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World's First Aeroderivative Based LNG Liquefaction Plant – Design, Operational Experience and Debottlenecking

Cyrus B. Meher-Homji
Bechtel Fellow & Senior Principal Engineer
Turbomachinery -LNG Technology Group
Bechtel Corporation, USA

David Messersmith
Manager, LNG Technology & Services Group
Bechtel Corporation, USA

Hans P. Weyermann
Principal Rotating Equipment Engineer
ConocoPhillips Company, USA

Gary Richardson
Sr. Rotating Equipment Engineer
Darwin LNG
Australia

Paul Pattrick
Maintenance Team Leader
Darwin LNG
Australia

Fernando Roberto Biagi
LNG Platform Manager
GE Oil and Gas
Florence Italy

Francesco Gravame
Project Manager, Upgrades
GE Oil and Gas
Florence, Italy

ABSTRACT

The Darwin LNG Facility is the world's first liquefaction facility to utilize high efficiency aeroderivative gas turbines for its refrigeration compressors. The plant's design, startup, successful operation for over four years, upgrade, and debottlenecking are described in this paper. The application of aeroderivative engines allows a significantly lower CO_2 footprint of 20-30% compared to the use of simple cycle industrial (heavy duty) gas turbines. This paper will cover the design of all of the turbomachinery, testing of machinery, startup, operational experiences, and debottlenecking activities in which the engines were upgraded. The plant was successfully commissioned and the first LNG cargo was shipped on February 14, 2006. Debottlenecking activities were completed in 2010.

INTRODUCTION AND OVERVIEW

Market pressures for thermally efficient and environmentally friendly LNG plants and the need for high plant availability have resulted in the world's first application of high performance aeroderivative gas turbines for the 3.7 MTPA Darwin LNG plant. Details regarding the importance of turbomachinery efficiency in the LNG process has been provided by Ransbarger (2007). The plant was successfully commissioned several months ahead of schedule and the first LNG cargo was supplied on February 14, 2006. This paper will describe the philosophy leading to the world's first aeroderivative based gas turbine plant, details of turbomachinery selection and testing, operating experience and debottlenecking activities that occurred in 2010.

The plant is a nominal 3.7 million tonne per annum (MTPA) capacity LNG plant at Wickham Point utilizes the ConocoPhillips *Optimized Cascade*® LNG Process[1]. It is located in Darwin Harbor, Northern Territory, Australia, and is connected via a 500-km, 26" subsea pipeline to the Bayu-Undan offshore facilities. The Bayu-Undan Field was discovered in 1995 approximately 500 kilometers northwest of Darwin, Australia in the Timor Sea. (See Figure 1). This field contains 3.4 TCF gas and 400 MMbbls of recoverable condensate and LPG.

The facility initially used six PGT25+ gas turbines with high speed power turbines for the three refrigeration services, with two drivers for propane refrigeration, two drivers for ethylene refrigeration and two drivers for methane refrigeration. This engine uses the LM2500+ gas generator coupled to a high speed power turbine (HSPT). The facility is the first LNG plant to use media based evaporative cooling power augmentation for the six PGT25+ engines. The significant improvement in thermal efficiency compared to a traditional industrial gas turbine, results in a reduction of fuel consumption which reduces greenhouse gas in two ways. First, there is a reduction in CO_2

[1] Optimized Cascade services are provided by ConocoPhillips Company, Phillips LNG Technology Services Company and Bechtel Corporation via a collaborative relationship with ConocoPhillips Company. Optimized Cascade, the Optimized Cascade logo, ConocoPhillips and its logo are trademarks of ConocoPhillips Company. Bechtel and its logos are trademarks of Bechtel Group Inc.

emissions due to a lower quantum of fuel burned. The second greenhouse gas benefit results from a reduction in the total feed gas required for the same LNG production. The feed gas coming to the facility contains carbon dioxide, which is removed in an amine system prior to LNG liquefaction and is released to the atmosphere. The reduction in the feed gas (due to the lower fuel gas requirement) results in a reduction of carbon dioxide emissions from the unit.

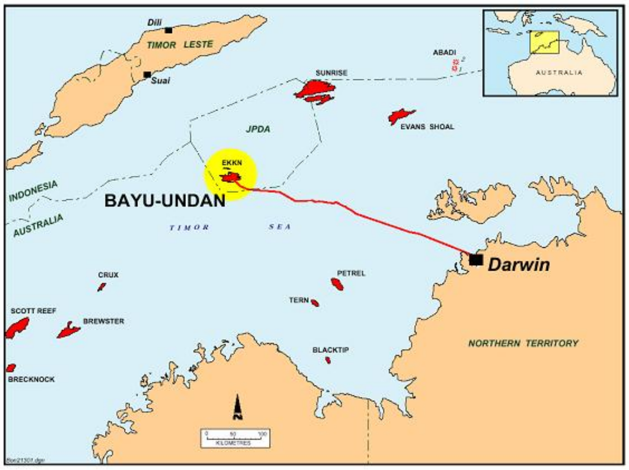

Figure 1. Bayu-Undan Gas Field and the Darwin LNG Plant Location.

The LNG plant incorporates several other design features to reduce greenhouse gas emissions. These include the use of waste heat recovery on the PGT25+ turbine exhaust. The waste heat is used for a variety of heating requirements within the plant. The facility also includes the installation of ship vapor recovery equipment. The addition of waste heat and ship vapor recovery equipment not only reduces emissions that would have been produced from fired equipment and flares, but also result in a reduction in plant fuel requirements. This reduction in fuel gas results in a lowering of carbon dioxide released to the atmosphere. Peterson et al (2001) stress the importance of thermal efficiency and its impact on emissions.

NOx control on the gas turbines is by means of water injection which maintains the flexibility to accommodate fuel gas compositions needed for various plant operating conditions,

An aerial photo of the completed plant is shown in Figure 2. This figure shows the plant layout and the 188,000 m^3 LNG tank and the 1350 meter long jetty. Details regarding the development of this LNG project have been provided by Yates [2002, 2005a, 2005b].

Figure 2. Aerial View of the 3.7 MTPA LNG Plant.

IMPLEMENTATION OF AERODERIVATIVE ENGINE TECHNOLOGY

The facility includes with a total of six refrigeration compressors configured as shown in Figure 3 in a 2+2+2 configuration (2 x propane compressor drivers, + 2 x ethylene compressor drivers and 2 x methane compressor drivers). Both the propane and ethylene trains had speed reduction gearboxes, with the methane being a direct drive. The power turbine is a high speed power turbine running at 6100 rpm.

The layout of a typical train is shown in Figure 4. The gas turbine and compressors are deck mounted with down nozzles on the compressor. The inlet system (filter and evaporative cooler system) is located axial to the gas turbine inlet. ventilation air for the enclosure is taken as shown in the figure and routed from the top of the package into the enclosure.

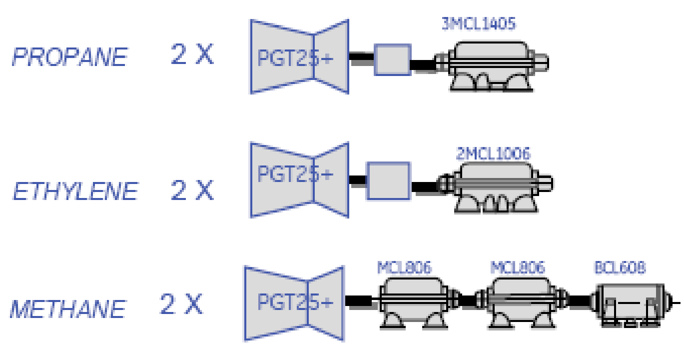

Figure 3. Overview of Turbomachinery Configurations.

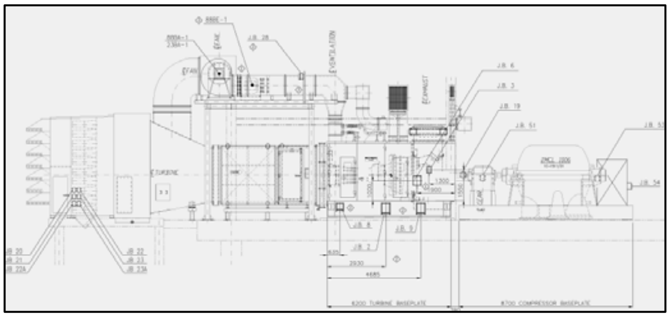

Figure 4. Layout of a Typical Gas Turbine Compressor.

A view of the gas turbines inlet systems is shown in Figure 5. As can be seen, four of the gas turbines have once through steam generators located on their stacks to capture heat and produce steam which is used for process needs. All six gas turbines utilize pulse type filters and media type evaporative coolers fitted with drift eliminators.

Figure 5. Gas Turbine Compressor Trains with Four Heat Recovery Units (OTSGs) on the left.

The PGT25+ engine used at the Darwin plant has a long heritage starting from the TF-39 aeroengine. This successful aeroengine resulted in the industrial LM2500 base engine which was then upgraded to the LM2500+. The PGT25+ is essentially the LM2500+ gas generator coupled to a 6100 RPM high speed power turbine (HSPT). The latest variant of this engine is the G4, rated at 34 MW. A comparison of thermal efficiency of aeroderivative engines compared to traditional industrial engines commonly used for LNG duty is shown in Figure 6.

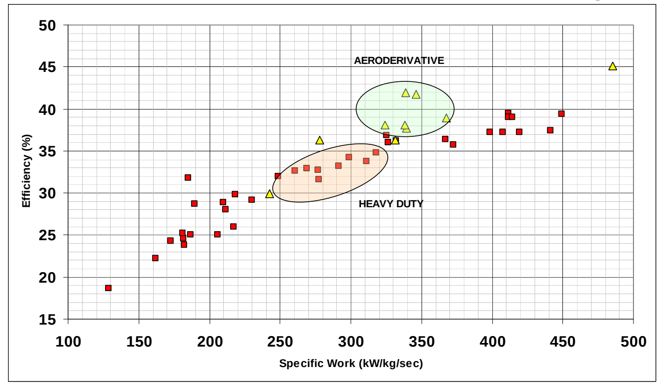

Figure 6. Comparison of Aeroderivative and Industrial Gas Turbines

The base engine was originally rated at 27.6 MW, and a nominal 37.5% ISO thermal efficiency. Since that time, its ratings have grown to its current level of 31.3 MW and a thermal efficiency of 41%.

The LM2500+ has a revised and upgraded compressor section with an added zero stage for increased air flow and pressure ratio by 23%, and revised materials and design in the high pressure and power turbines. Details may be found in Wadia et. al (2002). A view of the gas generator is shown in Figure 7.

Figure 7. LM2500+ Gas Generator Being Installed.

GAS TURBINE ENGINE DESCRIPTION

Axial Flow Compressor

The compressor is a 17 stage axial flow design with variable-geometry compressor inlet guide vanes that direct air at the optimum flow angle, and variable stator vanes to ensure ease of starting and smooth, efficient operation over the entire engine operating range. The axial flow compressor operates at a pressure ratio of 23:1 and has a transonic blisk as the zero stage[2]. The axial flow compressor is shown in Figure 8. As reported by Wadia et al (2002) the airflow rate is 84.5 kg/sec at a gas generator speed of 9586 RPM. The axial compressor has a polytropic efficiency of 91%. The transonic first stage blisk has 16 blades and an Aspect Ratio of 1.39 and has a stage pressure ratio of 1.39. Its tip speed is 415 m/sec (1363 ft/sec).

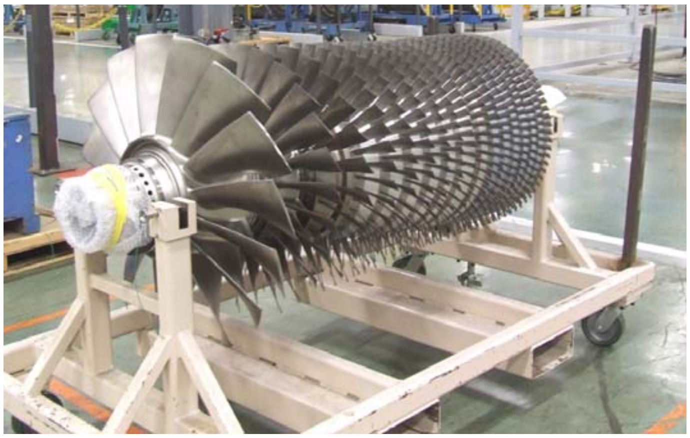

Figure 8. 17 stage Axial Flow Compressor of Gas Generator.

Annular Combustor

The engine is provided with a single annular combustor (SAC) with coated combustor dome and liner similar to those used in flight applications. The single annular combustor features a through-flow, venturi swirler to provide a uniform exit temperature profile and distribution. This combustor configuration features individually replaceable fuel nozzles, a full-machined-ring liner for long life, and an yttrium stabilized zirconium thermal barrier coating to improve hot corrosive resistance. The engine is equipped with water injection for NO_x control.

The water injection rate is a function of the fuel flow rate and is depicted in Figure 9.

[2] The zero stage operates at a stage pressure ratio of 1.43:1 and an inlet tip relative mach number of 1.19.

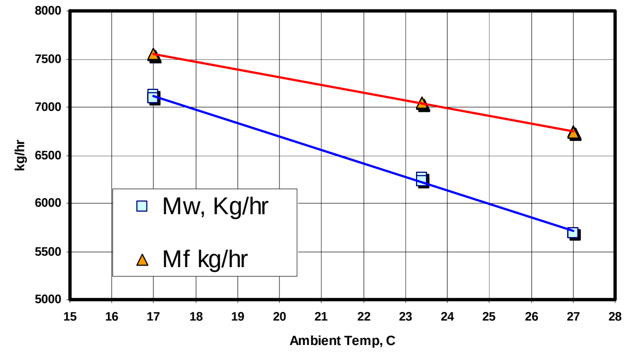

Figure 9. Water Injection Rate (Mw) and Fuel flow (Mf) for Different Compressor Inlet Temperatures.

High Pressure Turbine (HPT)

The HPT is a high efficiency air-cooled, two-stage design. The HPT section consists of the rotor and the first and second stage HPT nozzle assemblies. The HPT nozzles direct the hot gas from the combustor onto the turbine blades at the optimum angle and velocity. The high pressure turbine extracts energy from the gas stream to drive the axial flow compressor to which it is mechanically coupled. The HPT turbine is shown in Figure 10.

Figure 10. High Pressure Turbine Driving the Gas Generator Axial Flow Compressor.

High Speed Power Turbine

The gas generator is aerodynamically coupled to a high efficiency high speed power turbine. The high speed power turbine (HSPT) is a cantilever-supported two stage rotor design. The power turbine is attached to the gas generator by a transition duct that also serves to direct the exhaust gases from the gas generator into the stage one turbine nozzles. Output power is transmitted to the load by means of a coupling adapter on the aft end of the power turbine rotor shaft. The HSPT operates at a speed of 6100 RPM with an operating speed range of 3050 to 6400 rpm. The high speed two-stage power turbine can be operated over a cubic load curve for mechanical drive applications.

Engine-mounted accessory gearbox driven by a radial drive shaft

The engine has an engine-mounted accessory drive gearbox for starting the unit and supplying power for critical accessories. Power is extracted through a radial drive shaft at the forward end of the compressor. Drive pads are provided for accessories, including the lube and scavenge pump, the starter, and the variable- geometry control. An overview of the engine including the HSPT is shown in Figure 11.

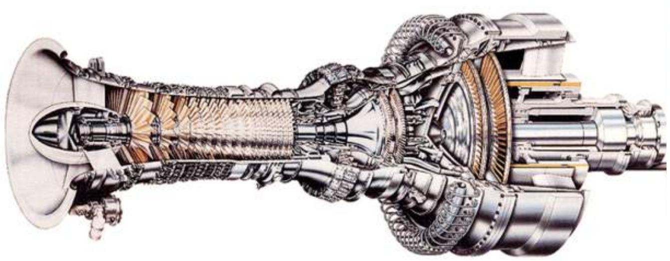

Figure 11. Layout of the PGT25+ Gas Turbine.

MAINTENANCE PLANS

A critical factor in any LNG operation is the life cycle cost that is impacted in part by the maintenance cycle and engine availability. Aeroderivative engines have several features that facilitate “on condition” maintenance. Numerous boroscope ports allow on-station, internal inspections to determine the condition of internal components, thereby increasing the interval between scheduled, periodic removal of engines. When the condition of the internal components of the affected module has deteriorated to such an extent that continued operation is not practical, the maintenance program calls for exchange of that module. This allows “on condition maintenance”, rather than strict time based maintenance.

The engine is designed to allow for on-site, rapid exchange of major modules within the gas turbine. On-site component removal and replacement can be accomplished in less than 100 man hours. The complete gas generator unit can be replaced and be back on-line within 48 hours. The hot-section repair interval for the aeroderivative is 25,000 hours on natural gas. However, water injection for NO_x control shortens this interval to 16,000 hours to 20,000 hours depending on the amount of water injected, which in turn iss set by the NO_x target level.

PERFORMANCE DETERIORATION AND RECOVERY

Gas turbine performance deterioration is of great importance to any LNG operation. Total performance loss is attributable to a combination of “recoverable” (by washing) and “non-recoverable” (recoverable only by component replacement or repair) losses. Recoverable performance loss is caused by fouling of airfoil surfaces by airborne contaminants. The magnitude of recoverable performance loss and the frequency of washing are determined by site environment and operational profile. Generally, compressor fouling is the predominant cause of this type of loss. Periodic washing of the gas turbine, by on-line wash and crank-soak wash procedures will recover 98% to 100% of these losses. The best approach to follow is to couple on line and off line washing. The objective of on line washing is to increase the time interval between crank washes. It should be noted that the cool down time for an aeroderivative is much less than that for a frame machine due to the lower casing mass. Crank washes can therefore be done with less downtime than heavy duty frame gas turbines. Gas turbine performance deterioration and compressor washing is covered in Meher-Homji et al (2001) and Meher-Homji et al (2004)

UPRATE OF THE PGT25+

A general advantage of using aeroderivative engines for LNG service is that they can be uprated to newer variants, generally within the same space constraints, and this might be useful feature for future debottlenecking as was done at the LNG plant and described later in the paper.

In 2010, the LNG facility instituted a debottlenecking project to upgrade all the gas turbines to the G4 variant. This uprated engine provides an increase in power of approximately 10%, by means of an increase in flow and a modest increase in control temperature.

The growth in power of this variant compared to the base engine is shown in Figure 12.

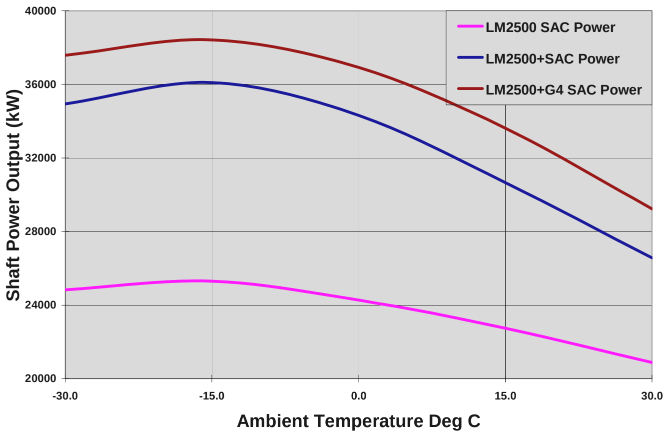

Figure 12. Growth of the LM2500+G4 Variant.

POWER AUGMENTATION BY EVAPORATIVE COOLING

LNG production is highly dependent on the power capability of the gas turbine drivers of the propane, ethylene and methane compressors. Industrial gas turbines lose approximately 0.7% of their power for every 1°C rise in ambient temperature. This effect is more pronounced in aeroderivative gas turbines where the sensitivity can increase to well over 1% per °C. The impact of ambient temperature on the engine power and air flow is depicted in Figure 13.

As aeroderivative machines are more sensitive to ambient temperature, they benefit significantly from inlet air cooling. The gas turbines utilize media type evaporative coolers. Details on media based evaporative cooling may be found in Johnson (1988).

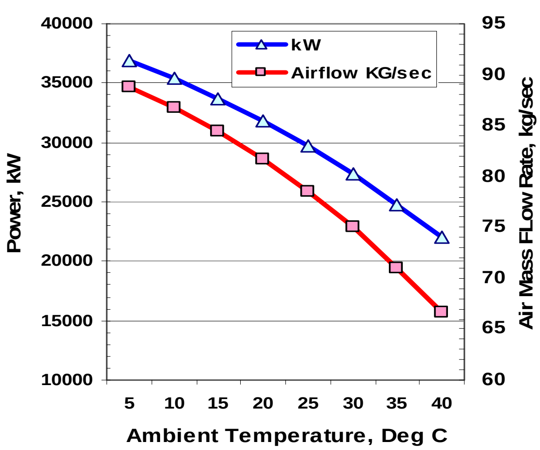

Figure 13. Variations in Power Output and Air Flow Rate for the Gas Turbine Driver.

Key advantages of power augmentation include:

- Boost of LNG production by lowering the gas turbine compressor inlet air temperature, increasing the air mass flow rate and power.
- Improvement of gas turbine thermal efficiency which results in lower CO_2 emissions/MW.

There is considerable evaporative cooling potential available in Darwin especially during the periods of high ambient temperatures as the relative humidity tends to drop as the temperature increases. The average daily temperature profile at Darwin is shown in Figure 14. The relationship of relative humidity and dry bulb temperature for a month is shown in Figure 15[3]. Details regarding the climatic analysis of evaporative cooling potential may be found in Chaker et al (2006)]. At the Darwin plant, evaporative cooling was considered to be the most cost effective approach reduce power fluctuations due to weather. The effectiveness of the evaporative cooling approach is much better, as expected, during the dry season, and is less effective during the wet season when the humidity is higher.

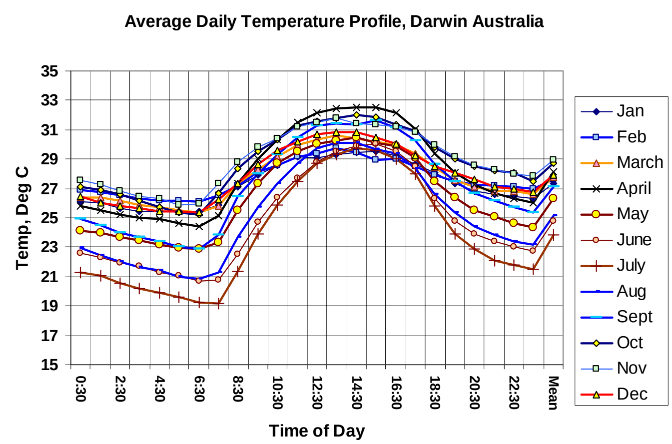

Figure 14. Temperature Profile Over Time of Day for 12 Months in Darwin.

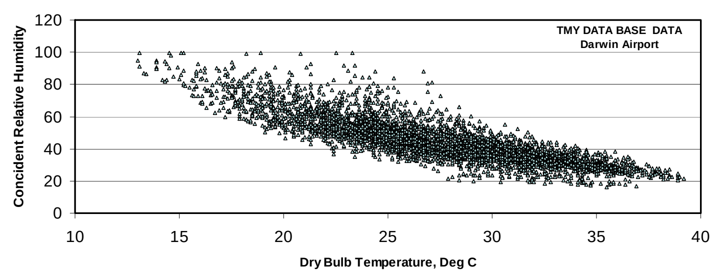

Figure 15. Relative Humidity vs. Dry Bulb Temperature (one month data).

[3] Data is from the TMY2 database, for Darwin Airport

The relationship between dry bulb temperature and wet bulb temperature over the year is shown in Figure 16.

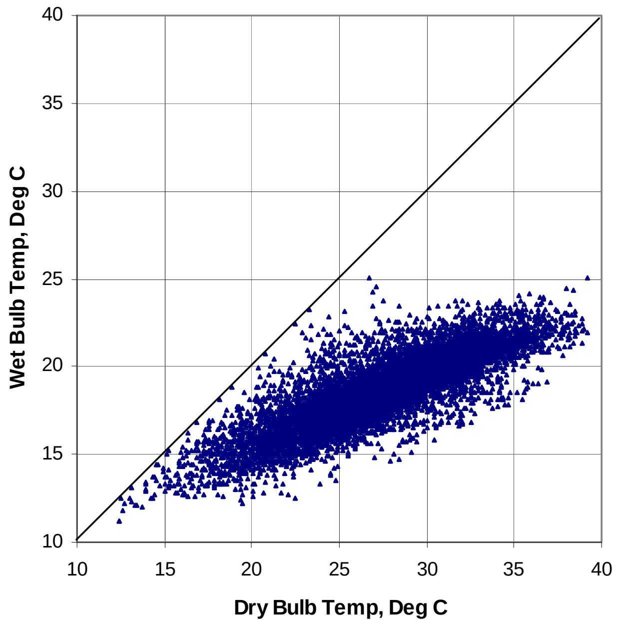

Figure 16. Dry Bulb and Wet Bulb Temperature Over the Year.

Media based evaporative coolers use a corrugated media over which water is passed. The media material is placed in the gas turbine air flow path within the air filter house and is wetted via water distribution headers. The construction of the media allows water to penetrate through it and any non-evaporated water returns to a catch basin. The media provides sufficient airflow channels for efficient heat transfer and minimal pressure drop. As the gas turbine airflow passes over the media, the air stream absorbs moisture (evaporated water) and heat content in the air stream is given up to the wetted media resulting in a lower compressor inlet temperature. A typical evaporative cooler effectiveness range is 85% to 90%, and is defined as follows:

$$\text{Effectiveness} = (T_{1DB} - T_{2DB}) / (T_{1DB} - T_{2WB})$$

Where,

T_{1DB} = Entering Air Dry Bulb Temperature
T_{2DB} = Leaving Air Dry Bulb Temperature
T_{2WB} = Leaving Air Wet Bulb Temperature

Effectiveness is the measure of how close the evaporative cooler is capable of lowering the inlet air dry bulb temperature to the coincident wet bulb temperature. Drift eliminators are utilized to protect the downstream inlet system components from water damage, caused by carry-over of large water droplets.

The presence of a media type evaporative cooler inherently creates a pressure drop which reduces turbine output. For most gas turbines, media thickness of 12 inches will result in a pressure drop of approximately 0.5 -1" water. Increases in inlet duct differential pressure will cause a reduction of compressor mass flow and engine operating pressure. The large inlet temperature drop derived from evaporative cooling, more than compensates for the small drop in performance due to the additional pressure drop.

Inlet temperature drops of around 10°C have been achieved which results in a power boost of around 8-10 %. A graph showing calculated compressor inlet temperatures (CITs) with the evaporative cooler for a typical summer month of January is shown in Figure 17.

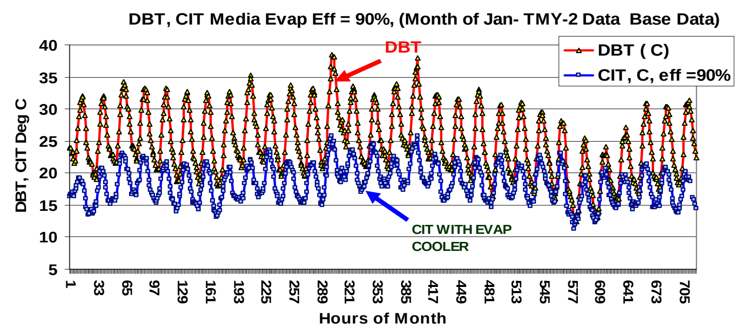

Figure 17. Calculated Compressor Inlet Temperature (CIT) Due to Evaporative Cooling over a Summer Month.

A graph showing compressor inlet temperature (after the evaporative cooler) is shown in Figure 18. The impact of evaporative cooling is evident from this bar graph in the suppression of extremely high temperatures.

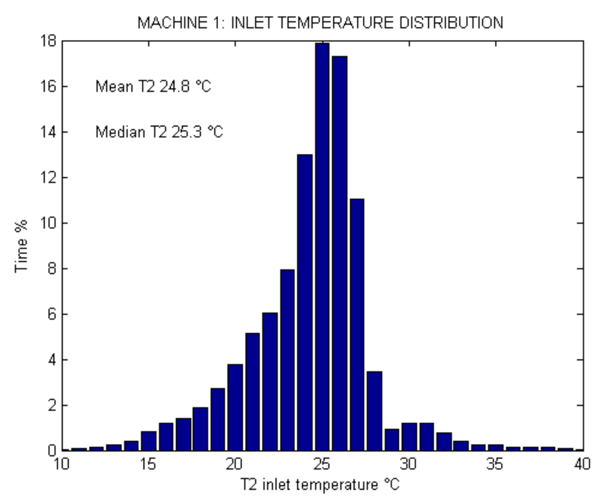

Figure 18. Bar Graph of Temperature – Time Distribution for a Gas Turbine.

REFRIGERATION COMPRESSORS

The design of LNG compressors involving large casing sizes, optimized impeller designs, high inlet relative Mach numbers, 3D flows, and the complexities of sidestream mixing, requires that a careful evaluation of the specific design and experience be made. The propane compressor is the most challenging machine in terms of flow coefficient and inlet relative Mach number. Because of the complexity of the compressor designs, process optimization has to be done in cooperation with the compressor designer to ensure that compressor selections are aerodynamically and mechanically robust while meeting process performance and operability requirements. This is an iterative process involving the compressor designer the process licensors and the EPC team.

Design fundamentals and terminology for centrifugal compressors can be found in Aungier [2000], Japikse [1996]. Details on LNG Compressor design may be found in Meher-Homji, et al [2007].

The design complexities, risks and compromises involved in the selection and design of large refrigeration compressors include aerodynamic and mechanical issues and constraints. The final compressor design involves several interrelated tradeoffs between aerodynamics, rotordynamics, impeller stress, efficiency and operating range. Understanding the complexities requires an appreciation of these interactions. Issues that are to be examined for each compressor selection include:

- Machine Mach number and inlet relative Mach number
- Selection of 2D and 3D impellers
- Impeller head per stage
- Range vs. Efficiency tradeoffs
- Head rise to surge and operating range
- Aerodynamic mismatching of stages
- Complexities of sidestream mixing
- Rotordynamic lateral behavior and stability.
- Casing stresses and designs.
- Need for model testing/ CFD analysis.

It is not advisable to set absolute limits on certain parameters as one might do for more traditional compressors and therefore a case by case study has to be made of each compressor service. A valuable discussion of the tradeoffs involved in compressor design is provided by Sorokes [2003].

From the perspective of compressor selection, design, and testing, close designer- user interaction and good communication is important to derive a robust compressor solution that will operate under varied operating conditions. Imposition of simple and rigid rules of thumb and specifications by the user that do not recognize that design compromises are inherent in compressor design will often result in non optimal designs.

COMPRESSOR SELECTIONS AND MODEL TEST

The machinery configurations for the LNG Plant were as follows:

- Propane: 2 X PGT25+ Gas Turbine + Speed reduction GB + 3MCL1405 Compressor. The compressors are horizontally split and utilize tandem dry gas seals

- Ethylene: 2 X PGT25+ Gas Turbine + Speed reduction GB + 2MCL1006 Compressor (Back to back design) The compressors are horizontally split and utilize tandem dry gas seals.

- Methane: 2 X PGT25 + Gas Turbine + MCL806 + MCL 806 + BCL608 (i.e., three casing compressor). The first two cases are horizontally split with the third high pressure case being a barrel design. All compressor casings utilize tandem dry gas seals.

The most challenging situation occurred with the first stage of the propane compressor where the unit operated at high inlet relative Mach number. Because of this, a model test was instituted as shown in Figures 19. The station designations were as shown in Figure 20. The purpose of this test was to evaluate the thermodynamic performance and operating range of the first wheel of the 3MCL1405 compressor which was equipped with a high Mach number 3D wheel. The stage used using a 34.5% scale model. The diameter of the model wheel was 400 mm and the setup included the inlet and return channel. Test instrumentation included the use of pitot Kiel type probes for total pressure, and thermocouples for total temperature. Flush mounted probes were used for static pressure and special probes were uses to measure flow angle and total pressure. The tests was run by controlling

- Mach Number
- Flow Coefficient.

The flow coefficient was explored from choke to surge at three speeds.

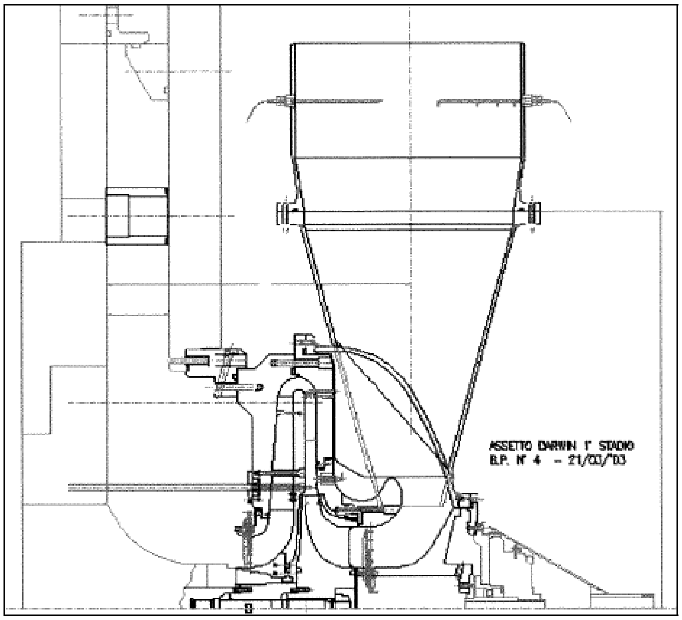

Figure 19. Model Test Setup of Propane First Wheel.

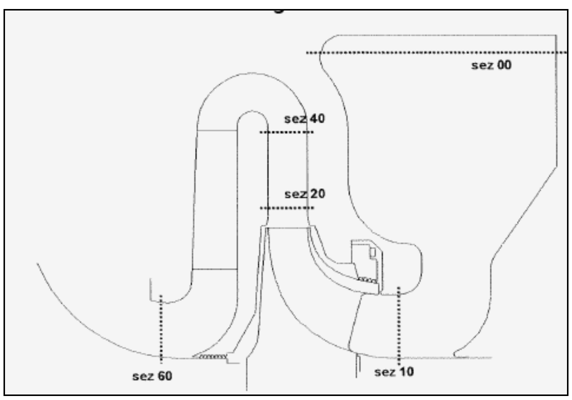

Figure 20. Station Designation for Model Test.

Both the propane and ethylene trains have speed reduction gearboxes. All compressors are horizontally split except for the last casing of the methane string which is a barrel design. The gas turbines and compressors are mezzanine mounted as shown in Figure 21, which facilitates a down nozzle configuration for the compressors. This facilitates the maintenance of the components as piping may be left in place during compressor dismantling

Figure 21. Photograph of Compressor Deck showing the Six Compressor Strings. From front to back- 2 x Methane compressors, 2 x Ethylene compressors and 2 x Propane Compressors.

DYNAMIC SIMULATION AND OPERATOR TRAINING SYSTEM THROUGH PROJECT LIFE CYCLE

As reported by Valappil et al (2004a and 2004b)] dynamic simulation has established itself as a valuable technology in the LNG industry. It is useful for a variety of purposes, including engineering and process studies, control system studies and applications in day-to-day operations and also for the development of dynamic Operator Training Systems (OTS). Process modeling, either steady state or dynamic can be carried out in the various stages of the LNG process lifecycle. The benefits of integrating these modeling activities have been realized in recent years. The dynamic model, evolving with the various stages of a plant lifecycle can be tailored for various applications within the project lifecycle as shown in Figure 22. The operability and profitability of the plant during its life depends on good process and control system design. Dynamic simulation helps to ensure that these aspects are considered early in the plant design stage. This eliminates or reduces any costly rework that may be needed later. The operator training system was implemented at the LNG plant and has proved to be a very valuable training tool allowing operators to examine and train for dynamic plant operation.

There are several benefits to be realized by using the dynamic simulation in the various stages of an LNG project. On the process side, dynamic simulation is an important tool for evaluating the anti-surge control system for the refrigeration compressors. The reliable protection of this equipment is critical for long-term smooth operation of the LNG plant. Also, dynamic simulation can be pivotal in the support of sizing of specific key relief valves, and the overall relief system and optimum selection of equipment sizes. LNG plants are also characterized by extensive heat integration, the operational implications (stability and startup) of which can be studied by simulation. Further, the effect of external factors like ambient

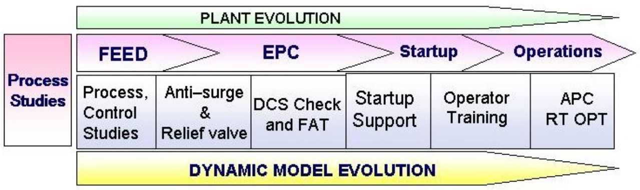

Figure 22. The Use of Dynamic Simulation Models.

conditions and compositional changes on the future plant operation can be analyzed to further optimize the design. Dynamic simulations are also fundamental to understand compressor behavior during operation and during transient conditions such as trips, and startup. Figure 23 shows the trip scenario on a centrifugal compressor.

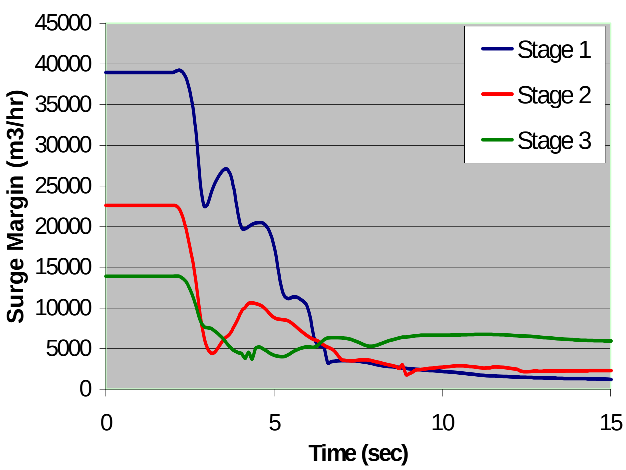

Figure 23. Response of a Refrigeration Compressor in a Trip. Surge Margins for the Three Stages are Shown as the Machine Speed Drops.

COMPRESSOR TESTING

All of the compressor casings and spare rotors received API 617 mechanical run tests. Gearboxes were tested per API 613, and each kind of compressor was given a Class 2 ASME PTC 10 Test. All the testing was concluded successfully.

ETHYLENE COMPRESSOR CASE STUDY

A case study relating to the ethylene compressor is presented here.

The ethylene compressor rotor, is shown in Figure 24. The compressors have two sections in a back-to-back configuration, with six impellers, and 220 mm dry gas seals. The rotors each weigh 5800 Kg, and are mounted in 200-mm tilting pad bearings with a length to diameter ratio (L/D) of 0.7. The maximum rotor diameter under the impellers is 420 mm, and the bearing span is 3.521 meters. The range from minimum to maximum continuous speed (MCS) is 4118 to 5087 rpm, with a trip speed of 5314 rpm. Shop (mechanical running) tests were performed on all three rotors

The compressor had six impellers (as shown in Figure 24) with the following key features:

- Two sections, in back to back configuration with three wheels each.
- Journal bearings were 200 mm, with L/D = 0.7
- Shaft diameter at impeller bore = 420 mm
- Bearing span =3521 mm
- Compressor speed range = 4118-5087 with trip speed being 5314 rpm.
- First critical speed (per rotor dynamic report) = 2,240 rpm
- Second critical speed (per rotor dynamic report) = 6,600-6,800 RPM.
- Bearing Type: Tilting pad, Load between Pad, 5 pads

Figure 24. Ethylene Compressor Rotor with six Impellers in Back-to-Back Configuration.

During the mechanical run test the ethylene rotor ran exceedingly smoothly, with vibration levels in the 5-11 micron pk-pk range. The acceptance level is 25.4 microns pk-pk.

However as the rotor reached the maximum continuous speed of 5087 rpm, a phase change of approximately 180 degrees was noted at the non-drive end and the vibration amplitudes on the non-drive end bearing, increased. The combination of the large phase change, with the rapid vector change in vibration near MCS, raised concerns that the second critical speed, which had been predicted to satisfy API 617 margins, might in fact be much closer than predicted.

The full shop test runs involved acceleration to maximum continuous speed, then a further acceleration to trip speed, followed by four hours at maximum continuous speed, with some variation of inlet oil temperature. One such run up and run down in shown in Figure 25.

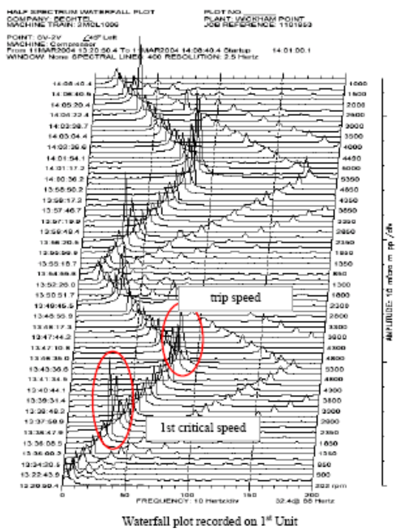

Figure 25. Waterfall plot of Machine Run Up and Run Down.

A series of run-up and run-down data on the same bode plot, obtained during the mechanical running test for the first rotor is shown in Figure 26. The first critical speed is around 2250 rpm and some probes exhibit a distinct split (double peak) in this first critical speed, particularly the non-drive end horizontal probe (shown in the bottom frame of Figure 26). In addition to the first critical speed characteristics, a large phase shift approaching 180 degrees occurs near maximum continuous speed (MCS) at both non-drive end bearings. The direction of this phase change reverses for an immediately successive pair of accelerations and decelerations to trip speed (run-up, run-down). The rapid vector change in vibration near MCS can be observed, together with relatively high vibration, exceeding 20 microns at trip speed.

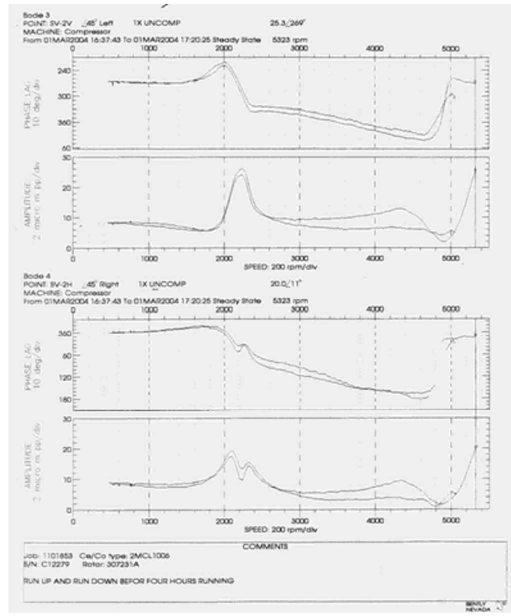

Figure 26. Ethylene Rotor Bode plot Showing Growth in Amplitude and Phase Shift when Operating at MCS.

Extensive testing and rotor dynamic modeling was done with the help of South West Research Institute working in conjunction with the OEM. Details are provide in Vannini et al (2006). The tests included:

- Detailed rotordynamic modeling including modeling of rotor bending and shear flexibility, shaft distributed mass and rotary/polar inertia discretized at each station, with mass, polar, and transverse inertias of mounted components such as impellers, sleeves, thrust disk, couplings, and nuts lumped at the station corresponding to the component’s center of gravity. The model accounts for the rotor stiffening caused by all interference fits, using the method of Smalley, et al. (2002).
- Free-Free modal testing of the rotor for calibration purposes. To help validate the model, free-free response to shaker excitation was obtained for one of the rotors. In this testing, the rotor was supported in a vertical orientation from a hook attached to the drive end of the rotor, as shown previously in Figure 27. Accelerometers are arrayed at 10 points along the rotor, and shaker excitation is applied near the bottom. The freely mounted rotor has very little internal damping, so the resonant response at natural frequencies of the rotor is very distinct and sharp.
- Experimental determination of the support stiffness. To optimize accuracy of a model for predicting rotor-bearing system dynamics, flexibility of the structure, which supports the bearings, can become important. The casing for these compressors is horizontally split, and the bearing support structure is outboard from the casing. This

structural configuration can contribute to support flexibility, particularly in the vertical direction.

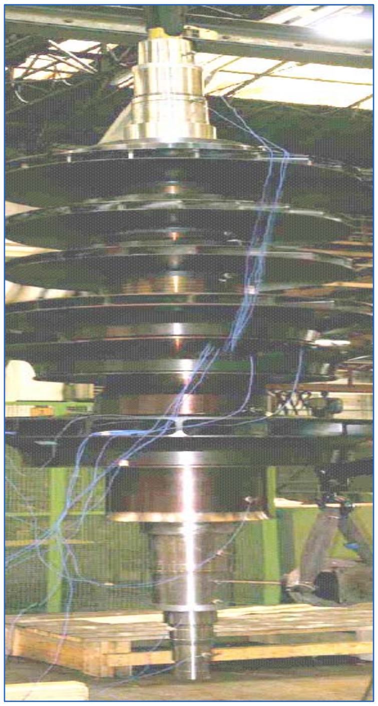

Figure 27. Free- Free Rotor Modal Testing

Figure 28 shows a photograph of the casing under test with loads applied by a dynamic shaker to help identify likely casing flexibility. This photograph illustrates the outboard bearing support. Figure 28 also shows schematically the different orientations of the shaker during these tests. Accelerometers were mounted at various points on the casing and, in combination with a load cell between the shaker and the casing, provided a basis for calculating impedances.

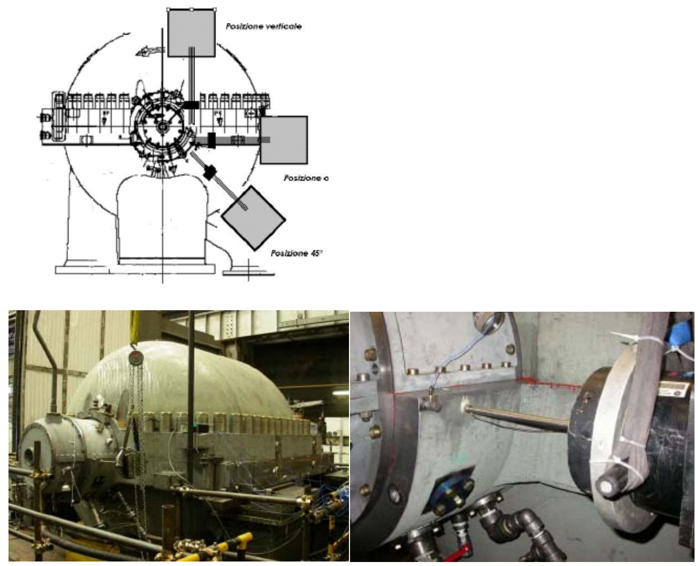

Figure 28. Support Stiffness Checks.

The analyses described above set to rest concerns about the second critical speed and the associated threat to integrity of two compressors for a critical application. Subsequent operation in the field has proved that the unit operates trouble free.

PLANT STARTUP AND OPERATING EXPERIENCE

In examining the performance of the LNG plant over approximately five years of operation, all expectations have been met.

During startup, some vibrations were noted on the compressor that were discovered to be the gas generator speed transferred to the compressor. A plot of this is shown in Figure 29.

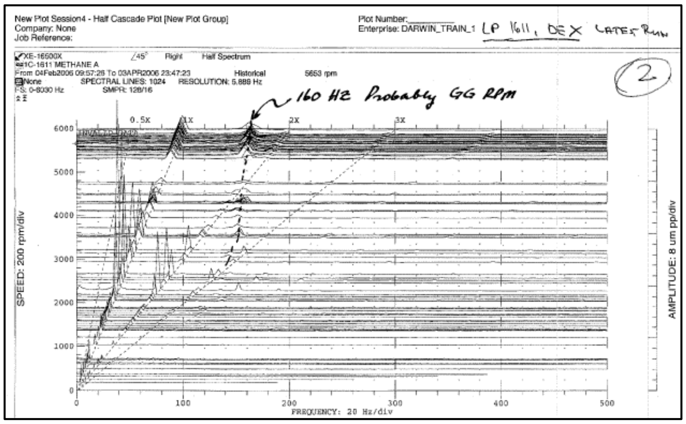

Figure 29. Gas Generator Vibration at 160 Hz Noted on Process Compressor End.

During startup, some adjustments had to be made to the T3 (compressor discharge temperature) control limits of the gas turbine after which performance was as expected. This change had no affect on engine life.

Issues relating to integrations and some lessons learned are provided below.

In terms of reliability and availability, the planned targets have been met and exceeded. One engine had to be removed during an inspection due to a combustor crack that was noticed, however this was not an underlying problem and a recent inspection at 16000 hours have indicated that all machines are operating within tolerances.

Key operational observations include:

- Hot section turnaround is approximately 2.5-3 days
- Frequency of crank washing is currently 14-16 weeks
- Long term power decay has been lower than the original assumed number
- Normal restart after a trip can be done in approximately 0.5 hours assuming no 4 hour lock out occurs. The turbine can be back on line within 20-30 minutes if there is a spurious trip.
- Shutdown duration for crank wash- 6 hours (from full production to full production).

Due to the “two trains in one” capability, variable speed operation and ability to rapidly start at settle out pressure, operational flexibility has been very high. The plant capability under different conditions is shown in Table 1.

Table 1. Plant Operation due to Two- Trains- in- One Philosophy

OPERATIONAL FLEXIBILITY	Nominal Operating Range (% of design)
Full Plant (6 turbines operating)	80 – 105%
One Turbine Off-line	60 – 80%
Three Turbines Offline* *At least one turbine on each cycle must be operating	30 – 60%
Plant Idle	0 – 30%

Some issues were identified relating to integration of aeroderivative engine operation with the plant DCS system.

There are three types of trips:

- **TYPE [a] Normal Shutdown**- in this the GG comes to core idle (approx 6800 rpm) where it is held for 5 minutes. The LPT is at approximately 1600 rpm at this stage. After this period, the unit is tripped and the GG and PT speeds come to zero. The Turning gear (TG) is then energized at this point.
- **TYPE [b] Full Load Emergency Trip**.- this is a trip based on a certain set of engine parameters that are deemed critical. In this trip the fuel is cut off and the turbine comes down and a 4 hour lock out is imposed (on a timer) unless the trip can be reset within 10 minutes and the starter motor engaged to initiate a 5 minute cooldown
- **TYPE [c] Motor Trip (Crank Trip)** – In this trip, the GG and power turbine speeds drop and the hydraulic starter motor is engaged at approximately 200 rpm GG speed which accelerates the GG to around 2000 rpm. After a 5 minute cool down, the starter motor deenergizes and after the GG comes to a standstill, the turning gear is energized. This must be done within a 10 minute timeframe else a four hour lock out results. The turning gear (TG) is not energized until the GG speed equals zero. The reason for this is that there is not enough gas energy to break away the PT, but if the TG is energized, then the load compressor speed will attain 80-200 rpm which may be damaging to the dry gas seals. Consequently, in this mode, there may be as long as 25 minutes between the trip and the time when the TG is energized, which could allow a bow to occur in the load compressor rotor.

To resolve or mitigate this an attempt should be made to understand and categorize trip parameters to minimize Type B trips and to try and move trip parameters from type C to A, as the logic of type B trips defeats the use of turning gears.

Water Injection Experience

In general the water injection has been successful and currently time between overhauls has been extended to close to

25000 hours. One combustor exhibited premature distress (Figure 30) but this was considered an outlier and subsequent operation and inspections have shown that the expected life has been met.

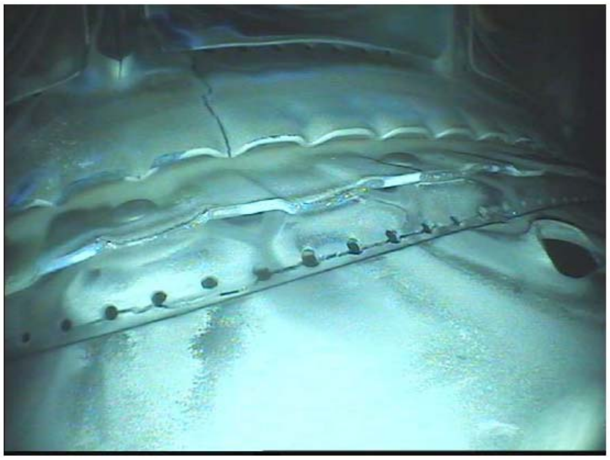

Figure 30. Combustor Distress.

A gear tooth failure was noted on one gearbox and was found during an inspection. The damage is shown in Figure 31. A small step change in vibration was noted during the event and is shown in Figure 32.

Figure 31. Gear Tooth Breakage.

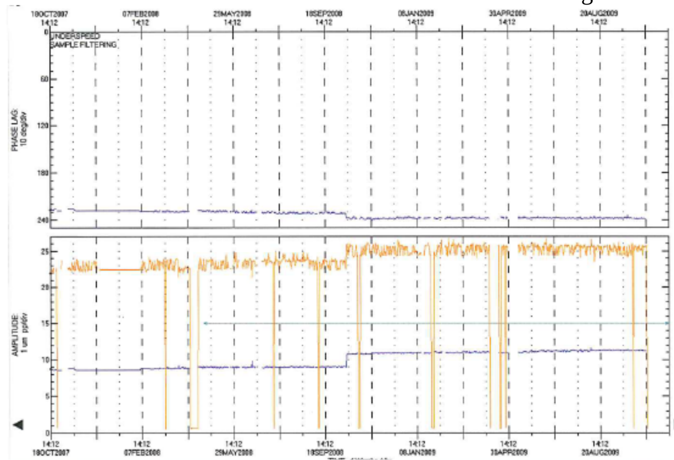

Figure 32. Small Step Change in Vibration and Phase Change – Gear Tooth Breakage.

PLANT DEBOTTLENECKING BY AERODERIVATIVE ENGINE UPGRADE

An advantage of using aeroderivative engines for LNG service is that they can be uprated to newer variants, generally within the same space constraints, and this is a useful feature for future debottlenecking.

The plant has implemented an upgrade to the PGT25+G4 engines in 2010. The LM2500+G4 (PGT25+G4) is the newest member of the LM2500 family of aeroderivative engines. The engine retains the basic design of the L engine but increases the power capability by approximately 10 percent without sacrificing hot section life. The modification increases the power capability of the engine by increasing the airflow, improving the materials and increasing the internal cooling. The number of compressor and turbine stages, the majority of the airfoils and the combustor designs remain unchanged from the LM2500. Details of this variant are described by Badeer (2005).

INITIAL DEBOTTLENECKING STUDIES

Several studies were conducted relating to debottlenecking activities. These included the evaluation of various options for power boost including inlet air chilling, and upgrade to the G4 engines. Several of these studies were done by using dynamic simulation software and the operator training system framework described in an earlier section. This evaluation allowed a quantitative feel for the impact of the upgrades on the LNG plant. The outcome of these studies indicated that the upgrade to the G4 option (water injected) would be the most effective in terms of cost and schedule. The power boost expected with the uprated engine compared to the original engine is shown in Figure 33.

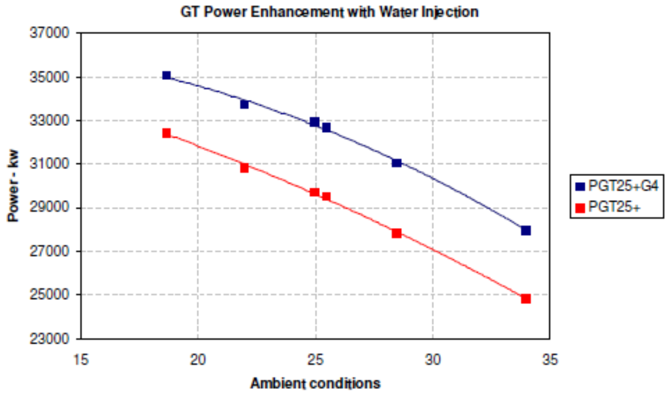

Figure 33. Power Uprate with the New G4 Engine Variant.

DEBOTTLENECKING PLANNING GOALS AND PLANNING TEAM

Once the decision was made as to the approach to debottleneck the plant, key implementation goals were defined. These included:

- Minimizing downtime due to the very large downtime costs of an LNG facility
- Certainty of outcome with respect to the schedule of downtime
- Minimizing the impact on existing layout and auxiliary systems.

A dedicated team was formed to examine and study all aspects of the debottlenecking activities. Extensive meetings were held with the turbine OEM that would be implementing the debottlenecking to cover all aspects of the debottlenecking activities. This included special testing procedures for the high speed power turbine, testing for the gas generators, and discussions relating to the software upgrades that would be required for the new engines.

A detailed engineering analysis and verification of complete auxiliary systems under the debottlenecked running conditions was conducted by the gas turbine OEM. The functionality of all the existing auxiliary systems (gear, control panel fuel skid.) were verified under the most extreme conditions to validate that they could be used.

ENGINEERING CHECKS FOR DEBOTTLENECKING

The following detailed engineering checks were carried out:

- Driven train verification, including coupling/load coupling, gear and centrifugal compressor shaft.
- Fuel gas and metering skid design checks.
- Water injection and metering valve checks
- GT enclosure ventilation system design check
- GT performance using Darwin fuel gas compositions and ambient conditions with/without evaporator cooler
- Inlet and exhaust ducts evaluation under new flow conditions
- Design checks of lube oil run down tank volume for new operating conditions.
- Study of synthetic and mineral lube oil system including cooler capacities for new conditions.
- Noise verification studies

Special attention was placed on the verification of performance and a performance map was issued based on cycle deck runs and experience of other G4 engine installations. Specific documents were also developed to summarize the data related to the exhaust gas temperature and speed profile as the once through steam generators (OTSGs) were going to be replaced during the same outage.

DEBOTTLENECKING ACTIVITY TIMELINE

The purchase order to the turbine OEM was issued in November 2008 and all six G4 gas turbine units were delivered by December 2009. The first two units were delivered in August 2009. This accelerated delivery allowed the early implementation of a changeout on the two propane gas turbine drivers in January-Feb 2010. The January installation was very successful and took only nine days. This included the change out of the damaged ethylene compressor speed reducing gear set. The remaining four gas turbines were changed out in April –May of 2010. For this second phase, the scope of work was extended to incorporate a change out of all process centrifugal compressors tertiary seals, as well as implementing the required propane gear box upgrade to accommodate the higher power of the upgraded turbine. The second phase installation in April – May of 2010 was particularly challenging as the work density on the site was very high, with multiple activities occurring including the changeout of the OTSGs. Careful planning and good communications allowed the work to proceed smoothly and without any HSE incidents. Mechanical completion was attained in 26 days and plant re-start was achieved on target.

All six gas generators were tested at the OEM"s facility in using a nozzle test (Figure 34). All power turbines underwent a special no load mechanical run test at the packager's facilities.

Darwin LNG accepted an invitation by the Alliance Pipeline Company, the first user that executed a G4 gas turbine upgrade with the HSPT module, to learn about their experiences when executing their upgrade project. This interaction not only allowed valuable technical insight based on their operating experience, but Alliance also shared all the lessons learned from their upgrades (including manpower and tooling requirements). Then they invited the plant turbine specialist to participate and work hand- in- hand with their staff at a turnaround with a PT upgrade, at one of their facilities.

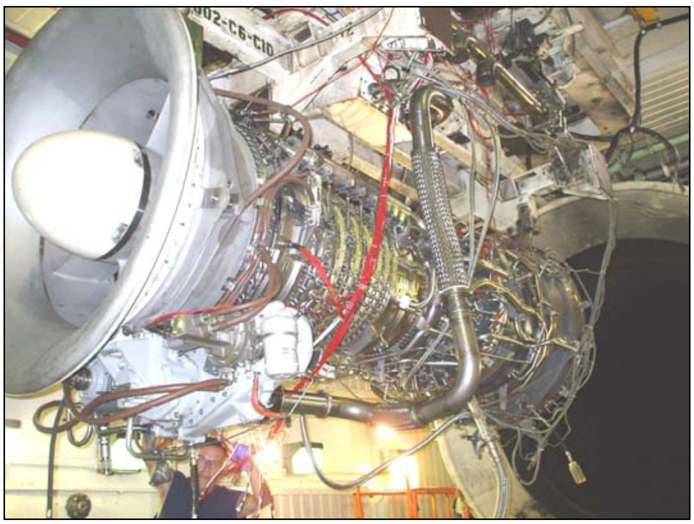

Figure 34. G4 Gas Generator Undergoing Nozzle Test.

PRE SHUTDOWN ACTIVITIES

A shut-down team consisting of engineering, maintenance and operation resources was assembled and mobilized well ahead of the shutdown. Representatives from the business unit, construction contractors and major OEM's were part of this team.

Pre-shut down planning included tasks such as:

- Developing necessary documentation relating to the installation by the project team. These documents included field modification forms, maintenance recommendations, job cards integrated with field quality control plans, HSE procedures etc.
- Integration of the turbine upgrade project with the other shut-down projects at the site, such as the OTSG change out.
- Special attention was paid to logistics tasks, optimization of space, in light of the multiple activities underway.
- Dedicated review of all special tools and some redesign was implemented. Some new tools were developed to save time during installation and for ergonomics.
- Communications during the full process were carefully managed with bi-weekly calls that were held for the full duration of the project from order to implementation. This covered engineering, manufacturing, and testing.
- During the installation phase at site, daily meetings were held to review progress, analyze potential problems and preemptively develop solutions.
- After the successful installation, "lessons learned" workshop was held to discuss in a transparent manner, all the lessons that could be learned to allow continuous improvement. This meeting was held immediately after the installation so that all issues were fresh in the mind of the team members.

AUXILIARY SYSTEM MODIFICATIONS/ CONTROLS MODIFICATIONS DURING SHUTDOWN

The following work in the auxiliary sytems was executed during the shutdown, as part of the upgrade project and to enhance the reliability to the compressor trains.

- Fuel metering valves were modified by increasing the plug size due to new fuel rate
- Modification of the control of the metering valve
- Fuel piping modifications to make it more robust and to facilitate the maintenance of this line
- Replacement of the warm up and inter stage valves installing new valves with limit switches
- Modification of the Seal Gas Panel, replaced the PSV and modified panel arrangement and control settings, this modification was needed due to the tertiary seal upgrade for the centrifugal compressor dry gas seals.
- Required software modifications to the gas turbine control system.
- Modifications to the software to accommodate fuel gas chromatographs.

IMPLEMENTATION OF THE DEBOTTLENECKING PROGRAM

The first phase of the debottlenecking program occurred in January 2010 and involved the upgrade of two engines to the new G4 variants, a task that was completed in nine days. The second phase which was more challenging due to the multiple activities that concurrently occurred on the compressor deck and surrounding areas was successfully completed in April-May 2010.

The implementation of the successful debottleneck project is pictorially indicated in Figures 35 through 43.

Figure 35. Change Out of OTSGs.

Figure 36. Propane Compressor Tertiary Seal Change out Activities.

Figure 37. Power Turbine Module

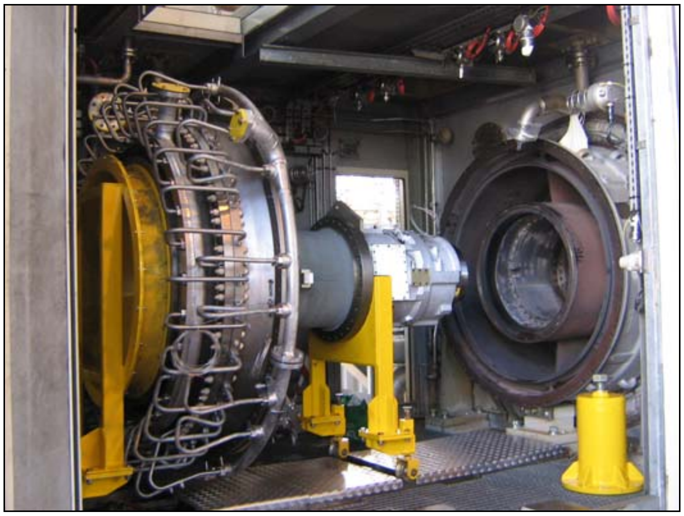

Figure 38 .   Power Turbine Change Out

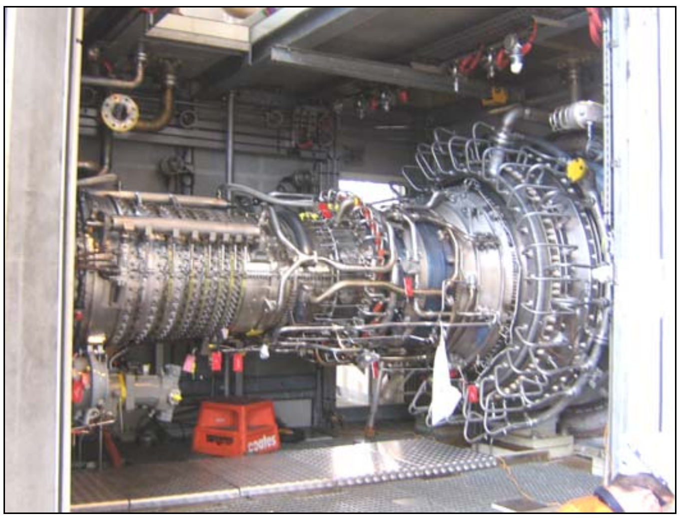

Figure 39. Gas Generator and Power Turbine.

Figure 40. Gas Generator Arriving at Compressor Deck Site.

Figure 41. Gas Generator Being Moved into Place for Installation into Enclosure.

Figure 42. High Speed Power Turbine Being Moved into place for Installation.

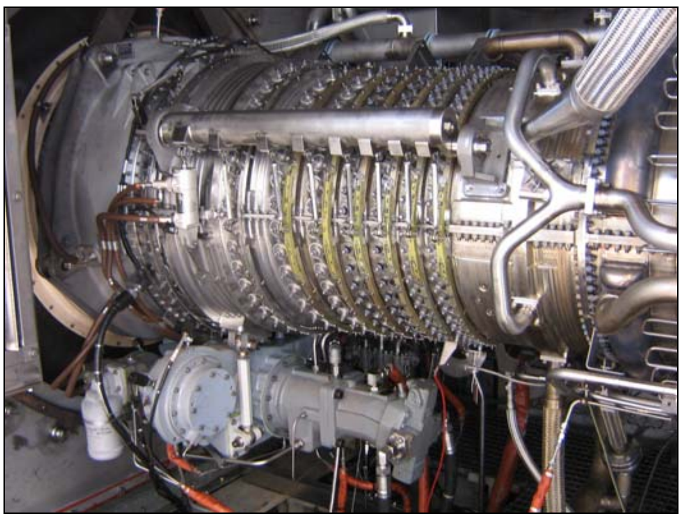

Figure 43. Gas Generator Showing Torque Tube for Variable Geometry Control.

CONCLUSIONS

This paper has provided an overview of the design, testing, commissioning, operating experience and debottlenecking of the world's first aeroderivative driven LNG liquefaction facility. The facility has been successfully operated for over 4 years and has met and exceeded its production goals. Several case studies and turbomachinery related considerations have been covered. The LNG industry now has several other LNG aeroderivative plants under construction and in the FEED stage.

The paper has described a highly successful debottlenecking effort that was implemented in two phases in 2010. In the first phase, two of the propane drivers were upgraded to the G4 variant thus producing approximately 10 percent higher power. The remaining four gas turbines were upgraded in June 2010, along with other ancillary equipment. Close planning and detailed execution studies enabled a very successful debottlenecking program to be executed within the planned timeframe.

NOMENCLATURE

DLE	Dry Low Emission
DBT	Dry Bulb Temperature
FEED	Front End Engineering Design
WBT	Wet Bulb Temperature
GG	Gas Generator
GT	Gas Turbine
HPT	High Pressure Turbine (GG Turbine)
HSPT	High Speed Power Turbine
MCS	Maximum Continuous Speed
OTS	Operator Training System
OTSG	Once Through Steam Generator
RH	Relative Humidity
SAC	Standard Annular Combustor
TG	Turning Gear

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