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AERODERIVATIVE GAS TURBINES FOR LNG LIQUEFACTION PLANTS – PART 2: WORLD'S FIRST APPLICATION AND OPERATING EXPERIENCE

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ABSTRACT

LNG market pressures for thermally efficient and environmentally friendly LNG plants coupled with the need for high plant availability have resulted in the world's first application of high performance aeroderivative gas turbines for a 3.7 MTPA LNG plant in Darwin. The six engines utilized are GE PGT25+ engines rated at 32 MW ISO driving propane, ethylene and methane compressors. The paper describes the design, manufacture, testing, and implementation of these units focusing on both the gas turbine and the centrifugal compressors. Power augmentation utilized on these units is also discussed. An overview of operating experience and lessons learned are provided. Part 1 of this paper provides a detailed analysis of why high thermal efficiency is important for LNG plants from an economic and greenhouse gas perspective.

1.0 INTRODUCTION

Market pressures for new 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 PGT25+ aeroderivative gas turbines for the 3.7 MTPA Darwin LNG plant. The plant was operational several months ahead of contract schedule and has exceeded its production targets. This paper will describe the philosophy leading to the world's first aeroderivative based gas turbine plant and future potential for the application of larger aeroderivative drivers which are an excellent fit for the ConocoPhillips *Optimized Cascade*SM LNG Process.

Aeroderivative engines fit the Optimized Cascade process because of the "two trains in one" design concept[1] that facilitates the use of available aeroderivative engines. The plant is able to operate at reduced rates of 50-70% in the event that one refrigeration compressor is down. The wide range of large aeroderivative engines allow flexibility in plant capacities using the Optimized Cascade process. Benefits of aeroderivative engines over large heavy duty single and two shaft engines include significantly higher thermal efficiency and lower greenhouse gas emissions, the ability to start without the use of large helper motors, and improved production efficiency[2] due to modular engine change outs. This paper covers several practical aspects relating to the application of aeroderivative gas turbines as refrigeration drivers and discusses design and implementation considerations.

2.0 OVERVIEW OF THE DARWIN LNG PROJECT

The Darwin LNG plant was successfully commissioned and the first LNG cargo was supplied to the buyers, Tokyo Electric and Tokyo Gas, on February 14, 2006. The Darwin plant represents an innovative benchmark in the LNG industry as the first to use aeroderivative gas turbine drivers. This follows another landmark innovation by ConocoPhillips - being the first to apply gas turbine drivers at the Kenai Alaska LNG plant built in 1969.

The Darwin plant is a nominal 3.7 million tonne per annum (MTPA) capacity LNG plant at Wickham Point, 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). Delineation drilling over the next two years determined the Bayu-Undan Field to be of world-class quality with 3.4 TCF gas and 400 MMbbls of recoverable condensate and LPG. In February of 2004, the Bayu-Undan offshore facility commenced operation with current production averaging 70,000 bbls of condensate and 40,000 bbls of LPG per day.

[1] Each refrigeration service is accomplished by at least two parallel trains.

[2] The production efficiency is defined as actual annual LNG production divided by the required annual LNG production.

The shareholders of the Darwin LNG project are ConocoPhillips (plant operator), with 56.72%, ENI with 12.04%, Santos with 10.64%, INPEX with 10.52%, and Tokyo Electric and Tokyo Gas with a combined 10.08%.

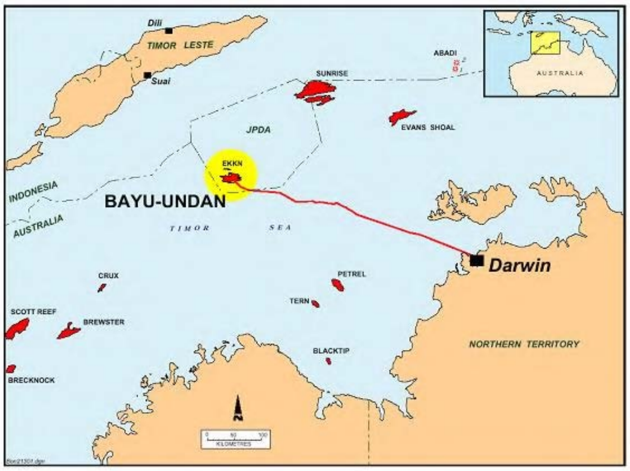

Figure 1. Bayu-Undan field location and the Darwin LNG plant.

The Darwin plant has established a new benchmark in the LNG industry by being the first LNG plant to use an aeroderivative gas turbine as refrigerant compressor drivers and also the first to use evaporative coolers. The GE PGT25+[3] is comparable in power output to the GE Frame 5D gas turbine but has an ISO thermal efficiency of 41% compared to 29% for the Frame 5D. This improvement in thermal efficiency results in a reduction of fuel consumption which reduces greenhouse gas in two ways. First, there is a reduction in CO_2 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 Darwin LNG 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.

The Darwin 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.

The Darwin LNG plant has been designed to control nitrogen oxide emissions from the gas turbines by utilizing water injection into the combustor. Water injection allows the plant to control nitrogen oxide emissions while maintaining the flexibility to accommodate fuel gas compositions needed for various plant operating conditions, without costly fuel treatment facilities that may be needed for dry low NO_x combustors.

The Darwin plant uses a single LNG storage tank, with a working capacity of 188,000-m^3 which is one of the largest above ground LNG tanks constructed to date. A ground flare is used instead of a conventional stack to minimize visual effects from the facility and any intrusion on aviation traffic in the Darwin area. The plant uses vacuum jacketed piping in the storage and loading system to improve thermal efficiency and reduce insulation costs. MDEA with a proprietary activator is used for acid gas removal. This amine selection lowers the regeneration heat load required, and for an inlet gas stream containing over 6% carbon dioxide, this lower heat load results in a reduction in equipment size and a corresponding reduction in equipment cost.

The Darwin LNG Project was developed through a lump sum turn-key (LSTK) contract with Bechtel Corporation that was signed in April 2003 with notice to proceed for construction issued in June 2003. An aerial photo of the completed plant is shown in Figure 2. Details regarding the development of the Darwin LNG project have been provided by Yates [1, 2].

Figure 2. Aerial view of the 3.7 MTPA Darwin LNG plant– the world's first liquefaction facility to use high efficiency aeroderivative engines. The 188,000 m^3 storage tank and the 1350-meter jetty and loading dock can also be seen.

3.0 PLANT DESIGN

The Darwin LNG Plant utilizes the ConocoPhillips *Optimized Cascade*[SM] LNG Process. This technology was first used in the Kenai LNG Plant in Alaska and more recently at the Atlantic LNG in Trinidad (four trains), Egypt LNG (two trains), and a train in Equatorial Guinea, started up in 2007. A simplified process flow diagram of LNG plant was presented in Part 1 of this paper [3].

3.1 Implementation of the PGT25+ GT & Compressor Configurations.

The Darwin LNG compressor configuration encompasses the hallmark two-in-one design of the Optimized Cascade process, 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). All of the turbomachinery was supplied by GE Oil and Gas (Nuovo Pignone). Both the propane and ethylene trains had speed reduction gearboxes, with the methane being a direct drive. The high speed power turbine design speed is 6100 rpm.

[3] This engine utilizes a LM2500+ gas generator, coupled with a two stage high speed (6100 rpm) power turbine developed by GE Oil and Gas. The complete unit is designated a PGT25+

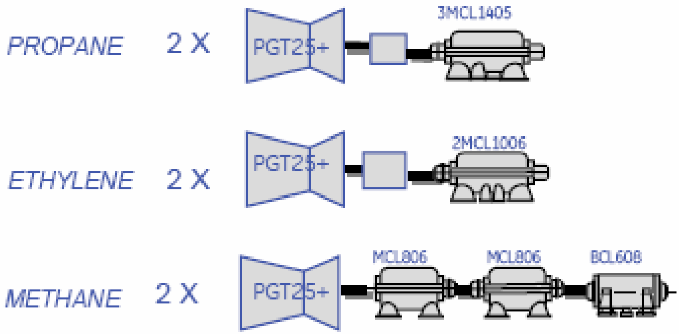

Figure 3. Overview of GT- Compressor configurations.

A view of the gas turbines inlet systems is shown in Figure 4. 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 mist eliminators.

Figure 4. Photograph of compressor trains at Darwin LNG. View of the inlet filter ducts and once through steam generators (heat recovery units) on four gas turbines on the left.

4.0 IMPLEMENTATION OF THE PGT25+ AERODERIVATIVE ENGINE

The PGT25+ engine used at the Darwin plant has a long heritage starting from the TF-39 GE aeroengine as shown in Figure 5. This highly successful aeroengine resulted in the industrial LM2500 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.

The LM2500+ 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%.

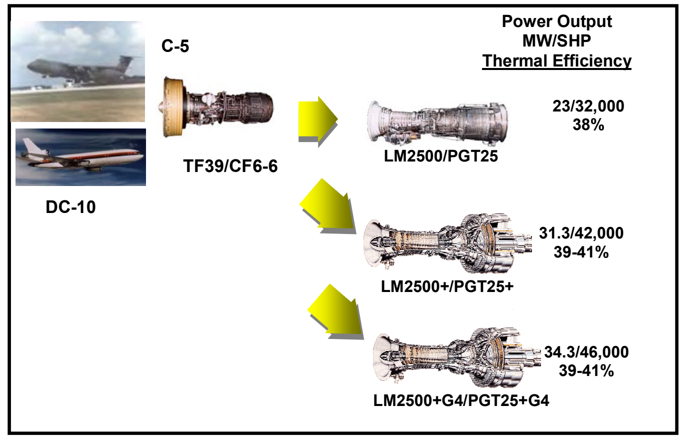

Figure 5. LM2500 engine evolution.

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 [4]. A view of the gas generator is shown in Figure 6.

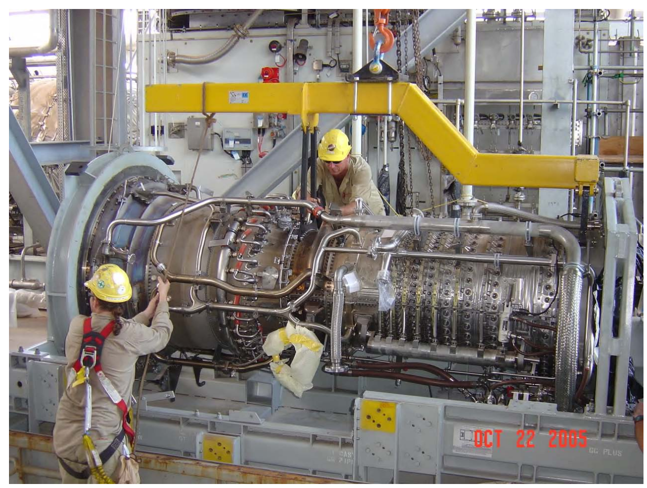

Figure 6. LM2500+ Gas generator being installed at Darwin LNG.

4.1 Description of the PGT25+ Gas Turbine

The PGT25+ consists of the following components:

4.1.1 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[4]. As reported by Wadia et al [4] 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%.

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

4.1.2 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.

4.1.3 High Pressure Turbine (HPT)
The PGT25+ 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.

4.1.4 High Speed Power Turbine
The PGT25+ 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.

4.1.5 Engine-mounted accessory gearbox driven by a radial drive shaft
The PGT25+ 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 7.

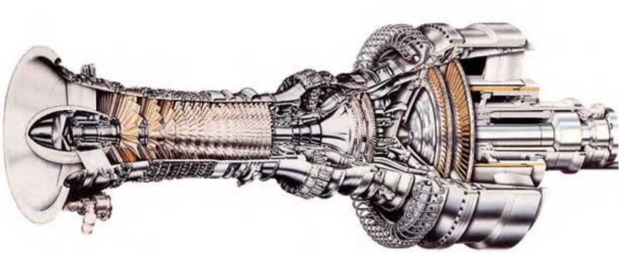

Figure 7. Overview of the PGT25+ gas turbine (Courtesy GE Energy).

4.2 Maintenance Plans and Experience

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 PGT25+ 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 NO_x target level[5].

4.3 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 is covered in references [5,6].

4.4 Potential Upgrades 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.

The LM2500+G4 is the newest member of GE's LM2500 family of aeroderivative engines. The engine retains the basic design of the LM2500+ but increases the power capability by approximately 10% without sacrificing hot section life. The modification increases the power capability of the engine by increasing the air flow, 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. The LM2500+ G4 engine is shown in Figure 8. Details of this variant may be found in [7].

[5] The level of water injection is a function of the NOx target level.

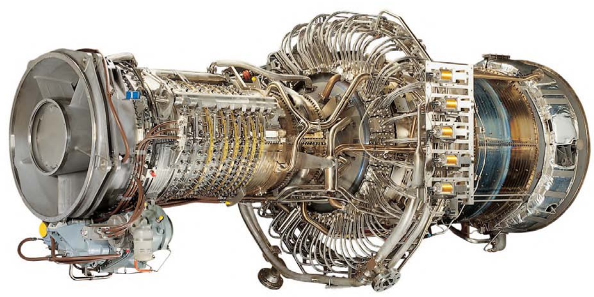

Figure 8. The uprated LM2500+ G4 engine -DLE variant (Courtesy GE Energy).

The growth in power of this variant compared to the base engine is shown in Figure 9.

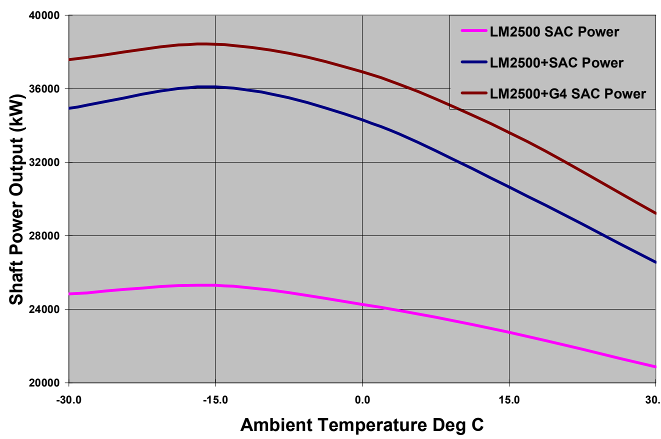

Figure 9. Growth of the LM2500+G4 variant (Courtesy GE Energy)

4.4 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 due to their higher specific work where the sensitivity can increase to well over 1% per °C. The impact of ambient temperature on the PGT25+ power and air flow is depicted in Figure 10.

As aeroderivative machines are more sensitive to ambient temperature, they benefit significantly from inlet air cooling. Darwin LNG utilizes media type evaporative coolers - another first for LNG refrigeration drivers. Details on media based evaporative cooling may be found in Johnson [8].

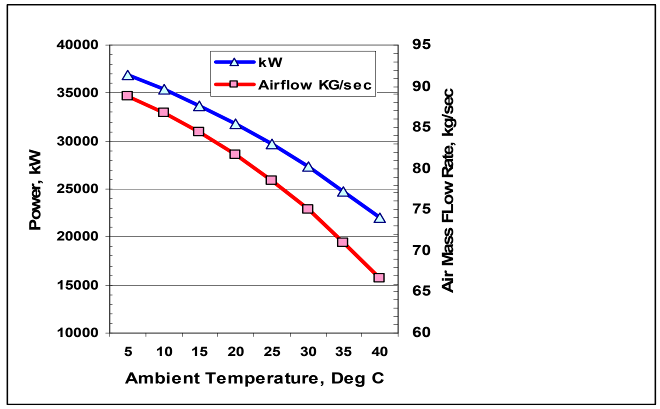

Figure 10. Variations in power output and air flow rate for the PGT25+ gas turbine.

Key advantages of power augmentation include:

- Boosts LNG production by lowering the gas turbine compressor inlet air temperature, increasing the air mass flow rate and power
- Improves the thermal efficiency of the gas turbine and results in lower CO_2 emissions

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 11. The relationship of relative humidity and dry bulb temperature is shown in Figure 12[6]. Details regarding the climatic analysis of evaporative cooling potential may be found in [9].

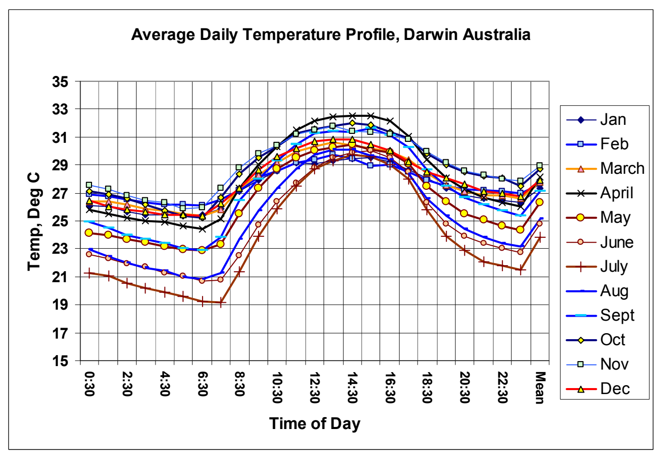

Figure 11. Temperature profile over time of day for 12 months in Darwin.

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

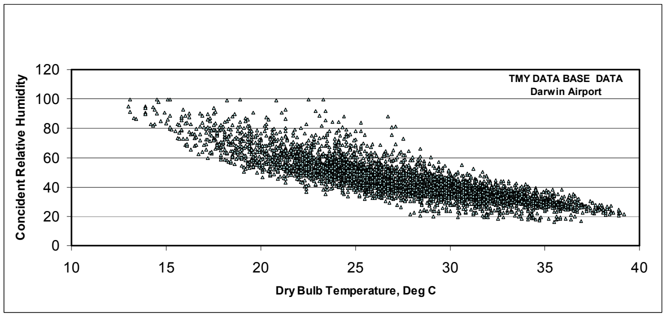

Figure 12. Relative Humidity vs. Dry Bulb Temperature, Darwin.

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:

Effectiveness = (T1DB – T2DB) / (T1DB – T2WB)

Where,

T1DB = Entering Air Dry Bulb Temperature
T2DB = Leaving Air Dry Bulb Temperature
T2WB = 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 at Darwin LNG 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 13.

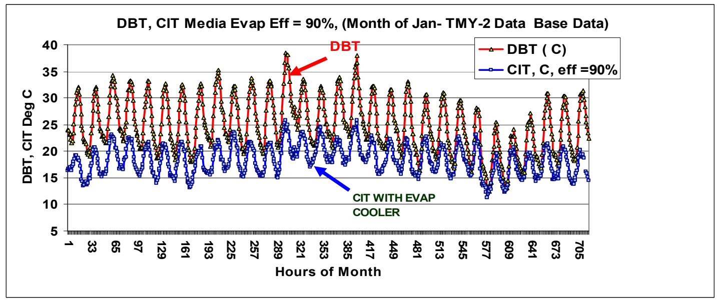

Figure 13. Calculated compressor inlet temperature (CIT) due to evaporative cooling over a summer month.

5.0 REFRIGERATION COMPRESSORS

5.1 Description of LNG Refrigeration Compressors- Design Process.

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. The design complexities, and compromises involved in the selection and design of refrigeration compressors will be covered in this section. 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 [10], Japikse [11]. Details on LNG Compressor design may be found in Meher-Homji, et al [12].

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 [13]. Another excellent reference is Japikse [11] which provides a qualitative graphical representation of design parameters on the performance, operating range and stress for centrifugal compressors. Both these references are valuable in helping engineers who are not aerodynamic specialists understand the design compromises that are needed.

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. Recognition should exist that turbocompressor aeromechanical design is a complex area where several advanced tools are available to optimize design. The design of LNG turbomachinery must be considered in an integrated manner so that all components including auxiliaries work well.

5.2 Compressor Selections

The configurations for the Darwin LNG Plant are as follows:

- Propane: 2 X PGT25+ Gas Turbine + Speed reduction GB + 3MCL1405 Compressor
- Ethylene: 2 X PGT25+ Gas Turbine + Speed reduction GB + 2MCL1006 Compressor (Back to back design)
- Methane: 2 X PGT25 + Gas Turbine + MCL806 + MCL 806 + BCL608 (i.e., three casing compressor)

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 14, which facilitates a down nozzle configuration for the compressors. This aids in the maintenance of the components as piping may be left in place during compressor dismantling

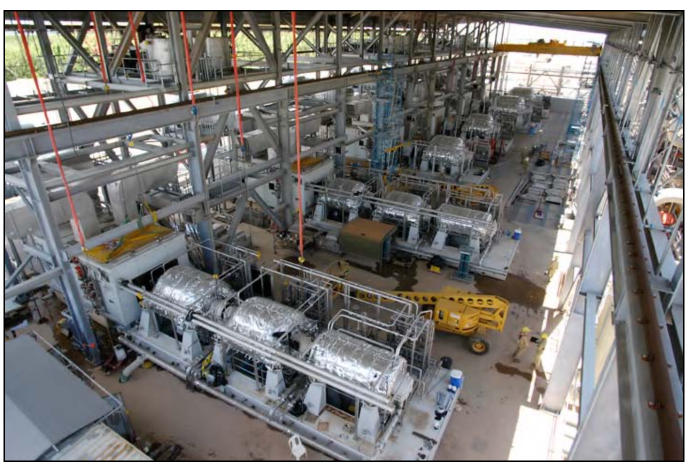

Figure 14. Photograph of compressor deck showing the six compressor strings. From front to back- 2 x methane compressors, 2 x ethylene compressors and 2 x propane compressors.

5.3 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.

5.3.1 Special Testing and Analysis on Ethylene Compressor

The ethylene compressor rotor, when hung from the drive end for modal testing is shown in Figure 15. The compressors have two sections in a back-to-back configuration, with five 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

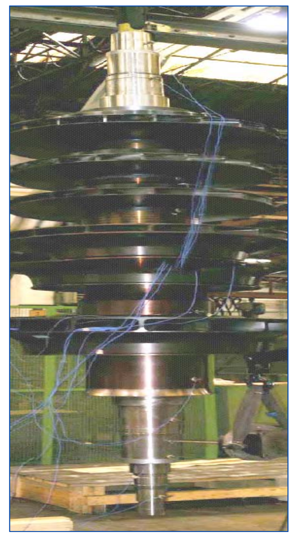

Figure 15.
Ethylene rotor undergoing modal testing.

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, 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.

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 16. 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 16). 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.

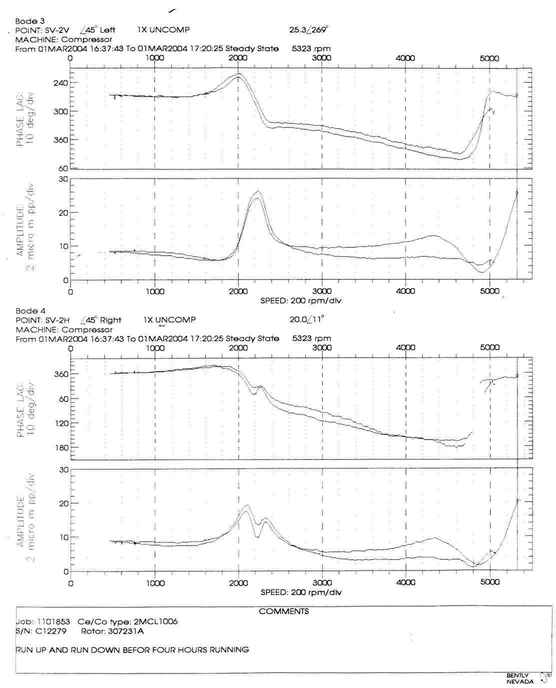

Figure 16. Ethylene rotor Bode plot showing growth in amplitude and phase shift when operating at MCS..

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 [14]. 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. [15].
- **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 15. 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 17 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 17 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

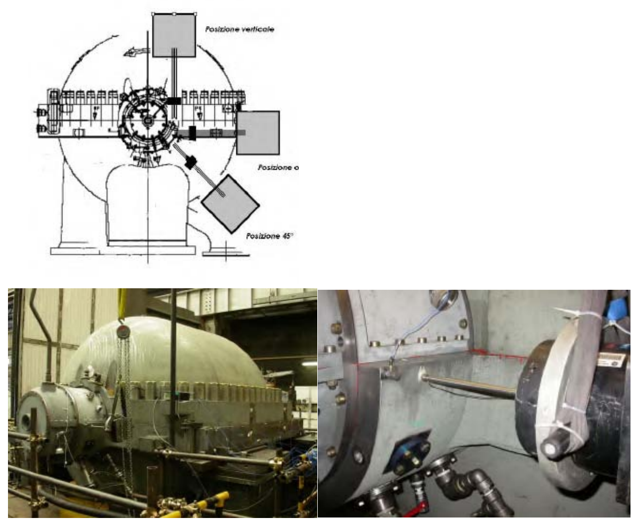

Figure 17. Support stiffness checks.

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.

6.0 OPERATING EXPERIENCE

6.1 Overall Results

Looking at the performance of the LNG plant over approximately two years of operation, all expectations have been met and exceeded. LNG production has exceeded predictions. The aeroderivative gas turbines that were the first application in a LNG plant have been successful. 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.

6.2 Issues of Integration to the Process with Respect to LM2500+ Trips / Lock out issues.

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 4 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.

Consequently, future projects will include:

- Joint evaluation of engine trip by the OEM, EPC and Process Licencor.
- Development / Evaluate a cause- effect diagram to understand and categorize trip parameters to minimize Type B trips.
- Try to move trip parameters from type C to A, as the logic of type B trips defeats the use of turning gears.

6.3 Operator Training System (OTS) and Dynamic Simulation

As reported by Valappil et al (16, 17] dynamic simulation has established itself as a valuable technology in the chemical process industries. 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 18. 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 Darwin LNG 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

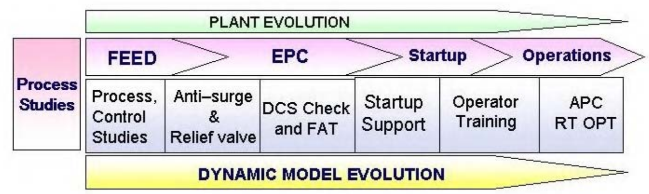

Figure 18. The use of dynamic simulation models used for Darwin LNG throughout the plant evolution process [16].

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 19 shows the trip scenario on a centrifugal compressor.

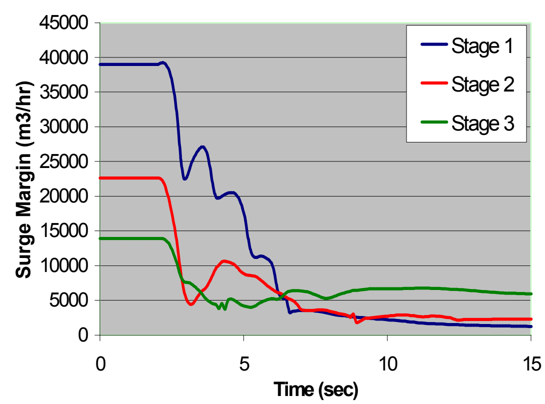

Figure 19. Response of a refrigeration compressor in a trip. Surge Margins for the three stages are shown as the machine speed drops. (Valappil et al, [17]).

7.0 CLOSURE

This paper has provided an overview of the application and operating experience of the world's first aeroderivative driven LNG Liquefaction facility. Part 1 of this paper described the underlying need for high thermal efficiency in the LNG market. The plant has been successfully operated for over two years and has met and exceeded its production goals.

Nomenclature

DLE	**Dry Low Emission**
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**
RH	**Relative Humidity**
SAC	**Standard Annular Combustor**
TG	**Turning Gear**

REFERENCES

[1] Yates, D., Schuppert, C., "***The Darwin LNG Project***," LNG14, 2005

[2] Yates, D., Lundeen, D., "***The Darwin LNG Project***, LNG Journal, 2005.

[3] Meher-Homji, C.B., Messersmith, D., Hattenbach, T, Rockwell, J., Weyermann, H.P., Masani, K., Thatcher, S., Maher, M, 2008, ***Aeroderivative Gas Turbines for LNG Liquefaction Plants- Part 1 : The Importance of Thermal Efficiency"***, ASME Paper No. GT2008-50839, ASME Turboexpo, Berlin, June 9-13, 2008.

[4] Wadia, A.R., Wolf, D.P., and Haaser, F.G., ***"Aerodynamic Design and Testing of an Axial Flow Compressor with Pressure Ratio of 23.3:1 for the LM2500+ Engine,"*** ASME Transactions, Journal of Turbomachinery, Volume 124, July 2002, pp 331-340.

[5] Meher-Homji, C. B., Chaker, M., and Motiwalla, H., 2001, ***"Gas Turbine Performance Deterioration,"*** *Proceedings of the 30th Turbomachinery Symposium*, Houston, Texas, September17-20, 2001.

[6] Meher-Homji, C. B., and Bromley A., 2004, "***Gas Turbine Axial Compressor Fouling and Washing***," *Proceedings of the 33rd Turbomachinery Symposium*, Houston, Texas, September 20-23, 2004.

[7] Badeer, G.H., ***"GE's LM2500+G4 Aeroderivative Gas Turbine for Marine and Industrial Applications,"*** GER 4250 (2005).

[8] Johnson, R.S., (1988), ***"The Theory and Operation of Evaporative Coolers for Industrial Gas Turbine Installations,"*** ASME Paper No: 88-GT-41, International Gas Turbine and Aeroengine Congress, Amsterdam, Netherlands, June 5-9, 1988

[9] Chaker, M., Meher-Homji, C.B., (2006) **"*Inlet Fogging of Gas Turbine Engines- Detailed Climatic Analysis of Gas Turbine Evaporative Cooling Potential for International Locations*,"** ASME Transactions- Journal of Engineering for Gas Turbines and Power, October 2006, Vol 128.

[10] Aungier, R. H., 2000, ***Centrifugal Compressor—A Strategy for Aerodynamic Design and Analysis,*** The American Society of Mechanical Engineers, New York, New York.

[11] Japikse, D., 1996, ***"Centrifugal Compressor Design and Performance,"*** Concepts ETI.

[12] Meher-Homji, C.B, Matthews, T, Pelagotti, A, Weyermann, H.P, (2007), ***Gas Turbines and Turbocompressors for LNG Service***, Proceedings of the 2007 Turbomachinery Symposium, September 11-13, 2007, Houston, Texas. Turbomachinery Laboratory, Texas A&M University.

[13] Sorokes, J. M., 2003, ***"Range Versus Efficiency—A Dilemma for Compressor Designers and Users,"*** Proceedings of PID Industrial and Pipeline Compression Sessions, ASME-IMECE 2003, Paper No. IMECE2003-4422.

[14] Vannini, G., Smalley, A.J., Hattenbach, T, Weyermann, H.P., Hollingsworth, J.R**.**, ***"Calibrating a Large Compressor's Rotordynamic Model- Method and Application,"*** 7th IFFoMM Conference on Rotordynamics, Vienna, Austria, 25-28 September 2006.

[15] Smalley, A. J., Pantermuehl, J. P., Hollingsworth, J. R., and Camatti, M., 2002, ***"How Interference Fits Stiffen the Flexible Rotors of Centrifugal Compressors***," *Proceedings of the IFToMM Sixth International Conference on Rotor Dynamics*, Sydney, Australia.

[16] Valappil, J., Mehrotra, V., Messersmith, D., and Bruner, P., 2004a, **"*Virtual Simulation of LNG Plant*,"** *LNG Journal,* January/February.

[17] Valappil, J., Messersmith, D., and Mehrotra, V., 2004b, ***"Dynamic LNG,"*** *Hydrocarbon Engineering,* October.