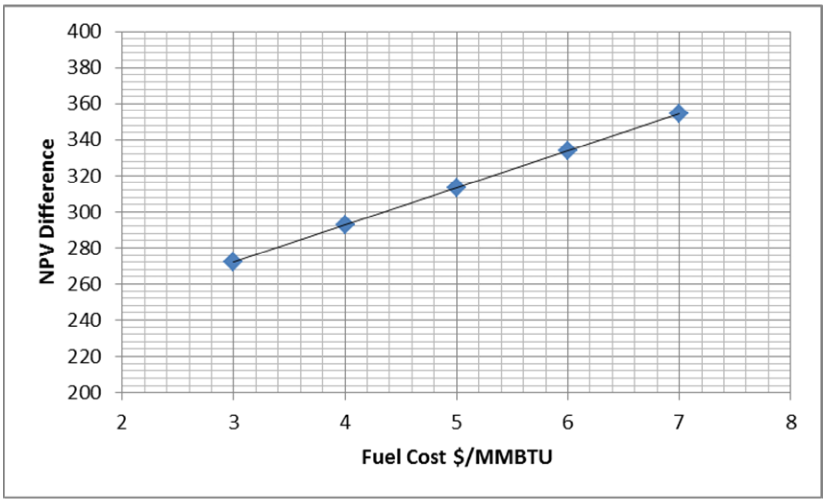

Figure 13. Sensitivity of NPV Differential with Fuel Cost.

In the case studied, the GTG CO_2 saved is of the order of 146,143 tonne/year and the fuel saved is $7.581 million.

The power capability of an Echogen sCO2 system coupled to a LM6000 engine is 13-15 MW at 15°C ambient temperature. In this analysis a much lower power level was considered, so it is possible that with higher power outputs, the economics would look even more attractive.

5.0 SUMMARY

The use of sCO_2 to generate electrical power within the Optimized Cascade Process can be economically and technically viable and attractive. The approach enables a further boost in efficiency of the already high efficiency aeroderivative gas turbines and a reduction in CO_2 emissions. For a typical 5- 6 MTPA facility utilizing 6 x LM6000 gas turbines, heat recovery from four gas turbines would provide more than adequate power and would avoid the complexity and operational challenges of steam systems and water chemistry. Under conditions where appropriate refrigeration power margins are available, the reduced fuel auto-consumption can be converted into LNG. Deployment of this technology would reduce CO_2 emissions and boost LNG production. While the use of sCO_2 is still in a demonstration phase, it is a viable technology for LNG liquefaction plants and is ready for deployment for heat recovery for gas turbines in the range of 25 to 45MW.

The technology is particularly viable with the use of aeroderivative units in relatively cold climates.

Acknowledgements: We gratefully acknowledge the assistance of Dresser-Rand and Echogen Power Systems.

NOMENCLATURE

AGRU	Acid gas removal unit	HP	High Pressure
BPST	Back Pressure Steam Turbine	sCO_2	Supercritical CO_2
CC	Combined Cycle	SP	Single Pressure
CHP	Combined Heat and Power	DP	Dual Pressure
CST	Condensing Steam Turbine	kW_t	Thermal Energy, in kW
DBT	Dry Bulb Temperature	LHV	Lower Heating Value
DLE /DLN	Dry Low Emission	LNG	Liquefied Natural Gas
GT	Gas Turbine	MP	Medium Pressure
H/P	Heat (Steam) / Power Ratio	TIC	Total Installed Cost
HRSG	Heat Recovery Steam Generator		

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