Enhancement of LNG Propane Cycle through Waste Heat Powered Absorption Cooling

A. Mortazavi[1], P. Rodgers[2], S. Al-Hashimi[2], Y. Hwang[1] and R. Radermacher[1]

[1]Department of Mechanical Engineering, University of Maryland, College Park, MD, U.S.A.
[2]The Petroleum Institute, Abu Dhabi, U.A.E.
prodgers@pi.ac.ae

Abstract

This paper presents a study conducted on the potentials of waste heat utilization at a liquefied natural gas (LNG) plant, to enhance propane cycle cooling and efficiency through absorption cooling utilizing hot exhaust gases from gas turbines. Based on the results presented, this approach is promising to enhance cycle performance and efficiency.

1. Introduction

In oil and gas industries waste heat utilization is applied to increase energy efficiency and reduce carbon emission. One of the gas industries is natural gas liquefaction, where natural gas is liquefied in order to reduce its volume for transportation. To permit liquefaction, natural gas needs to be cooled below -160 °C. Consequently for liquefaction considerable amount of energy is required. Hence increasing efficiency will lead to lower carbon emission. On the other hand due to the fact that natural gas is the cleanest fossil fuel and fuel demand is increasing, liquefied natural gas (LNG) market is growing; therefore LNG production capacity should be increased. This can be achieved by enhancing existing LNG plants or building new plants. Waste heat utilization will lead to an increase in efficiency and in some cases production capacity. Di Napoli [1] showed that gas turbine and steam boiler combined cycle plants are more energy efficient and economic than steam boiler cycles for LNG plants.

To utilize waste heat, there are three steps that need to be assessed, namely the identification of:

i. Waste heat sources
ii. Utility requirements (power, steam, cooling, desalinated water, *etc*).
iii. Waste heat utilization technologies suitable to meet utility requirements.

Figure 1 provides an overview for each of these steps in terms of sources, uses and technologies.

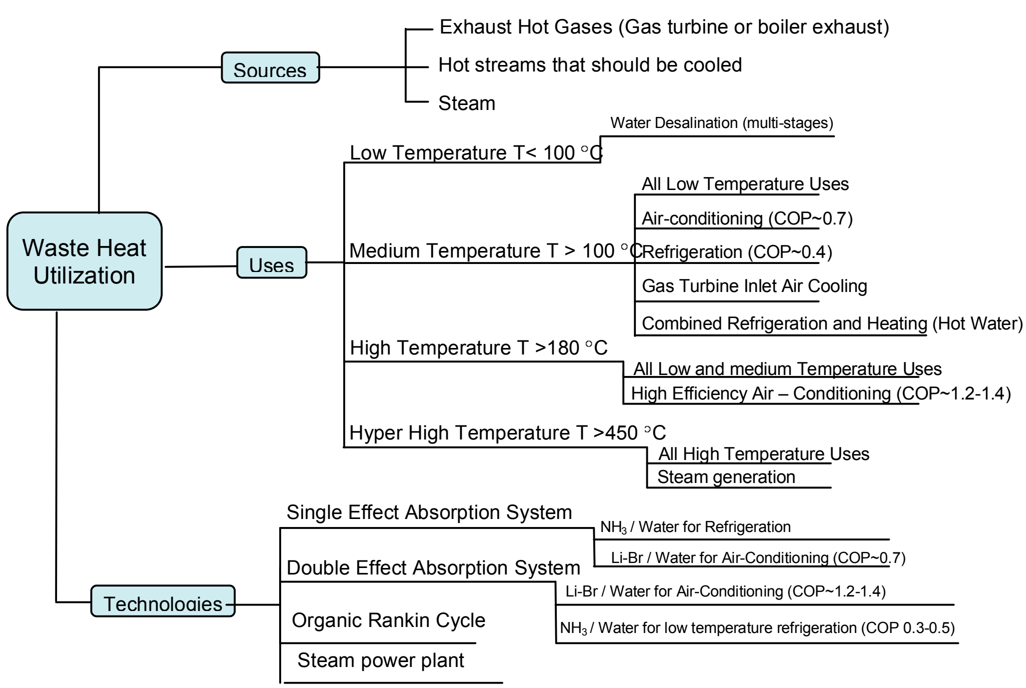


Figure 1. Waste heat utilization evaluation steps in terms of sources, uses, and technologies.

For the purpose of this paper, a waste heat source can be defined as any hot exhaust gases or other streams which should be cooled, and the heat of which is customarily rejected to the ambient. However the temperature (quality), mass flow rate (capacities) and location of such streams and the location of utility requirements play an important role in the feasibility of such waste heat utilization. The main sources of waste heat which are typically available at LNG plants are:

1. Gas turbine exhaust gases
2. Flared gases
3. Boiler exhaust gases

Based on the waste heat temperature and utility requirement, such waste heat could be utilized for:

- Water desalination
- Air conditioning
- Refrigeration
- Gas turbine inlet cooling
- Combined cooling and heating
- Steam generation
- Power generation

In hot climate regions such as the Persian Gulf and the red sea, the seawater temperature during the summer rises to 35 °C. This results in a significant decrease of the COP and production capacity of LNG plants because typically the condensers and desuper heaters of refrigeration cycles are seawater cooled.

Hwang [2] proposed that by utilizing the waste heat of a micro turbine for subcooling the refrigerant of a conventional vapor compression refrigeration system after the condenser it is possible to reduce annual energy consumption of the vapor compression cycle by 12%. This concept can be applied to a LNG plant.

In this paper only the waste heat of gas turbine exhaust gases are considered. The amount of waste heat which can be utilized from gas turbine exhaust gases depends on:

i. Operating condition of the gas turbine; at partial load operation the exhaust temperature is lower than at full load.
ii. The temperature to which hot gases can be cooled. The lower the temperature, the higher amount of available waste heat.

Two solutions were considered for waste heat utilization of the gas turbine exhaust gases, namely:

i. Sub-cooling propane after the condenser.
ii. Reducing propane cycle condenser pressure.

In this paper, to predict the available amount of waste heat from gas turbine exhaust, at a range of operating conditions and to assess the effects of possible enhancements to the propane cycle, ASPEN PLUS software [3] is used. Before outlining the modeling methodology a review of the LNG cycle under analysis is presented so as to identify how waste heat solutions could be applied to enhance efficiency and capacity during the summer months.

In this paper an LNG plant operating in the Persian Gulf, based on APCI liquefaction cycle, is considered. A schematic diagram of the LNG process is given in Figure 2.

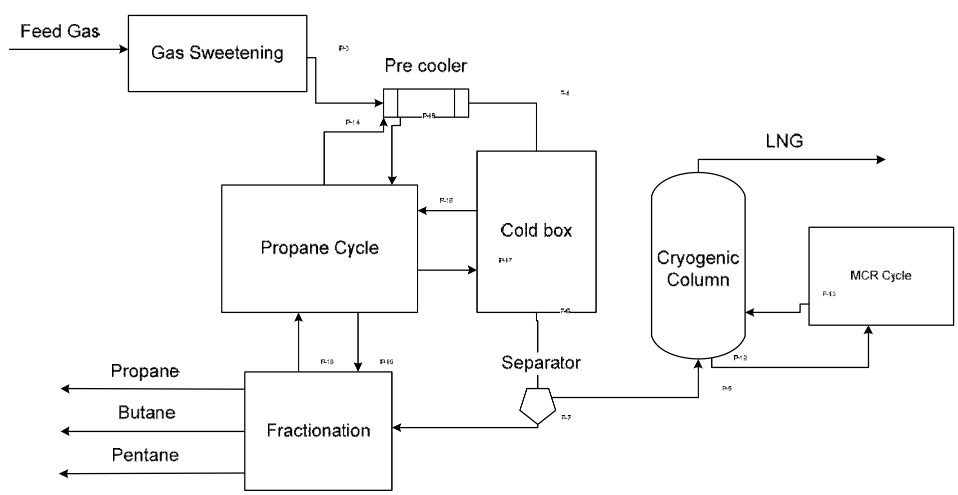


Figure 2. Schematic illustration of an LNG plant-process.

In Figure 2 the feed gas is passed through the gas sweetening plant for H_2S, CO_2, H_2O and Hg removal. Then by passing through the pre-cooler and cold box its temperature drops to about -30 °C and some components condense. In the separator, the remaining gas and condensate are separated. The condensate is being sent to the fractionation unit, where it will be separated to propane, butane and pentane (and heavier hydrocarbons). The gas is further cooled in the cryogenic column to below -161 °C and liquefied. The propane cycle provides the required cooling to the precooler, cold box and fractionation plant. The MCR (Mixed Component Refrigerant) cycle supplies the cooling demand of the cryogenic column.

2. Modeling Methodology

The ASPEN PLUS software was employed for simulating the thermodynamic cycles of both the gas turbine and propane cycle, with the software suitable for steady state process analysis. An ASPEN model is based on blocks corresponding to unit operations such as turbines and compressors as well as chemical reactors such as combustion chambers. By interconnecting the blocks using material (fluid), work and heat streams a complete process flow sheet can be constructed. ASPEN has a range of both databases containing thermodynamic and chemical data for a wide variety of chemical compounds and thermodynamic models for simulation of thermodynamic systems. Simulation is performed by specifying:

1. Flow rates, compositions and operating conditions of the inlet streams.
2. Operating conditions of the blocks used in the process, e.g. temperature and pressure.
3. Operating heat and/or work inputs into the process.

Based on these input data, ASPEN calculates flow rates, compositions and state conditions of all outlet material streams, as well as the heat and work output.

For modeling the property of substances both in the gas turbine and propane cycle models, the Peng-Robinson-Boston-Mathias equation of state property method is used. Convergence tolerance for all ASPEN models was set to 1×10^{-4}.

3. Waste Heat Utilization Analysis of Gas Turbines Exhaust Gases

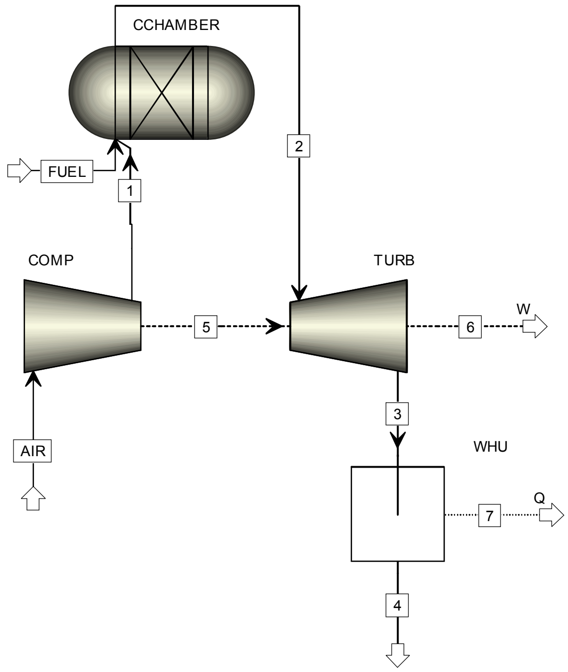
The gas turbine modeled in this study is a GE MS 5001/P(1972-1978) [4,5] which is used for electrical power generation with its specifications given in Table 1. The ASPEN model of the gas turbine is presented in Figure 3. In Figure 3 streams 1 - 4, 5 - 6 and 7 represent material, work and heat respectively. The combustion chamber (CCHAMBER) is modeled by performing chemical and phase equilibrium calculations using the Gibbs energy minimization. For modeling the gas turbine, compressor, turbine and generator efficiencies should be identified. These efficiencies are derived from vender data sheets. Model assumptions are:

1. Compressor isentropic efficiency is 0.83
2. Turbine isentropic efficiency is 0.85
3. Generator efficiency is 98%
4. Compressor mass flow rate is constant and equal to 118.046 kg/s
5. Compressor inlet and turbine outlet pressure drops are neglected.

Table 1. Gas turbine specifications [4,5].

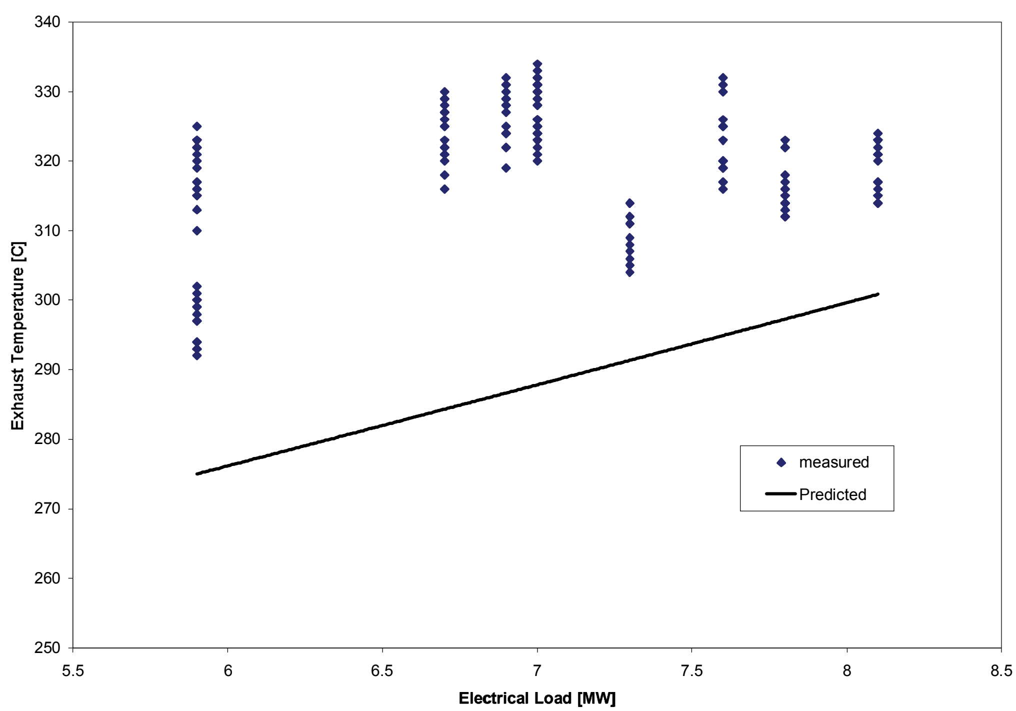
ISO Rated Power [MW]	Firing Temperature [°C]	Air Flow [Ton/hr]	Exhaust Temperature [°C]	Heat Rate [Btu/KW-hr]	Efficiency (%)	Pressure ratio
24.6	943	425	484	12140	28.1	10.5

To assess the predictive accuracy of the ASPEN model, turbine exhaust temperature was measured at the exhaust of the gas turbine by 18 temperature gauges which were installed by the vendor. A comparison of measured and model predicted exhaust temperatures are given in Figure 4 for different electrical loads at 21 °C, 27 °C and 40 °C ambient temperatures. As observed in Figures 4 at a given electrical load, predicted exhaust temperatures are lower than corresponding measured temperatures. Therefore predicted available waste heat quantities in this study can be considered as conservative for all operating conditions under analysis.

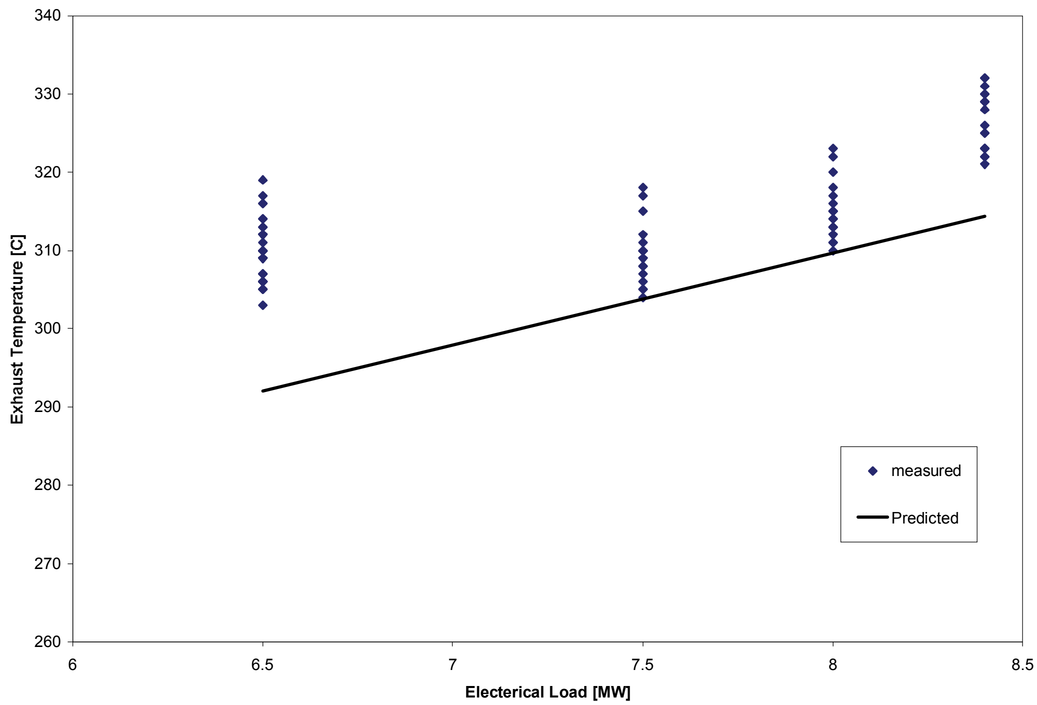


Stream	Description
Air	Inlet air to compressor
Fuel	Inlet fuel to combustion chamber
1	Compressor outlet
2	Turbine inlet
3	Turbine exhaust
4	Heat recovery unit exhaust
5	Compressor power
6	Turbine mechanical output
7	Available waste heat

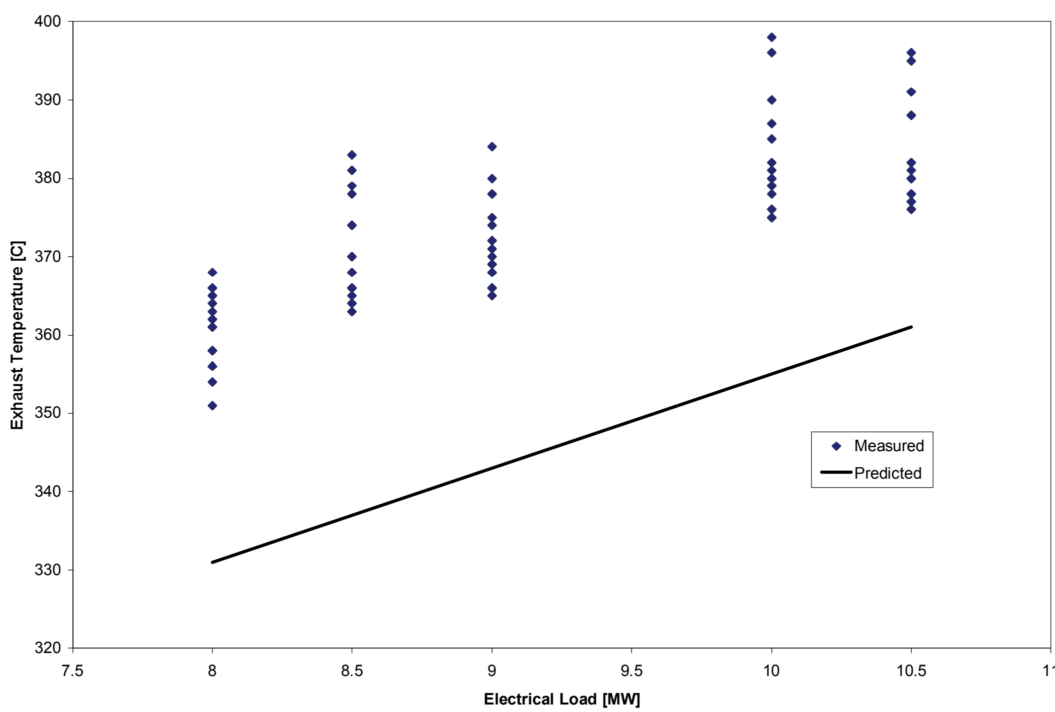
Figure 3. ASPEN block diagram of the gas turbine model.



a) Ambient air temperature = 21°C



b) Ambient air temperature = 27°C



c) Ambient air temperature = 40°C

Figure 4. Predicted and measured gas turbine exhaust temperatures at different electrical loads as a function of ambient air temperature.

Possible reasons why the predicted exhaust temperatures may be lower than the measured include:

1- The gas turbine under analysis is approximately 30 years old and its performance may have degraded even though regular maintenance has been applied. Consequently its efficiency may be lower which results in higher exhaust temperature than modeled. As shown in Figure 5, a reduction in the efficiency of compressor or/and turbine will result in an increase of the turbine exhaust temperature.
2- The generator efficiency may be lower than 98% owing to the fact that the generator is 30 years old. Hence for a given electrical load, a higher amount of mechanical work may be needed. Therefore, the exhaust temperature is higher for a given electrical output. As shown in Figure 5, a reduction in generator efficiency will result in an increase of the gas turbine exhaust temperature.

To assess the prediction discrepancies, a model sensitivity analysis was performed individually and collectively for:

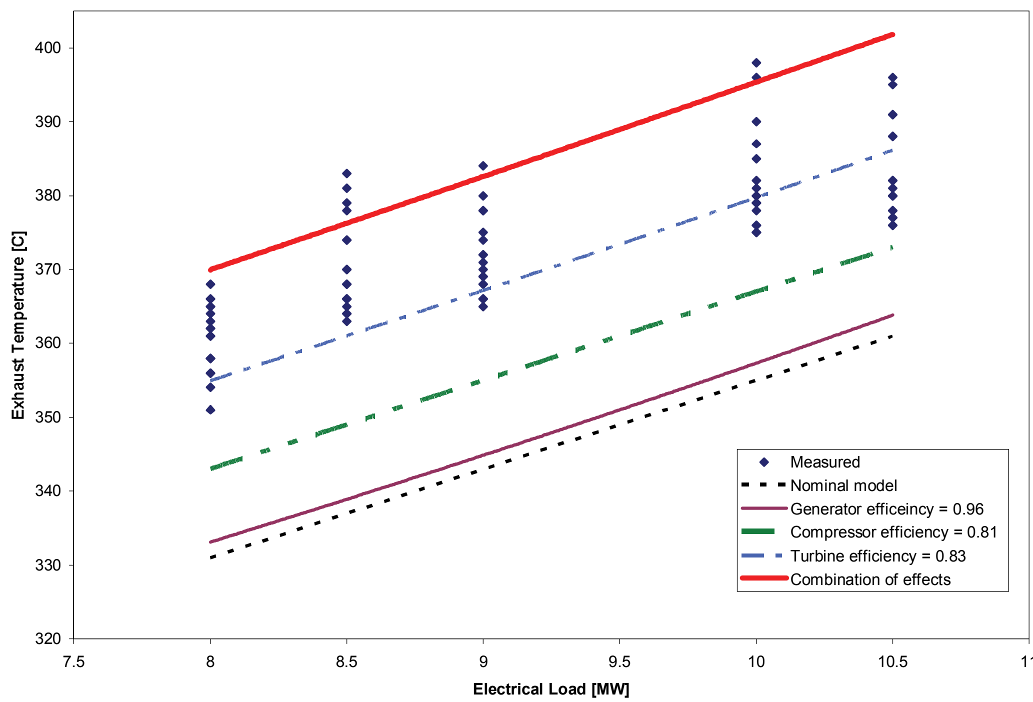
1. Compressor efficiency = 0.81-0.83
2. Turbine efficiency = 0.83-0.85
3. Generator efficiency = 0.96-0.98

These variables are considered independently and collectively with corresponding predictions given in Figure 5 for different electrical loads at 40 °C ambient temperature. Though not presented here similar trends were observed at 21 °C and 27 °C ambient temperatures.

As illustrated in Figure 5, the turbine and compressor efficiency have a considerable impact on gas turbine exhaust temperatures with the turbine efficiency having a higher impact. Generator efficiency has the least effect. It should be noted that by increasing the electrical load the generator efficiency impact on the exhaust temperature will increase, but it is not significant until 11 MW.

Considering that compressor, turbine and generator efficiencies may not be constant and may be degraded, it was decided to use the nominal values; 0.83, 0.85 and 0.98 as compressor, turbine and generator efficiency respectively for ensuring conservative amount of predicted available waste heat quantities.

Although no gas turbine exhaust temperature measurements were available at 45 °C ambient air temperature, confidence is gained that predicted gas exhaust temperatures would be conservative based on the trend observed in Figure 4.



Note: Ambient temperature = 40°C.

Figure 5. Prediction sensitivity of gas turbine exhaust temperature as function of compressor, turbine and electrical generator efficiency at different electrical loads.

For evaluating the amount of available waste heat, a heater block is placed after the turbine as indicated in Figure 3. The predicted amount of available waste heat for different gas turbine operating conditions and the minimum utilization temperature are given in Table 2 for the individual gas turbine simulated above.

Table 2. Estimated waste heat amounts for different operating conditions of a single gas turbine at an ambient air temperature of 45 °C.

Gas turbine operation condition	Minimum exhaust gas utilization temperature [°C]	Air mass flow rate [kg/s]	Turbine exhaust temperature [°C]	Turbine power generation (mechanical) [MW]	Available amount of waste heat [MW]
Part load	110	118.046	388	12.3	35.2
Part load	200	118.046	388	12.3	24
Full load	110	118.046	489	20.9	49.1
Full load	200	118.046	489	20.9	37.8

4. Proposed Waste Heat Utilization Solutions

Gas turbine exhaust gases can be utilized by absorption chillers for the production of cooling capacity. Hot gases with temperatures higher than 110°C can run a single effect absorption chiller utilizing lithium bromide/water as the working pair. Single effect absorption cycles have a COP value of typically 0.7 [6], which infers that the amount of cooling that they can provide is approximately 70 % of the heat supplied. Hot gases with temperatures above 180°C can run a double effect absorption chiller. Double effect absorption cycles have a COP of about 1 to 1.2 [6].

The estimated amount of cooling which can be generated by utilizing waste heat of one gas turbine with a single effect absorption cycle, double effect absorption cycle or combination of single and double effect are shown in Table 3 for different operating conditions and different minimum waste utilization temperatures. For the combination of single and double effect absorption chillers it is assumed that the double effect absorption chiller utilizes the waste heat down to 180 °C and single effect utilize the waste heat between 180°C to 110°C.

Table 3. Estimated cooling capacities of different types of absorption cycles at different gas turbine operating conditions and minimum waste heat utilization temperatures.

Turbine power output [MW]	Minimum waste heat utilization temperature [°C]	Estimated cooling capacity [MW]		
		Single effect	Double effect	Combined single and double effect
12.3	110	24.6	N/A	31.8
12.3	200	16.8	24	N/A
20.9	110	34.4	N/A	45.7
20.9	200	26.5	37.8	N/A

The capacity of combined single and double effect absorption cycles is not predicted for the cases with minimum utilization temperature of 200°C, due to the fact that the combined single and double effect absorption cycle is suitable for minimum utilization temperatures of less than 180°C.

Absorption chillers are able to produce chilled water, which can be employed either for:

1. Sub-cooling the propane after the condenser
2. Reducing the propane cycle condenser pressure by precooling the cooling water

These proposed strategies are outlined as follows.

4.1 Sub-cooling Propane after the Condenser

The propane cycle shown in Figure 6 was modeled using ASPEN plus. The model is based on the Process Flow Diagrams (PFD) and data sheets that were provided by the LNG facility. In the model, the seawater temperature was specified to be 35°C, with all pressures, mass flow rates and pressure drops specified as per data sheets and PFDs. Sub-cooling the propane after the condenser will increase both the efficiency and the capacity of the propane cycle. As propane with lower temperature has lower vapor quality after the expansion valve, which means a larger amount of liquid for evaporation, and therefore higher capacity. Sub-cooling propane after the condenser requires an additional heat exchanger to be installed downstream of the condenser.

The accuracy of ASPEN temperature predictions are assessed in Table 4 at various locations of the propane cycle, as defined in Figure 6. Good agreement exists at all locations between PFD and model predictions with maximum discrepancy of 2°C. This provides sufficient confidence in the use of the ASPEN model to assess both the effect of modifications on the propane cycle in terms of adding a subcooler after the condenser or when reducing condenser pressure. The predicted effects of adding a sub-cooler after the condenser of the propane cycle are shown in Table 5. As observed, by enhancing propane cycle with the sub-cooler the COP and cooling capacity are increased by 13% and 23% respectively.

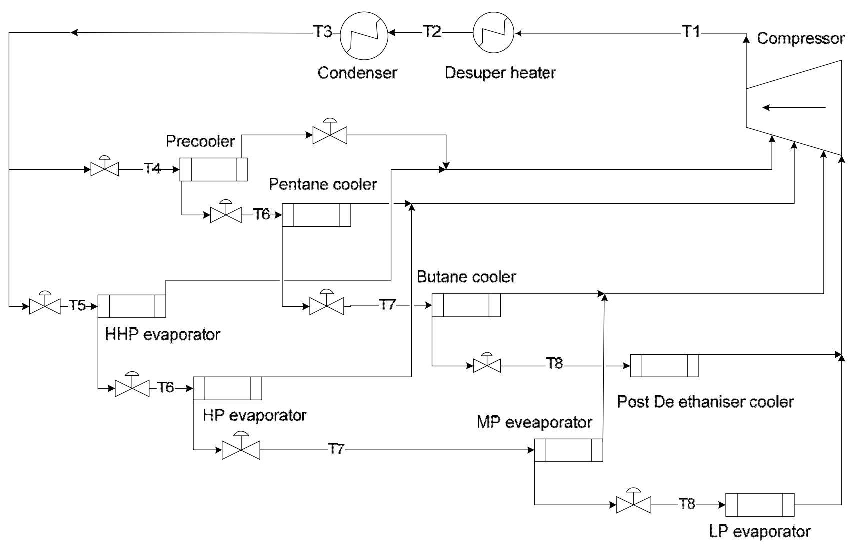


Figure 6. Propane cycle block diagram.

Table 4. Comparison of PFD temperatures with Aspen model predicted temperatures for the propane cycle for sea water cooling at 35°C.

Location	PFD Temperature [°C]	Model Predicted Discrepancy [°C]
Compressor discharge (T1)	59.4	+1.9
Desuper heater outlet (T2)	44.2	+0.3
Condenser outlet (T3)	43.3	---
Precooler (T4)	21.5	---
HHP evaporator (T5)	12.9	---
HP evaporator and pentane cooler (T6)	-2.0	+0.4
Mp evaporator and Butane cooler (T7)	-15.9	-0.2
LP evaporator and post de ethaniser cooler (T8)	-28.8	+0.6

Table 5. Predicted effects of adding sub-cooler 3on propane cycle parameters.

Parameter	Original propane cycle	Inclusion of sub-cooler after condenser	Performance change
Total COP	3.042	3.44	+13%
Total cooling capacity [MW]	104.86	128.91	+22.9
Compressor work [MW]	34.473	37.475	+8.7%
Propane mass flow rate at condenser inlet [ton/hr]	1490	1490	---
Compressor discharge temperature [°C]	61.3	62.6	N/A
Compressor discharge pressure [kPa]	1540	1540	---
Desuper heater cooling load [MW]	14.451	15.632	+8.2%
Condenser cooling load [MW]	124.88	124.88	---
Sub-cooler cooling load[MW]	N/A	25.872	N/A
Propane temperature at the inlet of liquid receiver [°C]	43.3	22	N/A

4.2 Reducing Propane Cycle Condenser Pressure

By reducing the temperature of the condenser and desuper heater cooling water, the condenser pressure decreases. Hence both the efficiency and capacity of the propane cycle will increase as reducing cooling water temperature will permit propane to condense at lower temperature and pressure. Consequently compressor work will decrease per unit mass of circulating refrigerant. Moreover, as mentioned previously, by decreasing the propane temperature, the quality of propane after the expansion valve decreases which leads to an increase in capacity. A closed chilled water loop was assumed for cooling the propane cycle condenser and desuper heater. As it was found by ASPEN modeling, it can be observed in Figure 7, lowering sea water temperature from 35°C to 30°C needs more amount of cooling than condenser and desuper heater cooling load. The predicted effects of the chilled water temperature on different parameters of the propane cycle are shown in Figures 8 to 10. In all simulations the propane mass flow rate was assumed to be constant.

It is observed in Figure 8 that by decreasing the condenser cooling water temperature the COP of propane cycle increases by 86% by reducing the cooling water by 21°C.

As observed in Figure 9 the cooling capacity increases by reducing condenser and desuper heater cooling water temperatures, with 23% increase by reducing the cooling water by 21°C. In addition corresponding compressor work decreases by 27%. In figure 10 it is observed that both compressor discharge pressure and temperature decrease by reducing the condenser and desuper heater cooling water temperature. Based on 14°C cooling water temperature of condenser and desuper heater, Table 6 indicates potential enhancements that may be achieved. Both COP and cooling capacity of propane cycle are estimated to increase by +68% and +23% respectively. In addition it is estimated that the compressor work decreases by 27%; which would reduce compressor steam consumption by 35 ton/hr. At least 154 MW of waste heat is required for maintaining the amount of cooling which is needed for cooling the condenser and desuper cooling. This is approximately equal to the waste heat of four gas turbines running near their full load and with minimum gas turbine exhaust gas utilization temperature of 110°C.

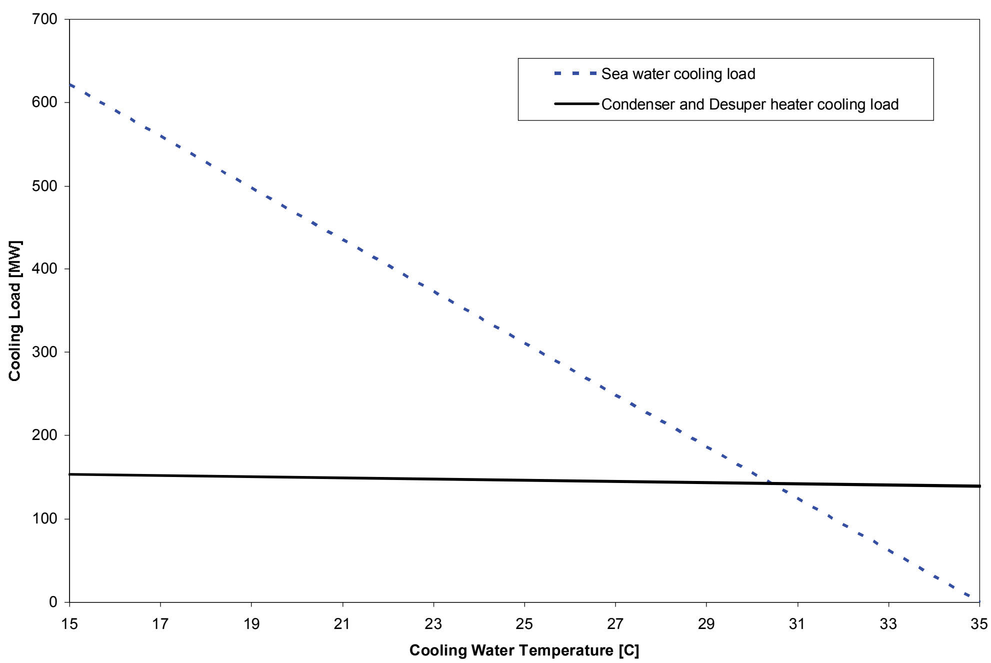


Figure 7. Predicted amount of cooling needed for cooling sea water from 35 °C to condenser cooling water temperature and condenser and desuper heater cooling load.

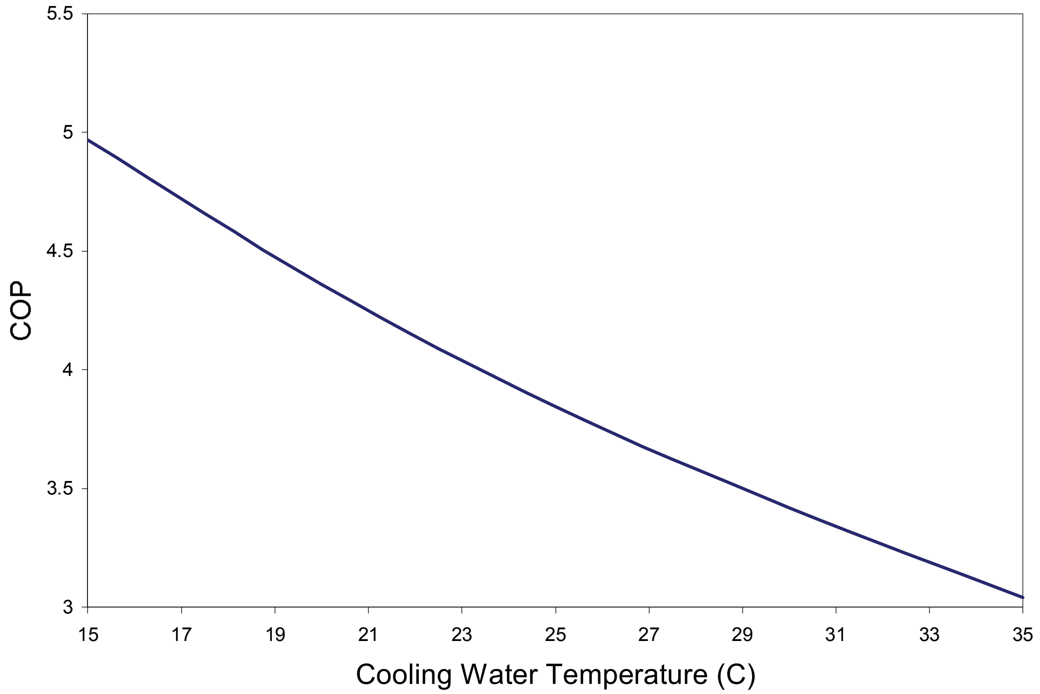


Figure 8. Predicted effect of cooling water temperature on total COP.

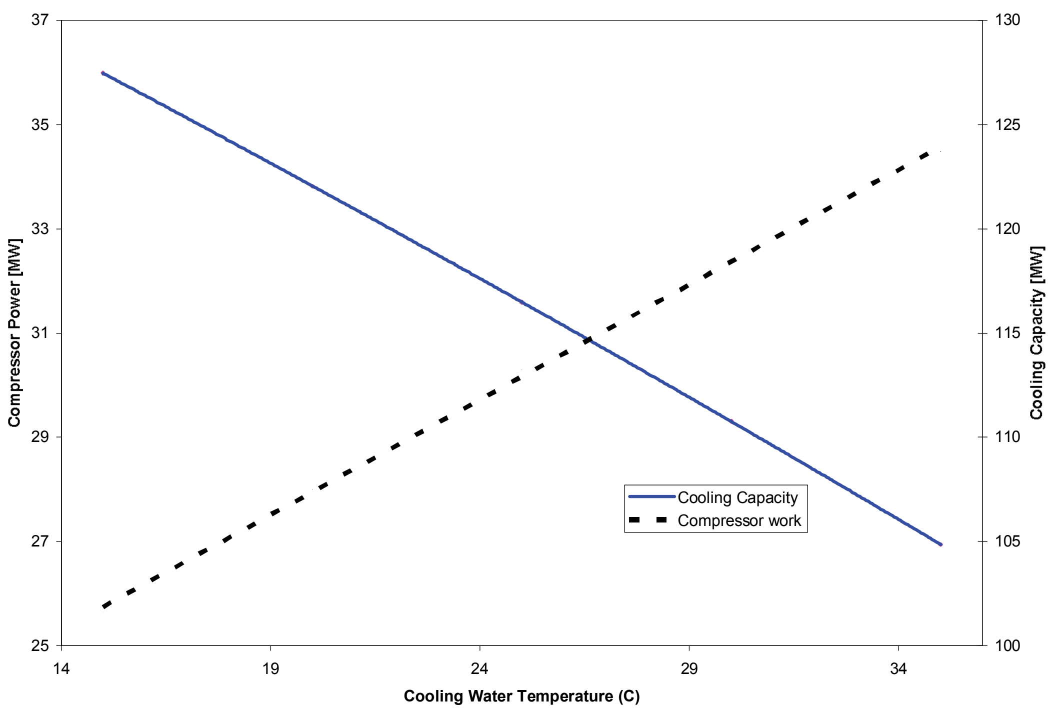


Figure 9. Predicted effect of cooling water temperature on total cooling capacity and compressor power.

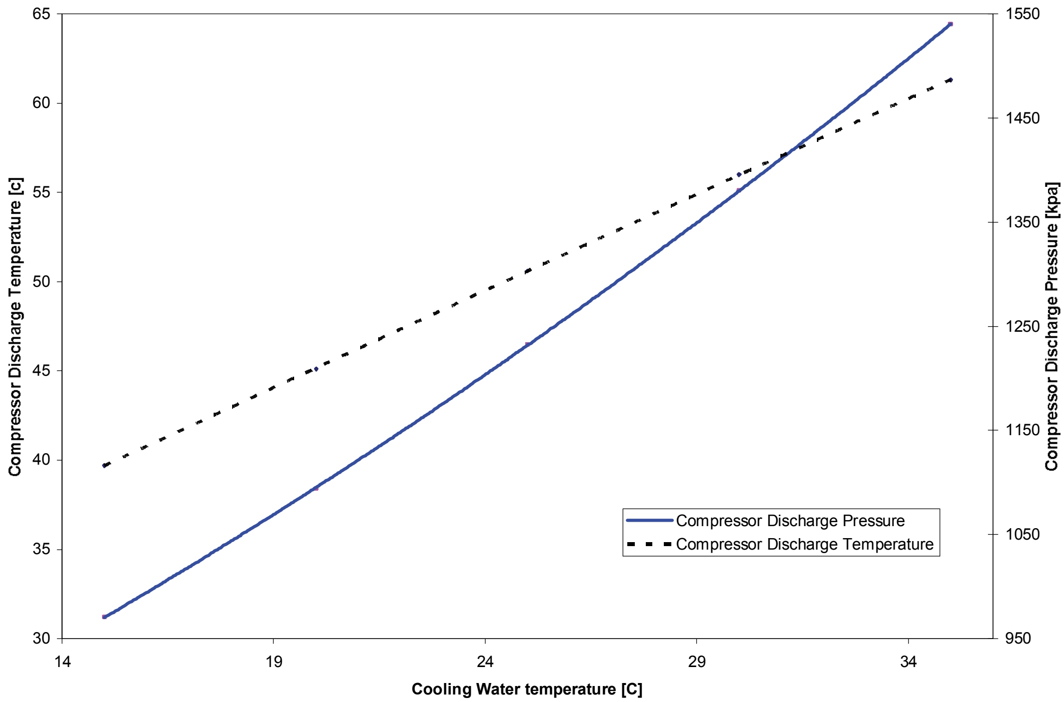


Figure 10. Predicted effects of cooling water temperature on compressor discharge pressure and temperature.

Table 6. Predicted effects of lowering the cooling water temperature of propane cycle condenser and desuper heater.

Parameter	Original propane cycle	Modified cycle*	Percentage Change
Total COP	3.04	5.11	+67.9
Total cooling capacity [MW]	104.86	128.52	+22.6
Compressor work [MW]	34.473	25.161	-27
Propane mass flow rate at condenser inlet [Ton/hr]	1490	1490	0
Compressor discharge temperature [°C]	61.3	38.6	N/A
Compressor discharge pressure [kPa]	1540	947.3	-38.5
Total condenser and desuper heater cooling load [MW]	139.33	153.68	+10.3
Cooling water temperature [°C]	35.0	14.0	N/A

* Note: Modified system refers to the system which uses waste heat for reducing condenser pressure.

5. Conclusions

In this paper the potential sources of waste heat were reviewed at an APCI LNG plant, with suitable technologies proposed for waste heat utilization. Two proposals were made, namely:

1. Utilizing exhaust gases of gas turbines for sub-cooling propane after condenser.
2. Utilizing exhaust gases of gas turbines for reducing propane cycle condenser pressure.

Proposal 1 may be financially the cheapest to implement, with proposal 2 requiring higher capital investments. However by applying proposals 2, potential benefits are estimated to be:

i. Propane cooling capacity could be increased by approximately 23%.
ii. Increase in propane cycle efficiency by 68%.

6. References

1. Di Napoli, R. N., Gas Turbine Prove Effective as Drivers for LNG Plants, *Oil and Gas Journal*, Vol. 78, No. 31, 1980, pp 47-52.
2. Hwang, Y., Potential Energy Benefits of Integrated Refrigeration System with Microturbine and Absorption Chiller, *International Journal of Refrigeration*, Vol. 27, No 8, 2004, pp. 816-829.
3. ASPEN Tech, http://www.aspentech.com/products/product.cfm?IndustryID=0&ProductID=69, last access on July 17, 2007.
4. Daiber, P.C., GER4196 - Performance and Reliability Improvements for the MS5001 Gas Turbines, GE Power Systems, http://www.gepower.com/prod_serv/products/tech_docs/en/gas_turbines.htm, last access on June 10, 2007.
5. GE Energy Gas Turbines, Gas Turbine Performances http://www.gepower.com/businesses/ge_oilandgas/en/downloads/gas_turb_perf.pdf, last access on October 29, 2007.
6. Herold, K., Radermacher, R., and Klein, S. A., 1996, Absorption Chillers and Heat Pumps, CRC Press.
7. Zheng, L., and Furimsky, E., ASPEN Simulation of Cogeneration Plants, *Energy Conversion and Management*, Vol. 44, No 11, 2003, pp. 1845-51.

Author Biographies

Mr. Amir Mortavazi is currently pursuing his Ph.D. in Mechanical Engineering at the University of Maryland. His current research is in modeling APCI LNG plants for waste heat utilization. He graduated from The Sharif University of Technology with a B.S. in Mechanical Engineering.

Dr. Peter Rodgers, Ph.D., is Associate Professor of Mechanical Engineering at The Petroleum Institute, U.A.E. He has extensive experience in thermofluid modeling and experimental characterization. His current research activities are focused on waste heat utilization in the oil and gas industry; the development of polymeric heat exchangers for sea water cooling applications; computational fluid dynamics; electronics reliability; and engineering education. He is presently a member of several international conference program committees, and serves as program co-chair for both EuroSimE 2009 and Energy 2030. He has authored or co-authored over 60 journal and conference publications.

Dr. Saleh Al Hashimi, Ph.D., an Assistant Professor in the Department of Chemical Engineering at The Petroleum Institute, United Arab Emirates. He has expertise in mathematical modeling, catalysis and waste heat management. He has been interested in applying novel systems to the petroleum industry to make better use of the waste heat generated. Some of his recent publications in this area focus on crude oil stabilization and polycarbonate plants.

Dr. Yunho Hwang, Ph.D., is a Research Associate Professor in the Department of Mechanical Engineering at the University of Maryland. His research focuses on developing comprehensive information for the detailed physics of transport processes, new cost-effective test methods, and innovative components and system. He is responsible for Alternative Cooling Technologies and Applications Consortium (ACTA) that is sponsored by the industry, government and research institutions.

Dr. Reinhard Radermacher, Ph.D., is a Professor in the Department of Mechanical Engineering at the University of Maryland. He holds an M.S. and Ph.D. in Physics from the Munich Institute of Technology. Dr. Radermacher is an internationally recognized expert in heat transfer and working fluids for energy conversion systems, including heat pumps, air-conditioners, and absorption chillers.