

GAS TURBINES AND TURBOCOMPRESSORS FOR LNG SERVICE

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ABSTRACT

The international liquefied natural gas (LNG) trade is expanding rapidly. Projects are being proposed worldwide to meet the industry forecasted growth rate of 12 percent by the end of the decade. LNG train designs in the coming years appear to fall within three classes, having nominal capacities of approximately 3.5, 5.0, and 8.0 million tons per annum (MTPA). These designs may coexist in the coming years, as individual projects choose designs that closely match their gas supplies, sales, and other logistical and economic constraints.

The most critical component of an LNG liquefaction facility is the refrigeration compressors and their drivers. These represent a significant expense and strongly influence overall plant performance and production efficiency. The refrigeration compressors themselves are complex and challenging to design due to high Mach numbers, large volume flows, low inlet temperatures, and complex sidestream mixing. Drivers for these plants include gas turbines that range in size from 30 MW units to large Frame 9E gas turbines. Aeroderivative engines have also been recently introduced. This paper comprehensively covers the design, application, and implementation considerations pertaining to both LNG plant drivers and compressors. The paper does not focus on any particular LNG process but just addresses turbomachinery design and application aspects that are common to all processes. Topics cover key technical design issues and complexities involved in the turbomachinery selection, aeromechanical design, testing, and implementation. The paper attempts to highlight the practical design compromises that have to be made to obtain a robust solution from a mechanical and aerodynamic standpoint.

INTRODUCTION AND OVERVIEW OF LNG TECHNOLOGY

The LNG Value Chain and LNG Market

The international liquefied natural gas (LNG) trade is expanding at a rapid pace. The estimated consumption of natural gas in 2006 was over 100 trillion cubic feet (Tcf), which represents a 60 percent increase over the past 20 years. By 2025 it is estimated that this consumption will be around 156 Tcf and account for 25 percent of the predicted world demand. Natural gas is brought to market either by traditional pipeline or by an LNG supply chain. For short distances between the natural gas source and the end user, pipelines are more economical but as this distance increases, LNG becomes more economically viable as shown in Figure 1. The crossover point for an offshore pipeline would be even less due to the higher construction costs. This figure is merely an indicative one and project specific criteria and prevailing economics would have to be applied on a case-by-case basis.

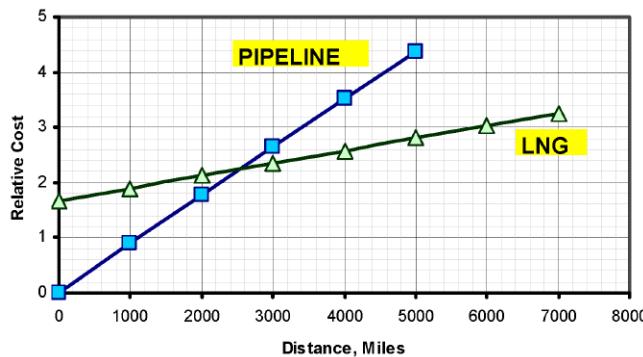


Figure 1. LNG Versus Onshore Pipeline Economics (Indicative Data Only).

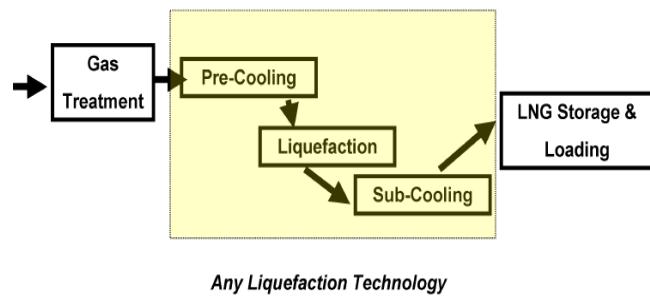
Converting natural gas to LNG is accomplished by chilling and liquefying the gas to a temperature of -160°C (-256°F). When liquefied, the natural gas's volume is reduced to 1/600th of its standard condition volume. This allows the efficient transportation of gas using specialized LNG tankers. The size of a liquefaction facility is usually stated in millions of tons per annum (MTPA). One metric ton of LNG is equivalent to 52 million Btu of gas (54.8 GJ). Useful conversions relating to LNG energy are provided in APPENDIX A.

The overall LNG supply chain includes:

- Natural gas production (wells) and transportation to the liquefaction plant.

- Liquefaction by chilling of the gas (it is this section that is the focus of the paper).
- Transportation from the liquefaction plant to the receiving terminal by LNG tanker.
- Regassification (at the receiving terminal in the receiving country).
- Distribution of gas to users via pipeline.

LNG projects are extremely capital intensive. The liquefaction plant can account for as much as 50 percent of the overall project value and is, by far, the most expensive component of the chain. The focus of this paper is on the turbomachinery that is used to liquefy the natural gas. The liquefaction process is shown generically in Figure 2. The refrigeration drivers and compressors are exceedingly critical in terms of their impact on plant performance and production efficiency. Depending on the specific process technology utilized, the refrigerants can be pure components or mixed refrigerants or a combination of these. The refrigeration compressors represent among the most complex and challenging compressor designs due to the high Mach numbers, large volume flows, and low inlet temperatures and complex sidestream flows.



Any Liquefaction Technology

Figure 2. LNG Liquefaction Plant Block Diagram.

LNG Processes

The paper does not focus on any particular LNG process technology but just on the turbomachinery aspects that are common to all processes. To get a feel for the types of services included, key processes available in the marketplace are covered below.

The most commonly used LNG liquefaction processes (Shukri, 2004) over the past decade are:

- *APCI propane precooled mixed refrigerant and AP-X*—This process uses two or three fluids: mixed refrigerant (C1, C2, C3, N2) and propane (commonly known as C3/MR), with nitrogen being added as another loop for the AP-X process).
- *ConocoPhillips Optimized CascadeSM LNG process*—This process uses three refrigerants: propane, ethylene, and methane. Each refrigerant has a parallel compression train.
- *Multifluid cascade process*—MFCP uses three mixed refrigerants for precooling, liquefaction, subcooling; all three are made up of components selected from C1, C2, C3, N2.
- *Shell DMR*—DMR uses dual mixed refrigerant process. The precooling is accomplished by a mixed refrigerant (mainly C2 and C3) rather than pure C3.

LNG Market Activity

As described by Houston (2007), the LNG industry has entered a transformational phase where, in recent years, the rate of growth has increased with a growth of 13 percent per annum expected through 2015. The LNG industry evolution over time and projections to 2015 are shown in Figure 3. A detailed review of the LNG industry can be found in Harris and Law (2007), and Houston (2007). An overview of the industry from 1989 to 2007 has been made by Glass and Lowe (2007).

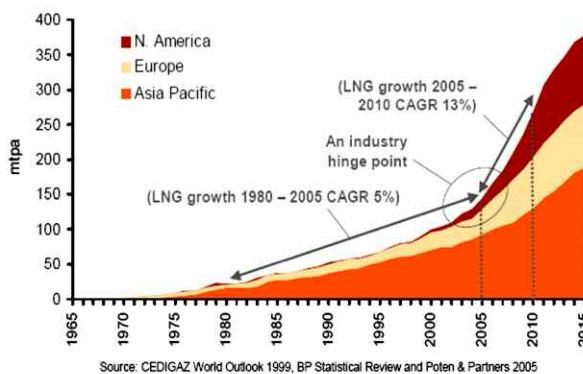


Figure 3. Growth and Projections of the LNG Industry. (Houston, 2007, Courtesy BG Group)

While there were only three new grass roots LNG plants in the 1980s, the 1990s saw the startup of six new grass roots projects in Malaysia, Oman, Qatar, Nigeria, and Trinidad. In addition, several existing facilities were revamped and expanded. Grass roots plants have been started up or are shortly to be started in Egypt, Australia, Equatorial Guinea, Indonesia, Norway, Russia, and Qatar. New projects are being developed in Nigeria, Angola, Qatar, and other locations. LNG train designs in the coming decade appear to fall within three categories, having nominal capacities of approximately 3.5, 5.0, and 8.0 MTPA capacities. These designs may coexist in the coming years, as different individuals choose the size, which matches their gas supplies, sales, and other logistical and economic constraints. Designs suggested for even larger capacities of 10 to 12 MTPA may be found in Kaart, et al. (2007). Details on the evolution of the LNG market may be found in Avidan, et al. (2002, 2003), and Wood and Mokhatab (2007).

As the LNG trade increases owners keep looking for ways to lower costs by benefitting from economies of scale. As plant capacities grow, this effort has concentrated on building larger single LNG train plants. LNG train size has been increasing from a typical 1.5 MTPA in the 1970s to a typical 2.5 MTPA design in the mid 1990s as shown in Figure 4.

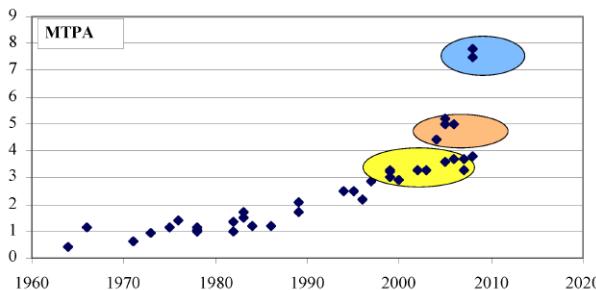


Figure 4. Increase in LNG Plant Sizes over Time.

LNG PLANT DRIVER OPTIONS

LNG Driver Choices Available

The three types of drivers available for LNG refrigeration compressors are:

- Steam turbines
- Gas turbines
- Electric motors

Prior to 1989, LNG plants utilized steam turbine drivers. As steam turbines could be designed and sized to uniquely fit specific power requirements, they were initially very popular. The world's first plant to use gas turbines was a plant in Kenai, Alaska (in 1969), that utilized six Frame 5 drivers. Currently the trend is to

use gas turbines as their power outputs, efficiencies, and overall efficiency have been steadily improving. Over the past decade, there is also an interest in an all electric drive solution for LNG with one plant being currently implemented. The general evolution of the market is indicated in Figure 5.

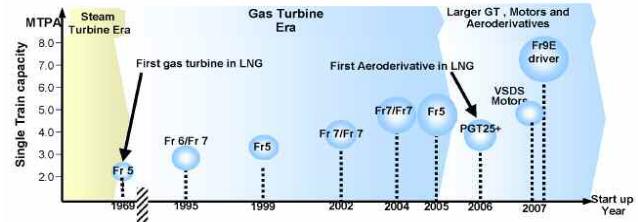


Figure 5. Evolution of Drivers for LNG Refrigeration Compressors.

Gas turbines are now the established driver of choice in the LNG industry and have several advantages over steam turbines. These include:

- Smaller plot space.
- Generally shorter delivery time.
- Lower transportation costs.
- Lower installation costs.
- Lower foundation costs.
- No need for expensive boiler feed water treatment.
- No need for cooling water.

Driver Selection Criteria

There are several key criteria that have to be considered in the selection of a driver:

- *Driver power capability*—The power developed by the driver must be appropriate for the worst case situation. In the case of a gas turbine this would take into account the highest ambient temperature, and a level of performance degradation. Typically conservative power margins are applied but this is an area that needs careful consideration especially with the application of dry low NOx or dry low emission machines. This will be discussed in detail in the gas turbine section.
- *Reliability and availability experience*—This is obviously a very important criterion, as the viability of the LNG plant is a strong function of the availability of the refrigeration equipment. Gas turbine vendors have extensive historical data relating to reliability and availability. The reliability takes into account forced outages, while the availability parameter includes scheduled maintenance and the planned maintenance schedules.
- *Capital cost*—The competitiveness and viability of an LNG project are very sensitive to the installed cost of the liquefaction facility, consequently the capital cost of the driver-compressor block is of critical importance. The cost of power generation for the facility must be also considered in an integrated fashion as the selection of driver type (single or dual shaft) will impact power generation requirements.
- *Technical issues*—These include the operating speed of the driver, driver speed variability, starting torque capability, and controllability.

LNG Plant Thermal Efficiency Considerations—Impact on Turbomachinery

With the increased emphasis on limiting global warming, and as fuel costs increase, the issue of overall thermal efficiency of the LNG facility has become more important. The thermal efficiency of an LNG facility depends on numerous factors such as gas composition, inlet pressure and temperature, and even more obscure factors such as the location of the loading dock relative to

the liquefaction process. Gas turbine selection, the use of waste heat recovery, ship vapor recovery, and self-generation versus purchased power all have a significant effect on the overall thermal efficiency of the LNG process. Process flexibility and stability of operation are issues of paramount importance and must be incorporated into the considerations regarding thermal efficiency as the value of a highly efficient process is diminished if plant reliability and availability are sacrificed.

Yates (2002) has provided a detailed treatment of the design lifecycle and environmental factors that impact plant thermal efficiency, such as feed gas characteristics, feed gas conditioning, and the LNG liquefaction cycle itself. Some of the key elements of this discussion are provided below.

A common consideration in evaluating competing LNG technologies is the difference in thermal efficiency. The evaluation of thermal efficiency tends to be elusive and subjective in that each project introduces its own unique characteristics that determine its optimum thermal efficiency based on the strongest economic and environmental merits for the project. Different technologies or plant designs cannot be compared on thermal efficiency without understanding and compensating for the unique differences of each project.

The definition of thermal efficiency also has proven to be subjective depending on whether an entire plant, an isolated system, or item of equipment is being compared. Thermal efficiency, or train efficiency, has been expressed as the ratio of the total higher heating value (HHV) of the products to the total HHV of the feed. The use of this definition fails to recognize the other forms of thermodynamic work or energy actually consumed by the process. For example, if purchased power and electric motors are used for refrigeration and flashed gas compression, this definition would not account for the work done by these motors. When evaluating the benefits of achieving a high thermal efficiency with a specific LNG plant design, a true accounting of all of the energy being consumed in the process must be considered.

Turndown capabilities of an LNG process (the ability to operate at part load conditions) also need to be considered when thermal efficiency and lifecycle comparisons are being made. Thermal efficiency comparisons are typically based on the process operating at design conditions. In an actual plant environment, this design point is elusive and an operator is always trying to attain an optimal point where the plant will operate at its peak performance under prevailing conditions. As the temperature changes during the day, impacting the performance of the air coolers, the turbines, or the process fluid and equipment, the operator needs to continually adjust plant parameters to achieve optimal performance. Designing a plant to allow an operator to continually achieve this optimum performance point will impact the overall thermal efficiency of the plant and lifecycle costs.

The efficiency of an LNG process is dependent on many features. The two most significant ones are the efficiency of heat exchange and the turbomachinery efficiency. The heat exchange efficiency is a function of the process configuration and selection of the individual heat exchangers, which sets temperature approaches. The turbomachinery efficiency depends on the compressor and gas turbine efficiencies.

The maturity of liquefaction processes on the market has now approached a point where changes in duty curve no longer represent the greatest impact. Two developments that have a significant impact on efficiency are the improvement in liquefaction compressor efficiency and the availability of high efficiency gas turbine drivers. Compressor efficiencies are well in excess of 80 percent and high efficiency gas turbines are available with simple cycle thermal efficiencies of above 40 percent. A detailed treatment of LNG plant process efficiency has been made by Ransbarger (2007).

Environmental Greenhouse Gas Considerations

From the discussion above, one can see that the selection of the gas turbine plays an important role in the efficiency, greenhouse gas emissions, and flexibility under various operating conditions.

Where high fuel costs are expected, the selection of a high efficiency driver becomes a strong criterion in the lifecycle cost evaluation. However, LNG projects are developed to monetize stranded gas reserves, where the low-cost fuel has favored heavy duty gas turbines. This situation is however changing and the value of gas is now growing. Further, in situations where the gas is pipeline or otherwise constrained, there is a clear benefit in consuming less fuel for a given amount of refrigeration power. In such cases, a high efficiency gas turbine solution where the saved fuel can be converted into LNG production can result in large benefits.

Aero derivative gas turbines achieve significantly higher thermal efficiencies than industrial gas turbines as shown in Figure 6. This figure shows the engines' thermal efficiency versus specific work (kW per unit air mass flow) for commonly used engines used for LNG service. The higher efficiency of an aero derivative can result in a 3 percent or greater increase in overall plant thermal efficiency. Further, there is a significant improvement in plant availability as a result of the ability to completely change out a gas generator (or even a complete turbine) within 48 hours versus 14 or more days that would be required for a major overhaul of a heavy duty gas turbine.

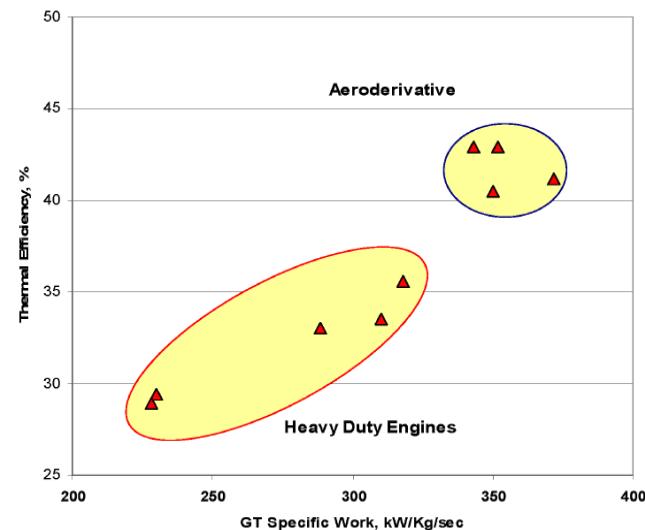


Figure 6. Map of ISO Thermal Efficiency Versus Specific Work of Commonly Used Frame Drivers and Aero derivative Engines.

If a high efficiency gas turbine is utilized, the improvement in thermal efficiency results directly in a reduction of specific fuel required per unit of LNG production. This reduction in fuel consumption results in a reduction in CO₂ emissions. This impact is depicted in Figure 7, which shows relative CO₂ emissions for various drivers. A similar beneficial greenhouse gas reduction comes from the use of waste heat recovery for various heating requirements within the plant. The use of this heat recovery eliminates greenhouse gas emissions that would have been released had gas fired equipment been used. The result of using waste heat recovery equipment can result in a reduction in greenhouse gases by approximately 9 percent of the total emissions without the installation of this equipment. A treatment of efficiency and emissions in the context of base load LNG plants has been made by Nelson, et al. (2001).

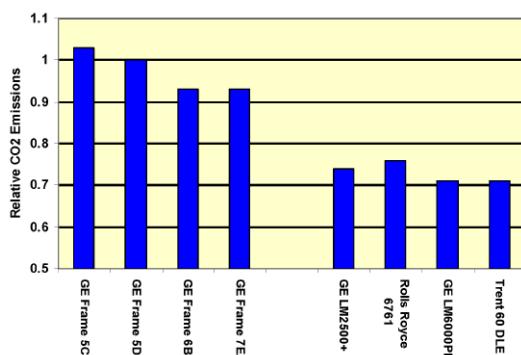


Figure 7. Relative CO₂ Emissions from Different Classes of Gas Turbines.

Combined cycle drivers provide an attractive design alternative for an LNG plant and have been studied but not implemented as of yet. Qualls and Hunter (2003) described how a combined cycle successfully reduces capital costs and increases thermal efficiency. This approach would include the use of mechanical drive gas turbines with heat recovery and mechanical drive steam turbines. The thermal efficiency of this approach is superior to most simple cycle plants. The main advantage of the combined cycle drivers is the reduction in fuel consumption by increasing cycle efficiency. Figure 8 (Tekumalla, et al., 2007) demonstrates the value of converting fuel gas savings into LNG for a 5.0 MTPA LNG plant.

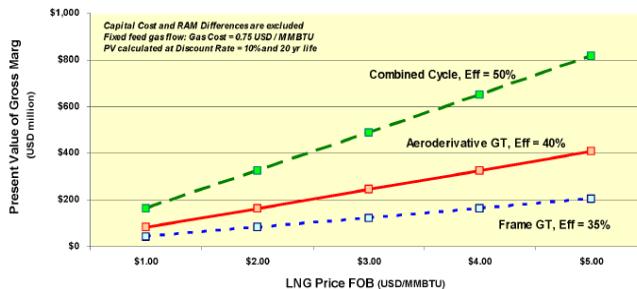


Figure 8. Value of converting Fuel Gas Savings into LNG. (Courtesy Tekumalla, et al., 2007)

GAS TURBINE DRIVERS

Mechanical Drive Gas Turbines

There are a wide range of gas turbines available on the market today that have been applied for pipeline and power generation service. For LNG service, the envelope is currently bracketed by the Frame 9E gas turbine in terms of power, and by aeroderivative engines for high simple cycle thermal efficiency. Salient parameters of some key gas turbines that could be used as LNG drivers are presented in Table 1. The concept of gas turbine specific power is a valuable one in evaluating gas turbines as several operational parameters are linked to this. A general trend of specific power versus power level is shown in Figure 9. The general trend as power increases is for higher specific powers. Aeroderivative engines have a higher specific power compared to their heavy duty counterparts.

Table 1. Salient Parameters of Gas Turbine Drivers.

| Gas Turbi | ISO Rate Power, k | Thermal Eff, % | Pressure Ratio | Air Flow Rate Kg/Sec | Number of Shafts | Output Shaft Speed |
|---------------|-------------------|----------------|----------------|----------------------|------------------|--------------------|
| PGT25+G4 HSPT | 34,328 | 41.3 | 24.4 | 88.43 | 2 | 610 |
| Frame 5D | 32,580 | 29.4 | 10.8 | 141.36 | 2 | 4670 |
| Frame 5E | 31,998 | 36 | 17 | 102 | 2 | 6100 |
| Frame 6B | 42,910 | 33 | 12.2 | 140.9 | 1 | 5111 |
| LM6000 | 44,261 | 42.8 | 29 | 128 | 2 | 3600 |
| Frame 7EA | 86,701 | 33 | 12.6 | 296 | 1 | 3600 |
| Frame 9E | 129,401 | 34.4 | 12.7 | 412 | 1 | 3000 |
| LMS100 | 100,000 | 44 | 40 | 207 | 3 | 3000/3600 |

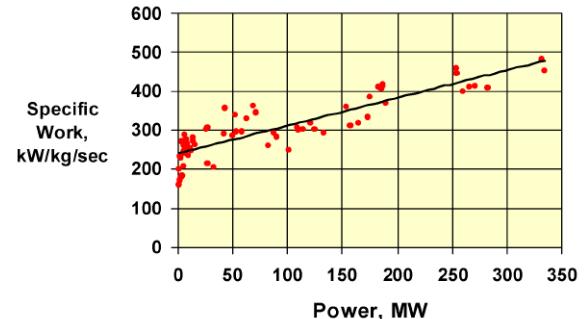


Figure 9. Trend of Gas Turbine Specific Work and Output.

Selection Considerations

A wide range of factors goes into the selection of a gas turbine driver. Key considerations include:

LNG Plant Production and Thermal Efficiency-Driver Efficiency and Reliability

LNG production is directly linked to refrigeration power and overall fuel consumption. CO₂ emissions are directly linked to the gas turbine's thermal efficiency. Depending on how the project values fuel, this can be a significant issue. The use of a variable area second-stage nozzle (located between the high pressure gas generator turbine and the low pressure power turbine) in the case of a two shaft gas turbine allows the area and hence the backpressure on the high pressure turbine to be modulated thus allowing the enthalpy drop between the two turbines to be controlled. This result is better part load efficiency and better control of speed with sudden load swings.

Multiple Shaft Versus Single Shaft Gas Turbines

Gas turbines that have traditionally been used for power generation application are typically single shaft machines with a limited speed range. Drivers such as the Frame 6, Frame 7, and Frame 9 fall into this category. These machines are incapable of starting up a large compressor string without the help of large variable speed drive starter motors. Split shaft machines may be heavy duty (such as the Frame 5D) or aeroderivative engines such as the LM2500+ that have free power turbines that allow very high startup torques. Some larger drivers such as the LM6000 are multispool machines, but with no free power turbine, that still exhibit a large speed range and excellent startup torque capability. The configurations of a single and split shaft gas turbine are shown in Figure 10.

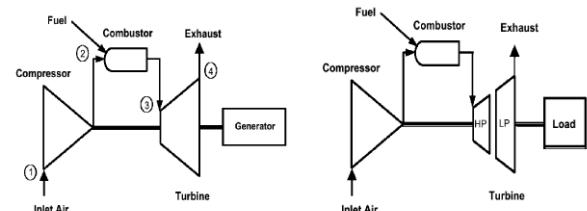


Figure 10. Single Shaft and Split Shaft Gas Turbine Configurations.

In a gas turbine engine, the operating point is defined by the match between the compressor and the turbine. A match point is simply a set of operating conditions (pressures, temperatures, and flows) where the compressor and turbine can work in unison and in equilibrium.

Matching is based on the compatibilities of flow, work, and speed. This means:

- The compressor work must match the work output of the turbine that drives it. In aeroderivative engines, the gas generator turbine

drives the compressor. In the case of a three-spool engine, the low pressure turbine work must match the low pressure compressor work and the high pressure turbine the high pressure compressor work. In a single shaft gas turbine such as the Frame 6B, 7EA, or 9E, the turbine work must equal the sum of the output and the gas turbine axial compressor work.

- The flow rates must be compatible because gas turbines are continuous flow machines. Air bleeds and fuel flow must be taken into account.
- The speeds of the compressor and its driving turbine must, by definition, be the same. In multiple spool engines, the free spools must be in equilibrium.

In the case of a single shaft gas turbine used for power generation, the operating point at a given temperature is along a corrected speed line as the mechanical speed is locked to the grid. After attaining full speed no load conditions (FSNL) the turbine is loaded by increasing fuel flow (i.e., increasing turbine inlet temperature) causing the operating point to move up the speed line. When these machines are applied to LNG compressor drives, there is a small speed range of approximately 5 percent available. In the case of a split shaft gas turbine such as the Frame 5D or LM2500+ load changes involve changes in the speed of the gas generator spool and therefore the trajectory of operating points is not along a constant speed line. With split shaft or multispool gas turbines, significant speed flexibility exists.

In the case of the turbine map, the flow chokes at a certain expansion ratio (for a given turbine inlet temperature) and thereafter the turbine flow function ($mV/T/P$) stays flat. The swallowing capacity of the gas turbine is Mach number limited and therefore a function of the turbine inlet temperature. Most gas turbines run choked with a choked nozzle and this defines the operating point on the compressor characteristic. Thus, for a given speed, the compressor operating pressure ratio can be calculated from the turbine "resistance." This calculation requires an iterative computation knowing both the turbine and compressor maps. Obviously as the flow is choked, the quantum of flow will be defined by the turbine inlet temperature (due to the influence of temperature on the Mach number).

The optimum operation at design conditions will be obtained with a certain hot effective nozzle area. For a power turbine to match a gas generator, the overall turbine effective area has to be equivalent to the hot effective nozzle area. Variations in effective area from the standard will modify performance. If the effective area is oversized by 1 percent, the gas generator cycle temperature is reduced at constant gas generator speed and a power reduction of between 2 and 4 percent is typical. If the effective area is undersize by 1 percent, the gas generator speed is reduced at constant temperature and a similar reduction in power occurs (Blauser and Gulati, 1984).

In mechanical drive split shaft gas turbines there is an optimal power turbine speed (optimal U/C ratio). Operating away from this speed will result in a loss in terms of power and efficiency. The power turbine optimal speed is a function of ambient temperature. Assuming the design is for International Organization for Standardization (ISO) conditions, as the ambient temperature increases the power turbine optimal speed drops. If adjustment of turbine geometry is available (variable power turbine inlet vanes) then the effect can be mitigated resulting in a flatter efficiency curve.

Details on gas turbine matching may be found in Saravanamuttoo, et al. (2001), Kurz and Brun (2000), and Meher-Homji, et al. (2001). Details on gas turbine performance are available in Walsh and Fletcher (1998).

Aeroderivative and Heavy Duty Gas Turbines

Both heavy duty and aeroderivative engines have their applications as LNG drivers and to some extent the selection is dependant on the LNG process technology. Some technologies favor single train

larger machines while others may favor redundant smaller machines. Both types of drivers can be successfully used in LNG applications. Fundamentally, aeroderivative units tend to operate at higher pressure ratios and firing temperatures as indicated in Figure 11.

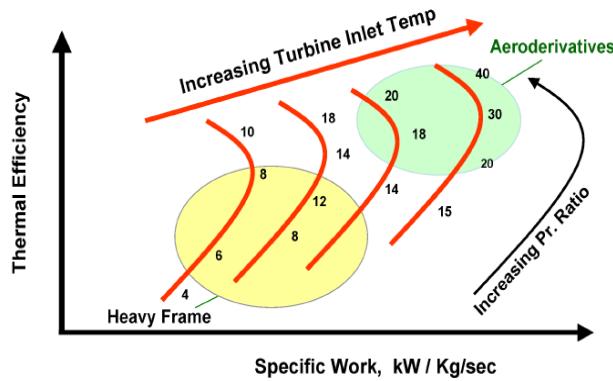


Figure 11. Differentiation between Heavy Duty and Aeroderivative Gas Turbines.

While a perception exists that heavy duty machines are more rugged, extensive mechanical drive experience with aeroderivatives (both offshore and onshore) have shown that good availabilities have been obtained even under hostile operating conditions. Site maintenance of aeroderivatives is more complex with engines typically being shipped to an authorized repair depot for service. On the other hand, heavy duty units can be maintained in situ, though the time required to remove the engine and perform the overhaul is considerably longer than that of an aeroderivative engine. The high power to weight ratio of an aeroderivative engine may be of importance in the event a floating LNG facility is being planned.

GT Site Performance—

Capacity and Efficiency

Gas-turbine (GT) ratings are traditionally established with a single operating point at ISO ambient conditions. ISO conditions are defined as a temperature of 60°F (15°C) zero inlet and exhaust losses and consider a relative humidity of 60 percent and allow that the maximum continuous speed is at least 105 percent of nominal rated speed.

During selection of a gas turbine all site conditions have to be taken into account as the plant performance will be dependant on the site temperatures predominantly.

Speed Range and Transient Characteristics

While two shaft gas turbines have an inherent speed range (typically 70 to 105 percent), single shaft gas turbines have a very limited range typically around 5 percent. The speed range of the MS6001B for example, is bounded on the high end by encroachment limits on first-stage compressor blading resonance conditions and at the low end by third-stage turbine bucket resonance conditions. The MS7001EA upper end limit is determined by the stress-temperature design life criteria for the turbine section buckets, while the lower-end limit is determined by compressor stall margin. Compressor stall margin considerations also preclude continuous operation on either machine at any lower speed points (Ekstrom and Garrison, 1994).

Startup and Shutdown Issues

While split shaft turbines have excellent torque-speed characteristics allowing easy startup of compressor strings, the larger single shaft gas turbines are very limited in startup torque and consequently a starter motor has to be utilized.

Typically two shaft gas turbines will allow a rapid startup due to the torque capability of the engine. With a single shaft driver, depressurization of the compressor string is typically required to enable a startup. This can be avoided by the use of very large starter motors as has been done on a Frame 9E project (Saulsbury, et al., 2007) that allow startup under full settle out pressure conditions. Most plants utilizing single shaft units do however have to depressurize prior to start.

Load Coupling Selection

The coupling must support the LNG train rotordynamics and torsional analysis and also accommodate thermal growth misalignment as the turbine heats up. The coupling between the gas turbine and the driven equipment must meet several requirements. First it must continuously transmit the required full-load torque and comfortably withstand the transient peak torques associated with upset conditions. It may also be required to transmit or isolate thrust loads and must accommodate alignment shifts due to thermal growths. The coupling has to complement the torsional and lateral rotordynamics of the full LNG string.

Accessory Systems

The selection of accessory systems is of critical importance in terms of the operational reliability of the LNG Plant. A machine such as the Frame 5D will utilize its own mineral oil system for the compressors. In the case of large gas turbine drivers such as the Frame 7 or Frame 9, independent API 614 (1999) lube oil systems can be utilized to service both the gas turbine and driven equipment. With aeroderivative engines, the gas generator utilizes a synthetic oil system, and the power turbine and remaining driven equipment would have their own mineral oil system.

Emissions—Water and Steam Injection and DLN/DLE

While emission requirements are project and site specific, the general trend is to limiting NOx emissions. This can be done by the use of lean head end combustor for certain gas turbines or by water injection, or by the utilization of dry low combustor technology known as DLN (dry low NOx) for the frame units and DLE (dry low emission) for the aeroderivative units. The general trend is for plants to have dry control technologies, but fuel considerations (fuel switching, and Wobbe numbers) may impact this decision.

Typical Gas Turbine Configurations

Some typical configurations utilizing a range of gas turbines including Frame 5, 7EA, and 9E are provided below. Very often combinations of drivers are used such as a Frame 7EA and a Frame 6B.

- *Configurations using mixed refrigerants*—Configurations here include either two Frame 7EA machines or one Frame 6B and one Frame 7EA. For example a Frame 6 would drive a propane compressor, while the Frame 7EA would drive a mixed refrigerant compressor resulting in a 3 to 4 MTPA plant. The gas turbines would have starter-helper variable speed drive motors. In some variations of this configuration one could have two Frame 7EA turbines: one driving mixed refrigerant compressors and the other Frame 7EA driving a propane and a stage of mixed refrigerant. In some cases, the refrigerants would include propane, mixed refrigerant, and nitrogen. For example a plant using three Frame 9E units with large helper motors driving propane, mixed refrigerant, and nitrogen to produce approximately 8 MTPA.
- *Configurations utilizing pure refrigerants*—A typical configuration here would be six Frame 5D or six LM2500+ gas turbines in a 2+2+2 configuration driving propane, ethylene, and methane compressors, respectively. Such configurations can derive 3 to 4 MTPA and larger plants have been developed using additional drivers. Alternatively, heavy duty drivers can also be used (2 × Frame 7 + 2 × Frame 6) to derive larger outputs.

Design Information for Mechanical Drive Gas Turbines Used in LNG Service

Some of the salient design features of commonly used gas turbines in the LNG sector are described here.

Frame 5D Gas Turbine

The Frame 5D is a heavy duty two shaft gas turbine that has been widely applied as a driver for LNG plants. Some of the salient design features include:

- 32 MW, thermal efficiency = 29.5 percent
- 17-stage axial compressor; pressure ratio 10.8:1
- Air flow rate = 310 lb/sec
- Reverse flow, multichamber (can-annular) combustion system (12 chambers)
- Single-stage gas generator turbine
- Single-stage power turbine (4670 rpm rated speed) with variable nozzles
- Two baseplate configurations (gas turbine flange-to-flange unit and auxiliary system), integrated enclosure is with the baseplate
- Standard configuration: 49 by 10 by 12 ft (15 by 3.2 by 3.8 m); weight 110 tons
- Combustors: DLN combustion system available (DLN-1) or lean head end combustors can also be used

The Frame 5D is shown in Figure 12. The Frame 5D machine is an uprate of the Frame 5C, and utilizes a Frame 6B axial compressor. The gas generator rotor and single-stage gas generator turbine (the high pressure turbine) are shown in Figure 13. The machine has variable power turbine nozzles between the high pressure turbine and the power turbine. When the variable-stage nozzle is opened, the backpressure on the high pressure turbine is lowered, resulting in greater energy utilized across the high pressure turbine. The ability to modulate the enthalpy drop between the gas generator turbine and the power turbine of this engine results in good part load efficiency and a better control of speed with sudden load changes.

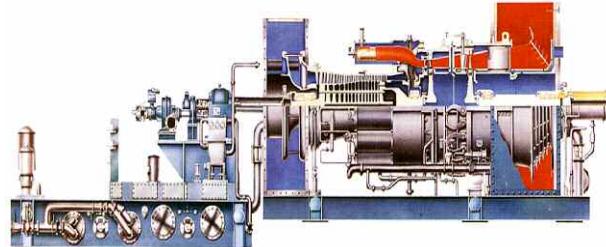


Figure 12. Layout of a Frame 5D Gas Turbine.



Figure 13. Frame 5D High Pressure Rotor (Axial Compressor and High Pressure Turbine).

LM2500+ (PGT25+) Aeroderivative Engine

This 31.3 MW ISO engine has been successfully implemented in an LNG facility in Australia, being the world's first application of a high efficiency aeroderivative engine in LNG service. The evolution of this engine is shown in Figure 14. The engine is derived from the LM2500 engine, by increasing its airflow and pressure ratio by 23 percent. The airflow increase was derived by adding a "zero-stage" transonic stage to the axial flow compressor. Salient design features of this engine include:

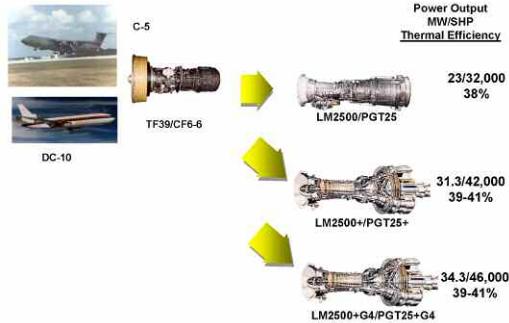


Figure 14. Evolution of the LM2500 Engine. (Courtesy GE Energy)

- 17-stage axial compressor; PR 23:1
- Annular combustion chamber (30 fuel nozzles)
- Two-stage gas generator turbine
- Two-stage high speed power turbine (6100 rpm)
- The gas generator, power turbine, and auxiliary system are mounted on a single baseplate
- The enclosure is integrated with the baseplate for easy maintenance
- Standard configuration: 21 by 12 by 13 ft (6.5 by 3.6 by 3.9 m); weight 38 tons
- Dry low emission (DLE) combustion system or steam or water injection systems for NOx abatement

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. As reported by Wadia, et al. (2002), the axial flow compressor operates at a pressure ratio of 23:1 and has a transonic blisk (integral blades and disk) as the zero-stage. The zero-stage operates at a stage pressure ratio of 1.43:1 and an inlet tip relative Mach number of 1.19. The LM2500+ airflow rate is 84.5 kg/sec at a gas generator speed of 9586 rpm. The axial compressor has a polytropic efficiency of 91 percent.

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 PGT25+ high pressure turbine (gas generator turbine or 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 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.

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 scavenging pump, the starter, and the variable-geometry control. An overview of the engine including the HSPT is shown in Figure 15.

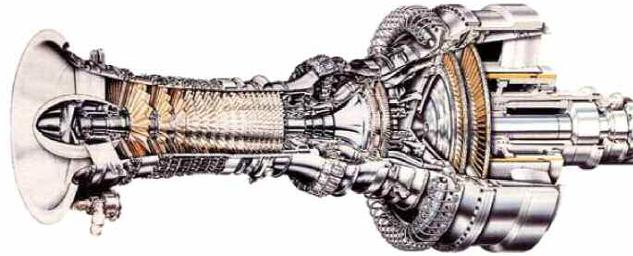


Figure 15 Overview of the PGT25+ Gas Turbine. (Courtesy GE Energy)

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 borescope 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 onsite, rapid exchange of major modules within the gas turbine. Onsite component removal and replacement can be accomplished in less than 100 man hours. The complete gas generator unit can be replaced and be back online within 48 hours. The hot-section repair interval for the aeroderivative is 25,000 hours on natural gas. However, water injection for NOx control shortens this interval to 16,000 hours to 20,000 hours depending on the NOx target level.

Upgrades of the PGT25+

An advantage of using aeroderivative engines for LNG service is that they can be upgraded to newer variants, generally within the same space constraints, and this is a useful feature for future debottlenecking. The LM2500+G4 is the newest member of the LM2500 family of aeroderivative engines. The engine retains the basic design of the LM2500+ 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+. The LM2500+ G4 engine is shown in Figure 16. Details of this variant are presented by Badeer (2005). The growth in power of this variant compared to the base engine is shown in Figure 17.

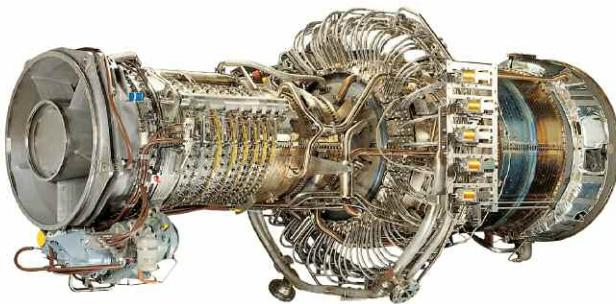


Figure 16. The Upgraded LM2500+ G4 Engine—DLE Variant. (Courtesy GE Energy)

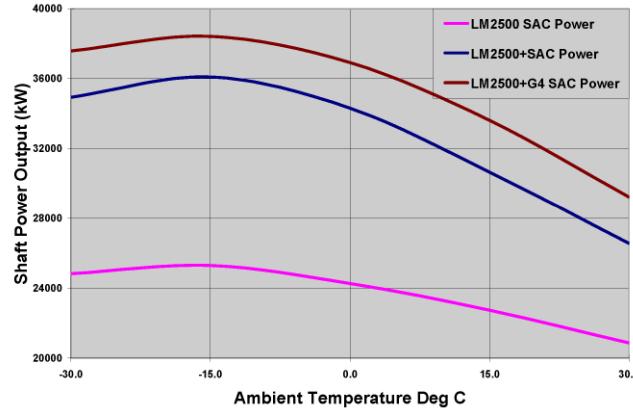


Figure 17. Growth of the LM2500+G4 Variant. (Courtesy GE Energy)

Frame 6B Single Shaft Gas Turbine

The Frame 6B is a power generation machine rated at approximately 44 MW and a thermal efficiency of 33 percent and has been widely used in mechanical drive service. The output speed of this machine is 5111 rpm. An overview of this machine is shown in Figure 18. Some of the salient design features include:

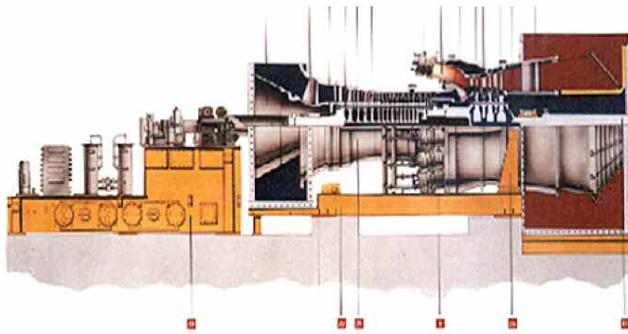


Figure 18. Overview of the Frame 6B Gas Turbine.

- Hot end drive
- Two bearing machine
- Axial compressor 17 stages (12:1 pressure ratio), air flow rate = 310 lb/sec
- Inlet guide vanes (IGVs)
- Bolted construction
- 10 can annular chambers
- 25 ppm NOx (15 percent O2) with gas fuel
- Three-stage turbine
- Bolted construction
- Advanced materials
- Enhanced cooling
- Standard or DLN1 combustors

Frame 7EA Gas Turbine

The Frame 7EA has been extensively used for mechanical drive service. It is rated at 86 MW and a thermal efficiency of 33 percent and operates at a pressure ratio of 12.5:1. The air mass flow rate is 296 kg/sec. The machine has an output speed of 3600 rpm. An overview of the machine is shown in Figure 19.

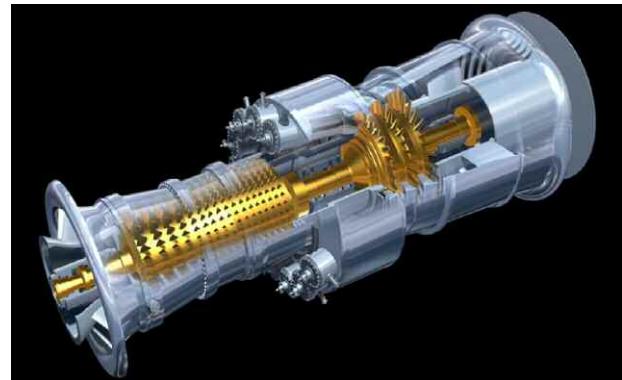


Figure 19. Overview of the Frame 7EA Gas Turbine.

- Hot end drive
- Three bearing machine
- Axial compressor 17 stages (12.6:1 pressure ratio), air flow rate = 653 lb/sec
- IGVs
- Bolted construction
- 14 combustors
- 25 ppm NOx (15 percent O2) with gas fuel
- Three-stage turbine
- Bolted construction
- Advanced materials
- Enhanced cooling
- Standard or DLN1 combustors
- Output speed 3600 rpm

Frame 9E Gas Turbine

The Frame 9E is a scaled up 50 Hz version of the Frame 7. It is rated at 129 MW and a thermal efficiency of 34.4 percent and operates at a pressure ratio of 12.7:1. The air mass flow rate is 413 kg/sec. The machine has an output speed of 3000 rpm, which is an advantage for the propane compressor service as a slower speed helps in dealing with Mach number limitations of the compressor as will be discussed in the compressor section ahead.

The first application of Frame 9E machines for LNG service are currently underway with string tests being completed. Details on this application may be found in Saulisbury, et al. (2007), including an extensive design review process to qualify this engine for mechanical drive service.

A view of the Frame 9E driver is provided in Figure 20. Some salient features of this driver are:

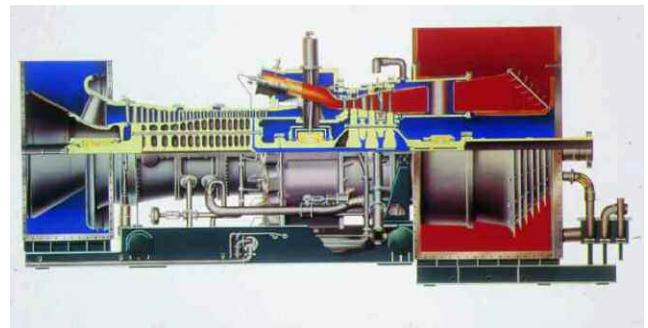


Figure 20. Overview of the Frame 9E Gas Turbine.

- 17-stage axial flow compressor (12.7:1 ratio)
- 16 though-bolt connection
- Variable IGV
- First seven stages, corrosion resistant material/coating
- Radial inlet casing
- Hot end drive—three bearings
- 3000 rpm speed
- 14 combustion chambers
- Combustion liner in Hastelloy® X + TBC
- Transition piece in Nimonic® 263 + TBC
- Standard combustor or DLN1 available
- Three-stage turbine, first two stages air-cooled

Effect of Site Conditions on GT

Performance with Respect to LNG Drivers

It is important to understand the effects of site conditions on gas turbine performance as they directly will impact the LNG production rate. Site conditions that have to be considered (Meher-Homji, et al., 2001) include:

- Effect of ambient temperature
- Effect of altitude (i.e., pressure)
- Inlet filter pressure losses
- Exhaust system pressure losses
- Ambient humidity effects
- Influence in changes in fuel heating value

Influence of Ambient Temperature

This is a factor of major importance for LNG driver applications as LNG production is linked to refrigeration driver power. Unless a gas turbine inlet cooling technique is used, there is little that can be done regarding the ambient temperature. Inlet cooling techniques might include traditional refrigeration chilling, inlet evaporative cooling by use media, or inlet fogging. Turbine output is a strong function of the ambient air temperature with power output dropping by 0.54 to 0.9 percent for every 1°C rise in ambient temperature (0.3 to 0.5 percent per 1°F). Gas turbines can experience power output drops of around 14 to 20 percent when ambient temperatures increase from 15°C (59°F) to 35°C (95°F). There is also a concurrent heat rate increase of about 5 percent.

Aeroderivative gas turbines exhibit even a greater sensitivity to ambient temperature conditions. A representation of the power boost capability for given inlet cooling potential for different types of gas turbines is shown in Figure 21. The drop in performance due to high ambient temperatures can be further aggravated with gas turbine recoverable and unrecoverable performance deterioration due to several factors as presented in Meher-Homji, et al. (2001).

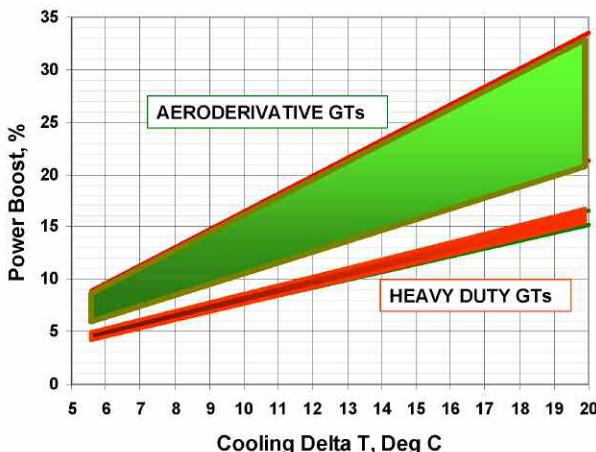


Figure 21. Power Boost Due to Inlet Cooling for Frame and Aeroderivative Engines.

An analysis and simulation of 91 gas turbines were conducted to evaluate the sensitivity to ambient temperature in terms of the net work ratio of the engines. The net work ratio is defined as the output of the gas turbine divided by the total turbine work (i.e., the output + axial compressor work). Results of these simulations are shown in Figure 22 (Chaker and Meher-Homji, 2007). This graph shows that units with lower net work ratios (such as the aeroderivatives) tend to have a greater sensitivity to ambient temperature.

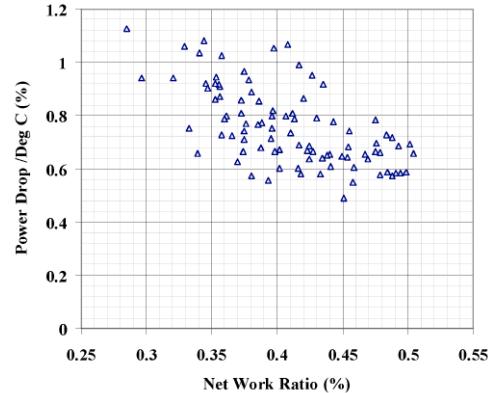


Figure 22. Power Drop in Percent/°C Versus Net Work Ratio for 91 Gas Turbines Simulated. (Courtesy Chaker and Meher-Homji, 2007)

The impact of inlet temperature on the salient parameters of a Frame 7EA in simple cycle operation is provided in Table 2. This table depicts the change in key parameters as inlet ambient temperatures are varied from 10 to 40°C (50 to 104°F). The table is based on the turbine operating at 100 percent rating, in an inferred turbine inlet temperature (TIT) control mode, control curve limit, inlet loss of 10 millibar (.145 psi) and exhaust losses of 12.45 millibar (.181 psi). The table shows the compressor work and turbine work, and also the compressor and turbine specific work variation. It is the relationship between the two specific works that results in the drop in output at higher temperatures. The situation is shown graphically in Figure 23.

Table 2. Operating Parameter of a Frame 7EA Gas Turbine Showing Effect of Ambient Temperature. (Chaker and Meher-Homji, 2007)

| T _{amb} °C | Output kW | M _a , kg/se | CDP Bar | PR | Heat Rat KJ/kWhr | W _c , kW | Sp. Comp Wk | Wt kW | Sp. Turb Wk |
|---------------------|-----------|------------------------|---------|-------|------------------|---------------------|-------------|---------|-------------|
| 10 | 88,041 | 298 | 12.9 | 12.9 | 10,790 | 107,344 | 360 | 197,426 | |
| 15 | 85,213 | 292 | 12.64 | 12.64 | 10,867 | 105,244 | 360 | 192,481 | |
| 20 | 82,433 | 286 | 12.38 | 12.38 | 10,945 | 102,949 | 360 | 187,385 | |
| 25 | 79,730 | 280 | 12.11 | 12.11 | 11,029 | 100,717 | 360 | 182,377 | |
| 30 | 76,954 | 273 | 11.84 | 11.84 | 11,130 | 98,428 | 360 | 177,277 | |
| 35 | 74,136 | 267 | 11.56 | 11.56 | 11,245 | 96,106 | 361 | 172,104 | |
| 40 | 71,314 | 260 | 11.29 | 11.29 | 11,374 | 93,757 | 361 | 166,896 | |

Frame 7EA Relative Compressor and Turbine Work, Referenced to Tamb = 10°C

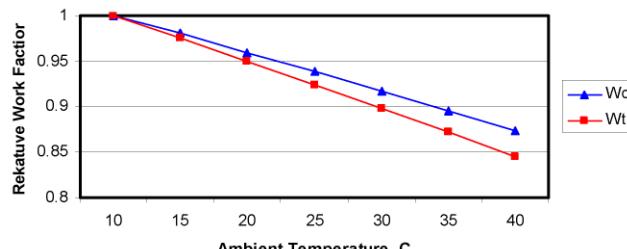


Figure 23. Change in Compressor and Turbine Work Referenced to Reference Value at Compressor Inlet Temperature (CIT) = 10°C (50°F). (Chaker and Meher-Homji, 2007)

Large single shaft gas turbines such as the Frame 7EA and Frame 9 exhibit a significant drop in power when the speed drops below the design speed (3600 rpm for the Frame 7EA and 3000 rpm for the Frame 9E). While this is of importance for power generation applications, the application of large single shaft machines for mechanical drive applications is even more important. This is because the reduced corrected speed N_{corr} (defined as N/\sqrt{T}) of the shaft reduces the volumetric flow through the machine. The effect is further compounded at high ambient temperatures where the denominator of the corrected speed term grows as T increases, thus further decreasing the corrected speed. Normally at constant speed, the volumetric flow through a gas turbine (at constant IGV settings) is fixed.

Influence of Altitude (i.e., Ambient Pressure)

High altitude reduces the air density and consequently the mass flow rate resulting in a drop in power with increasing altitude. As a rule of thumb, the power drop is approximately 3 to 4 percent per 1000 ft altitude. There is also a drop in mass flow rate similar in magnitude to the loss of power. Nothing can be done about this loss other than supercharging the gas turbine inlet with a blower, a solution that is not popular in practice. Most LNG plants however are located at sea level so altitude is not a significant factor.

Influence of Inlet Filter Pressure Losses

This is an important parameter as it is controllable to some extent and will affect turbine engine performance. The pressure drop across the inlet filter reduces the density to the inlet of the gas turbine thus reducing the mass flow rate. The lower pressure at the turbine inlet implies a lower expansion ratio across the turbine and a consequent elevation in exhaust gas temperature.

As a rule of thumb, a 1 inch water gauge inlet pressure loss will cause a 0.4 percent drop in power and a 0.12 percent increase in heat rate. Ambient temperature, relative humidity, and climatic conditions such as fog and smog will strongly impact compressor fouling. In some cases, turbines have been known to trip due to excessive backpressure caused due to morning fogs. In other cases, sandstorms have also been known to create high differential pressure conditions.

Influence of Exhaust System Pressure Losses

An increase in the backpressure due to excessive stack losses or the presence of silencers causes a backpressure on the turbine section, which decreases the expansion ratio across the turbine. This drop in expansion ratio results in an increased exhaust gas temperature (EGT). As a rule of thumb, a 1 inch water gauge exhaust pressure loss will cause a 0.15 percent drop in power and a 0.12 percent increase in heat rate.

Influence of Ambient Humidity

As a practical matter, the effect of ambient humidity is very small and is dependent on the control philosophy. For most performance deterioration evaluations, it can be neglected. A gas turbine operating at fixed firing temperature would produce more power with high humidity. This is because of the increased C_p value due to the presence of the water. However, the situation is not so straightforward due to control system complexities.

In the case of two shaft gas turbines, as a general rule, humid air is lighter than the dry air, so the compressor gets unloaded and will tend to speed up (assuming parity of other parameters). Further, the specific heat of wet air is higher than that of dry air, so the available specific enthalpy at the turbine inlet, at assuming the same inlet temperature, will be higher for wet air. The impacts of the control system and a corrected parameter control approach for several parameters including off-design operation in terms of power turbine speed has been presented by Casoni, et al. (2004).

Influence in Changes in Fuel Heating Value

Gas turbines operate on a wide range of gaseous and liquid fuels. A useful number to understand the effect of gaseous fuel is the Wobbe number defined as:

$$\text{Wobbe No.} = LHV / \sqrt{\rho G} \quad (1)$$

Some original equipment manufacturers (OEMs) define a temperature compensated (or modified) Wobbe number by including the fuel temperature in °R as a multiplier to the specific gravity (SG) term in the denominator. Fuel with significant heavy hydrocarbons present will have a higher Wobbe number while the presence of inerts such as nitrogen and carbon dioxide will lower the Wobbe number.

In general, the use of low Btu fuel (i.e., low lower heating value [LHV]) will cause an increase in power because of the extra mass flow through the turbine. A secondary effect is that the increased turbine mass flow causes an increase in pressure ratio. Changes in heating value become significant in gas turbines that operate on multiple fuels in that the effects on performance have to be distinguished from deterioration. A treatment of performance impact due to variations in fuel heating value is made by Cloyd and Harris (1995). With DLN combustors, the rate of change of Wobbe number must be carefully evaluated when switching fuels.

Gas Turbine Performance Deterioration

In sizing and evaluating any LNG driver, it is important to consider and understand the issue of gas turbine performance deterioration. Typically, units are sized with a certain percent (e.g., 6 or 7 percent) reserved for deterioration. The causes of gas turbine deterioration fall essentially into two categories: performance degradation and mechanical degradation with performance degradation being the more significant.

A further classification of performance deterioration into recoverable and unrecoverable deterioration can also be made:

- *Recoverable deterioration*—Can be removed by actions during operation of the gas turbine.
- *Unrecoverable deterioration*—Can be recovered by an overhaul but not during operation.
- *Permanent deterioration*—Residual deterioration present even after a major overhaul.

A detailed treatment of performance degradation is made by Kurz and Brun (2000) and Meher-Homji, et al. (2001). Performance retention action has to be taken in a scientific manner and often has to be tailored to the specific gas turbine under consideration. There are always some projects where a pronouncement is made that efficiency is not of importance because fuel is “free.” This is a mistaken notion because there is always an opportunity cost associated with excessive fuel usage, for example, under certain conditions of supply limitation, minimizing the fuel usage can result in increased LNG product.

Turbine and Compressor Losses

Efficiency is one of the most important parameters for gas turbines, fundamentally because the net output is the difference between turbine output and the work consumed in the compressor. With the ratio of compressor work to total turbine work being approximately 0.5, clearly component efficiencies are of critical importance and will create a large change in gas turbine output. The deterioration of efficiency in the compressor and turbine is of critical importance from a performance retention standpoint.

With advances in gas turbine technology, it is common to find compressor and turbine efficiencies at 90 percent and above. An excellent review of turbomachinery losses is made by Denton (1993).

The three forms of losses can be conveniently classified as:

- *Profile losses*—Profile losses are the losses generated in the blade boundary layer well away from the end walls.
- *Endwall losses*—Also referred to as “secondary losses” these losses arise partly from the secondary flows generated when the annulus boundary layers pass through the blade row.
- *Tip leakage losses*—These arise from the leakage of the flow over the tips of rotor blades and the hub clearance of the stator blades. The losses depend on whether the blades are shrouded or unshrouded.

While these are often considered as discrete losses, the loss mechanisms are really interrelated.

Dirt Deposits and Fouling

Deposits form mainly on the compressor blading causing loss of flow capacity and efficiency and this mechanism is a predominant form of recoverable deterioration. Fouling deterioration is described by Meher-Homji (1990). The turbine section can also foul. The effects of fouling are a drop in output and a worsening of heat rate. There are several peripheral problems that may be created such as cooling hole blockage, imbalance, blade root lockup, and hot corrosion. Because this is a topic of major importance, it will be treated in a dedicated section ahead.

A simulation to indicate the effect of compressor fouling using consumer available proprietary software has been made considering a 39.6 MW ISO Frame 6B in simple cycle configuration. Natural gas (21,518 Btu/lb) has been considered and typical inlet and outlet pressure drops (4 and 5 inch WG) for simple cycle were considered. The machine has an ISO pressure ratio of 11.8:1 and a mass flow rate of 304 lb/sec. The firing temperature is 2020°F.

The simulation was run at an ambient temperature of 59°F, and imposing deterioration in the following sequence:

- Step 1: New and clean, mass flow drop = 0 percent, compressor efficiency drop = 0 percent
- Step 2: Mass flow drop = 1 percent, compressor efficiency drop = 0.833 percent
- Step 3: Mass flow drop = 2 percent, compressor efficiency drop = 1.67 percent
- Step 4: Mass flow drop = 3 percent, compressor efficiency drop = 2.5 percent
- Step 5: Mass flow drop = 4 percent, compressor efficiency drop = 3.33 percent
- Step 6: Mass flow drop = 5 percent, compressor efficiency drop = 4.167 percent
- Step 7: Mass flow drop = 6 percent, compressor efficiency drop = 5 percent

The output and heat rate variation with the deterioration steps is shown in Figure 24. The output at the end of the seventh deterioration step has dropped 5.5 MW while the heat rate has increased by 850 Btu/kW hr. The change in mass flow rate, compressor discharge pressure, and compressor discharge temperature corresponding to the simulated deterioration steps is shown in Figure 25. The drop in efficiency causes the compressor discharge temperature to increase by approximately 19°F and the compressor discharge pressure to drop by 10 psia.

The type of foulants entering the compressor varies widely from site to site. Deposits of oil and grease are commonly found in industrial locations as a result of local emissions from refineries and petrochemical plants. These types of deposits act as “glue” and entrap other materials entering the compressor. Coastal locations

usually involve the ingestion of sea salt, desert regions attract dry sand and dust particles, and a variety of fertilizer chemicals may be ingested in agricultural areas.

Compressor foulants are often classified as being “oil soluble,” “water soluble,” or “water wettable,” but experience has shown that they typically are a combination of these types. For example, although sea salt is essentially water soluble, its retention within the compressor may be significantly influenced by trace quantities of oil and grease. In this case, the use of water alone for washing may not be sufficient, and a chemical detergent would be required for effective compressor cleaning. The susceptibility of a gas turbine axial flow compressor to fouling is controlled by the following major factors (Stalder, 1998):

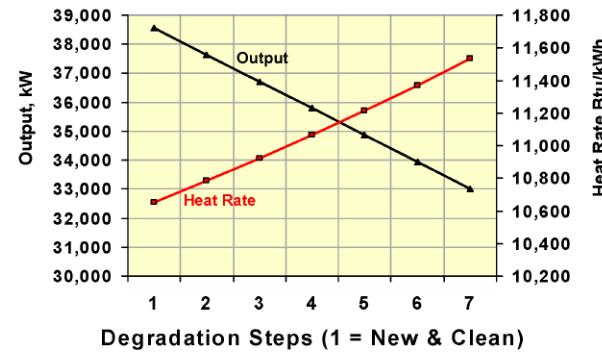


Figure 24. Output and Heat Rate Deterioration with Successive Levels of Compressor Fouling.

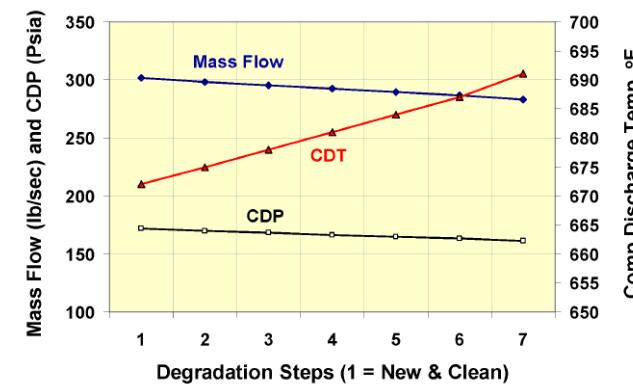


Figure 25. Changes in Mass Flow, Compressor Discharge Pressure, and Temperature with Successive Levels of Compressor Fouling.

- Gas turbine design parameters
- Location of the plant and the environment
- Plant design and layout
- Atmospheric parameters
- Plant maintenance practices

The rate of compressor fouling deterioration and foulant type are strongly affected by predominant wind directions. Factors that must be taken into account include:

- Orientation and elevation of air inlet ducts.
- Location of water cooling towers or air-to-air heat exchangers.
- Possibility of gas turbine exhaust recirculation into the air inlet.
- Orientation of lube oil breathers and vapor extractors.
- Location of highways and adjacent plants that can contribute to the fouling problem.

CFD predictions can be used to understand the phenomenon especially in grass roots installations. Other important plant design

parameters that affect the rate of compressor fouling are the selection of appropriate air inlet filtration systems (self-cleaning, depth loading, etc.), the selection of filter media, the number of filtration stages, weather louvers, inertial separators, and mist eliminators. Design parameters such as the air face velocity through the filters, filter loading, and their behavior under high humidity and high pressure drops are also critical.

Erosion

Erosion changes airfoil shape, contours, and surface finish. This typically does not pose a major problem to land-based turbines due to the air filtration system present.

Increases in Clearances in Vanes, Blades, and Seals, I.E., Wear

Wear is a significant problem especially during the early stages of engine operation. It can be caused by thermal growth problems and centrifugal growth. Tip clearance increase of blading is a serious cause of deterioration as it reduces the stage efficiency. Tip stalls can also be caused by combinations of blade tip erosion and consequent loss of clearance. Increase in clearances is a form of unrecoverable deterioration.

Hot Section Problems

This can include detached liners, cracks, or unbalanced nozzles and is a form of unrecoverable deterioration. Problems of this nature may, in most instances, be detected by means of evaluation of exhaust gas temperature (or interturbine temperature) spreads. Problems such as hot section corrosion, erosion, and fouling also exist and may be recoverable during a refurbishment.

Other Sources of Deterioration

In addition to the items mentioned above, other sources of deterioration include:

- Excessive increase in inlet filter differential pressure.
- Excessive backpressure—this can be of importance when heat recovery steam generators (HRSGs) are present.
- Increased mechanical losses (gearboxes, bearings, couplings, etc.).
- Internal losses.
- Stator nozzle plugging (or turbine fouling).
- Overboard leakage.
- Change in high pressure flow function (mv T/P).

Details on gas turbine degradation and detection methods are presented by Saranavanamuttoo and Lakshminarasimha (1985), and Diakunchak (1991, 1993). Modeling details of gas turbine performance deterioration may be found in Lakshminarasimha, et al., 1994.

Nonrecoverable Deterioration

Unrecoverable performance deterioration is the residual deterioration that will exist and detract from gas turbine performance even after the gas turbine has been washed and cleaned. Damage that occurs to the flow path due to erosion, corrosion, and increased tip and seal clearances or due to distortion of the casing itself will contribute to this.

During a major overhaul, deteriorated gas turbine components are cleaned or replaced, and airfoils are recoated (compressor and turbine section as needed) and tip and seal clearances restored as close as possible to the initial condition. Even after the overhaul, however, there will be some performance loss due to:

- Casing distortion.
- Increased flow path surface roughness.
- Airfoil untwist.
- Increased leakage areas.

While this holds true for heavy duty gas turbines, aeroderivative engines can be returned to a “zero hour” condition typically after a major overhaul at 50,000 hours.

Mechanical Degradation

Causes of mechanical degradation include wear in bearings and seals, coupling problems, excessive vibration and noise, or problems in the lube oil system. Probably the most common indicator of mechanical degradation has been vibration. It is most important to note that several problems that manifest themselves as vibration may in fact have underlying causes that are aerodynamic (or performance) related in nature.

Bearing problems are often caused by low oil pressure (malfunction in pump or leaks), line blockage, and excessive loads due to factors such as misalignment. The lube oil supply pressure and scavenge temperatures can be measured and correlated to a parameter such as rotor speed. The expected pattern during speed changes can be noted and subsequent checks made during transients.

Combustor fuel nozzles can at times plug up. There can be several causes for this such as coking, erosion, and misassembly. Temperature distortions can create a host of problems in the hot section. Severe temperature distortions can create serious dynamic loads on blading, possibly inducing fatigue problems. The pattern of the exhaust gas temperature (EGT) spreads can be monitored during transient conditions to indicate nozzle problems. Blade failures that can be induced by performance and mechanical factors have been detailed by Meher-Homji and Gabriles (1998).

Gas Turbine Operating Environment, Filtration, and Compressor Washing

As air passes through the intake and filtration system, it proceeds at a very low velocity with filter face velocities being typically around 3 m/sec (9.84 ft/sec). As it approaches the compressor face, the air accelerates to a high velocity (0.5 to 0.8 Mach number). This results in a significant static temperature reduction. The saturation air temperature also drops. If the relative humidity is high enough, it is possible that the static air temperature falls below the saturation air temperature. This causes condensation of water vapor, which is a common occurrence in most gas turbines when ambient relative humidity is high.

Filters tend to unload salt (leaching effect) under high ambient humidity conditions and this is a factor that is often neglected. It is this factor that causes the sudden fouling of compressors during periods of ambient fog. Particles then form nuclei for the water droplets and start to adhere to the blading. As the air progresses to the rear compressor stages, it gets hotter and drier, and typically causes less fouling in the rear stages.

Stalder and Sire (2001) have conducted detailed analytical and experimental work on salt percolation through gas turbine air filters and some of the salient observations and results are presented below.

- The ingestion of airborne sea salt through a filter can result in high temperature corrosion damage. With the high temperature engines available today, this is of significant importance as turbine manufacturers limit the total amount of salt entering the engine to low values such as 0.2 to 0.5 ppmw. Total contaminants that enter the engine must be calculated based on airflow, water injection flow, and fuel flow. The total contaminant entering the engine is given by:

$$TCL = I_f + [I_{air} \times A/F] + [I_w \times W/F] + [I_S \times S/F] \quad (2)$$

where,

| | |
|-----------|--|
| TCL | = Total contaminant level, ppmw |
| I_f | = Contaminant level in the fuel, ppmw |
| I_{air} | = Contaminant level in the air, ppmw |
| I_w | = Contaminant level in the injection water, ppmw |

| | |
|------------------|--|
| I _{stm} | = Contaminant level in the injection steam, ppmw |
| A/F | = Air to fuel ratio for the gas turbine |
| S/F | = Steam to fuel ratio |
| W/F | = Water to fuel ratio |

- With regard to limits of airborne contaminants, the following guidelines are provided: Maximum allowable level of NaCl in treated air is 0.01 ppmw (which is equivalent to 0.03 ppmw Na in the fuel assuming an air to fuel ratio of 75). Some specifications set an airborne limit for Na + K + V + Pb not to exceed 0.005 ppmw. Specific OEM requirements should be considered, as this is dependant on blade metallurgy, coating technology, and cooling approach. The performance of filters in limiting ingress of salt is therefore extremely important.
- Studies performed on used filters indicate that percolation of salt laden water can occur in both surface and depth loading types of filters. The mechanism is supported by the presence of dissolved acid gases (SO_x and NO_x) found in industrial environments.
- The large area of the air filter media may also act as a reaction site for various chemical processes and reactions. The real challenge is to determine the efficiency of the media under real life operational conditions where aggressive gases may exist in the environment.
- Optimal selection of filter media is of supreme importance and the system must be tailored to climatic conditions.
- Good filtration must be coupled with a carefully planned online and offline washing regime to minimize salt deposits on compressor airfoils.

Further details relating to compressor fouling and washing may be found in Meher-Homji and Bromley (2004).

Gas Turbine Design Reviews

As the size of the LNG market grows, the trend toward larger gas turbines implies that new gas turbines need to be considered for compressor driver duty. While the LNG industry has conventionally been very conservative, the high investment costs and drive toward more efficient plants have induced the need to consider newer gas turbines. For example aeroderivative engines and large Frame 9Es have recently been introduced. There are other potential engines on the market that could also be considered. Typically, potential users, process licensors, and engineer, procure, and construct groups (EPCs) will conduct detailed design reviews of engines to evaluate design, operations, and maintenance issues. While most vendors consider much of this information proprietary, they are usually willing to share it during meetings after the signing of appropriate confidentiality agreements. Issues that should be investigated include:

Aeromechanical Design Review

- Obtain detailed design information—scaling used for design, past vintages used for new engine, etc.
- Design modifications/package modifications for the proposed engine.
- Review of engine test data.
- Startup torque capabilities—torque versus power turbine speed curves.
- Mechanical and aero design review.
- Blade aeromechanical design including blade Campbell diagrams, stress levels, etc.
- Reliability availability and maintainability (RAM) analyses conducted during design—RAM factors.
- Ability to water wash online.
- Inlet distortion limits.
- Thrust balance issues.
- History of DLN/DLE combustor stability problems, design solutions and results, stability and operability. DLN/DLE combustor tuning requirements. Cold weather considerations for DLN/DLE if appropriate.

- Control philosophy of engine for mechanical drive.
- Rotordynamics and vibration at variable speed operation—presence of modes in speed range and the severity, unbalance sensitivity, sensitivity to internal alignment, sensor locations, 1× rev sensitivities, vendor quality assurance (QA) acceptance levels, vibration limits, etc.
- Balancing criteria and sensitivity at different criticals including regions of unbalance sensitivity.
- NO_x levels and other emission levels.
- Drop load capability and step load change capacity especially for DLN/DLE (lean extinction).
- Part load operation for DLN/DLE.
- Discussion of power augmentation approaches.
- Fuel issues, variations, and control for both DLE and non DLE.
- Water wash requirements, procedures—on and offline.
- Bearings.
- Maintainability issues/borescope ports, etc.
- Installation and design checklists criteria, etc.
- Condition monitoring equipment available.
- Lube oil debris monitoring package in the case of roller/ball bearings.
- Review of capability to startup compressor strings.
- Ability to handle rapid drop load—compressor in surge or full recycle.
- Inlet system design review.
- Specs for air/water/fuel.
- Component materials and coatings used. (Options available if any.)
- Ongoing technical programs on the mechanical drive/power generation.
- Obtain list of service bulletins/service letters, etc.
- Detailed review of all auxiliary systems.

Operability and O&M Review

- Discussion with end users of the engine or similar engines.
- Operational problems and solutions.
- Experience with trips (forced outages) and their duration.
- Experience with recoverable and nonrecoverable deterioration.
- Have any parts had to be replaced prematurely—shrouds, liners, etc.
- Issues with gearbox, scavenge pumps, and other auxiliaries.
- Issues with hydraulic starters if appropriate.
- Stall or surge issues.
- Variable geometry schedule problems.
- Ongoing issues with DLN/DLE combustors.
- Catastrophic failure history.
- Tip rubs on hot shutdowns, etc.
- Details on operating hours, maintenance history.
- Spare engine availability, rotatable section availability in the case of aeroderivative engines.
- Maintenance philosophy—maintenance intervals with and without contractual service agreements (CSA).
- Long-term service agreements (LTSA)/(CSA) discussions.
- Historical maintenance costs.
- Spare parts philosophy.
- Vendor's repair facilities availability—and proximity to the proposed application.
- Availability of remote monitoring and diagnostic capability.

LNG String Startup with Single and Split Shaft Gas Turbines

With two shaft or multispool gas turbines, starting an LNG compressor string even at settle out pressure (SOP) is not an issue. A two shaft turbine has very high torque capability at low speed. With a single shaft gas turbine, however, starting a large compressor string presents a significant problem calling for the application of large starter motors and depressurization of the string in order to

limit the size of the starter motor. Details on startup and motor sizing as applied to single shaft gas turbines is presented in Heckel and Davis (1998).

In the normal starting sequence of a gas turbine after the accessories systems are activated, the gas turbine is brought to crank speed for three to six minutes while unfired to allow fresh air to purge through the inlet and exhaust systems. The purge time depends on the equipment configuration and local operating practices and has the purpose of removing any possibly explosive gas-air mixtures from the system. The duration of this purge can be longer for units with HRSGs. Then the gas turbine is allowed to coast down to optimal firing speed, the starting device (typically a motor and torque converter) is reactivated, the igniters are activated, fuel is admitted to the combustion chambers, flame is confirmed, and a one-minute warm-up period is observed. The fuel flow is then increased on a programmed scheduled basis while the gas turbine and train accelerate to 92 percent of rated speed. At that point the compressor bleed valves are opened, the inlet guide vanes are opened to the normal operating range, and the gas turbine is ready to accept load at a predefined ramp rate. When a single-shaft generator-drive gas turbine goes through the startup cycle, it has the benefit of merely driving an unloaded generator. The generator main circuit breaker is open and the friction, windage, and inertia of the two-pole generator rotor are relatively low. Therefore, a modestly-sized starting motor (or other starting means) can bring the gas turbine to the self-sustaining speed of about 60 percent, and then a very slight net available torque from the gas turbine takes over and continues to accelerate the set to the empowerment point of about 92 percent speed. The net available torque always exceeds the load torque imposed by the generator (Ekstrom and Garrison, 1994).

With a single-shaft gas-turbine driving an LNG compressor string, there is the need to significantly unload the driven equipment during the startup process or provide much greater starting assistance in order to get to the 92 percent (speed) where the gas turbine can contribute significant power. Consequently the starting system has to accommodate system inertia, gas turbine windage loads, and the refrigeration compressor aerodynamic load that is a function of the compressor speed, gas density, and process conditions. Typically a single shaft frame machine will have a large variable frequency drive (VFD) motor that could range from 6 MW to 30 MW that would accelerate the string.

DLN/DLE Considerations for Mechanical Drive Gas Turbines

Several of the gas turbines in LNG service currently are equipped with DLN combustors. Water injection may have to be used under situations where fuels exist that cannot be accommodated by dry technologies, but, in general, the trend is toward the application of dry combustors. Currently the technology applied for Frame 5, Frame 6B, and Frame 7E and 9E units in mechanical drive service are DLN-1 combustors capable of deriving 25 ppmv NOx.

An excellent overview of combustor technology is provided by Correa (1998), Davis and Black (2000), and Pavri and Moore (2001). The mechanism on thermal NOx production was first postulated by Yakov B. Zeldovich and is indicated in Figure 26, which shows the flame temperature as a function of equivalence ratio. The equivalence ratio is a measure of fuel-to-air ratio in the combustor normalized by stoichiometric fuel-to-air ratio. In other words:

$$\text{Equivalence Ratio} = (\text{FAR}) / (\text{FAR}_{\text{stoichiometric}}) \quad (3)$$

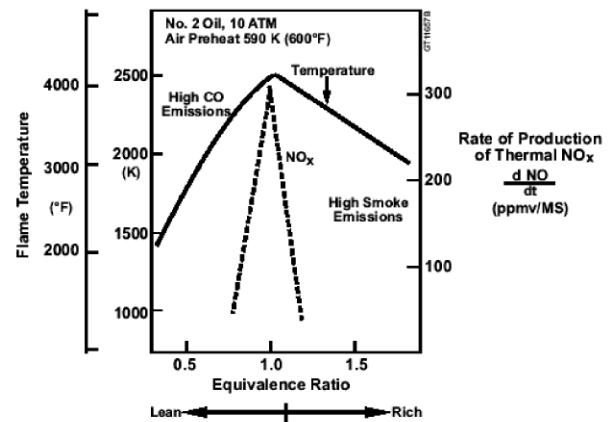


Figure 26. NOx Production and Flame Temperature Versus Equivalence Ratio.

At an equivalence ratio of one, the stoichiometric flame conditions are reached. The flame temperature is highest at this point and NOx levels peak. At equivalence ratios less than 1, the combustor operates in a lean zone. At the values greater than 1, the combustor is fuel "rich." All gas turbine combustors are designed to operate in the lean region. As can be seen in the figure, thermal NOx production rises very rapidly as the stoichiometric flame temperature is reached. Away from this point, thermal NOx production decreases rapidly. This theory then provides the mechanism of thermal NOx control.

For some gas turbine drivers, lean head end (LHE) combustors can be used to reduce NOx levels by 15 to 30 percent. An example of this is the LHE combustor for the Frame 5 gas turbine shown in Figure 27. The lean head combustor has additional holes allowing more air to be introduced at the head end thus leaning out the flame zone and shortening flame length (thus reducing residence time).



Figure 27. Lean Head End Combustor.

The DLN-1 two-staged premix combustor commonly used on single shaft LNG drivers is shown in Figure 28. The combustion system includes four major components: fuel injection system, liner, venturi, and cap/centerbody assembly. The combustion system operates in four distinct modes, illustrated in Figure 29. The six primary nozzles are shown in Figure 30 and the secondary nozzle in Figure 31. Collectively, these components form two stages in the combustor. In the premixed mode, the first stage

thoroughly premixes the fuel and air and delivers a uniform, lean, unburned fuel-air mixture to the second stage. Uniform mixing is important to avoid any hot spots. The operating modes are as follows:

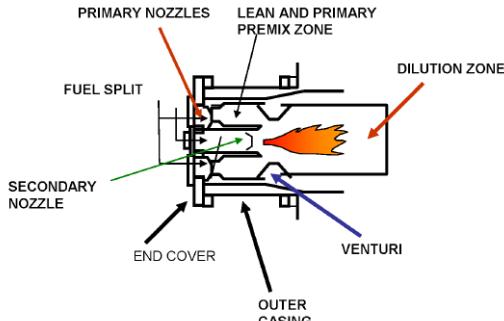


Figure 28. Layout of DLN-1 Combustor.

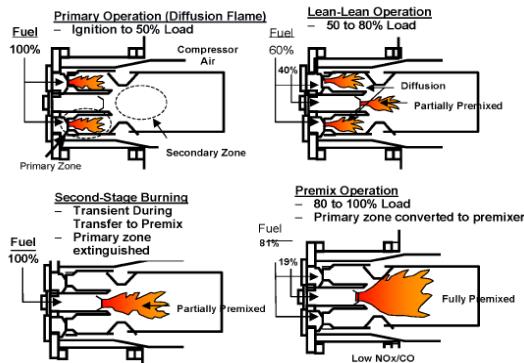


Figure 29. Modes of Operation of DLN-1 System.



Figure 30. Primary Fuel Nozzles.



Figure 31. Secondary Fuel Nozzles.

• **Primary**—In this mode, fuel is supplied to the primary nozzles only and a diffusion flame exists in the primary stage only. This mode of operation is used to ignite, accelerate, and operate the machine over low- to mid-loads, up to a preselected combustion reference temperature.

- **Lean-Lean**—In this mode, fuel is supplied to both the primary and secondary nozzles and a flame will exist in both the primary and secondary stages. This mode of operation is used for intermediate loads between two preselected combustion reference temperatures (the specific load percent depends on if inlet bleed heating is provided or not).

- **Secondary**—In this transient mode, fuel is supplied to the secondary nozzle only resulting in a flame in the secondary zone only. This mode is a transition state between lean-lean and premix modes and this mode is needed to extinguish the flame in the primary zone, before fuel is reintroduced into what becomes the primary premixing zone.

- **Premix**—In this final mode, fuel is supplied to both primary and secondary nozzles, but the flame is in the secondary stage only and the primary section is used for premixing only. This mode of operation is achieved at and near the combustion reference temperature design point and results in low NO_x. In this final mode over 80 percent of the fuel goes into the premix primary region and the remaining goes into the secondary.

A different approach is used in the annular combustors of an aeroderivative engine such as the LM2500 or the LM6000 where annular combustors exist. The LM gas turbines also use lean premixed combustion with fuel staging to maintain the narrow flame temperature window. A representation of the combustor showing a few nozzles is shown in Figure 32.



Figure 32. DLE Combustor-LM Aeroderivative Annular Combustor.

Gas Turbine Power Augmentation

LNG production is highly dependent on the power capability of the gas turbine drivers. Industrial gas turbines lose approximately 0.7 percent 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 percent per °C. The impact of ambient temperature on the PGT25+ power and air flow is depicted in Figure 33.

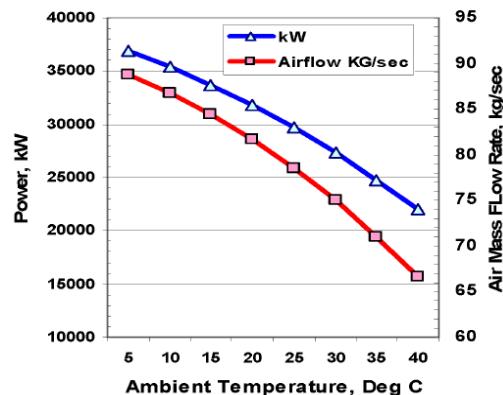


Figure 33. Variations in Power Output and Air Flow Rate for a PGT25+ Gas Turbine. (Courtesy Meher-Homji, et al., 2007)

Some of the key advantages of power augmentation are that it boosts LNG production, improves the thermal efficiency of the gas turbine, and results in lower CO₂ emissions. Power augmentation can be by evaporative cooling or inlet chilling.

An example of the application of evaporative cooling applied to an LNG plant (Meher-Homji, et al., 2007) is presented here. There is considerable evaporative cooling potential available 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 34. The relationship of relative humidity and dry bulb temperature is shown in Figure 35. Details regarding evaporative cooling potential may be found in Chaker and Meher-Homji (2006 and 2007). An overview of evaporative media cooling is provided in Johnson (1988) and McNeilly (2000).

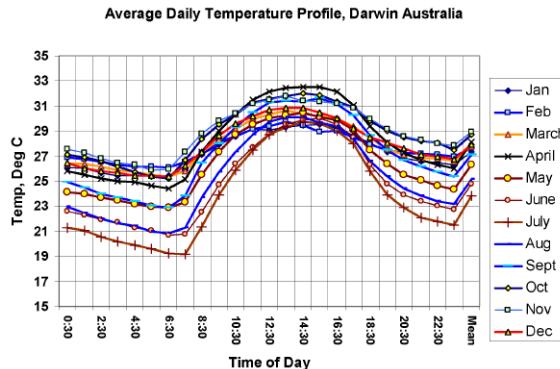


Figure 34. Temperature Profile Over Time of Day for 12 Months in Darwin. (Courtesy Meher-Homji, et al., 2007)

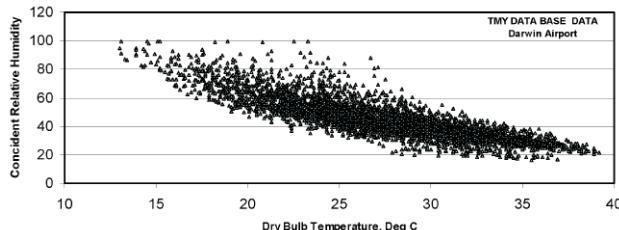


Figure 35. Relative Humidity Versus Dry Bulb Temperature, Darwin. (Courtesy Meher-Homji, et al., 2007)

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 nonevaporated 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 percent to 90 percent, and is defined as follows:

$$\text{Effectiveness} = (T_{1DB} - T_{2DB}) / (T_{1DB} - T_{2WB}) \quad (4)$$

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 carryover of large water droplets.

The presence of a media type evaporative cooler inherently creates a pressure drop that reduces turbine output. For most gas turbines, media thickness of 12 inches will result in a pressure drop of approximately 0.5 to 1 inch 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. Details on media evaporative coolers may be found in Johnson (1988).

Inlet temperature drops of around 10°C (18°F) have been achieved at a Darwin, Australia, LNG plant, which results in a power boost of around 8 to 10 percent. A graph showing calculated compressor inlet temperatures (CITs) with the evaporative cooler for a typical summer month of January is shown in Figure 36.

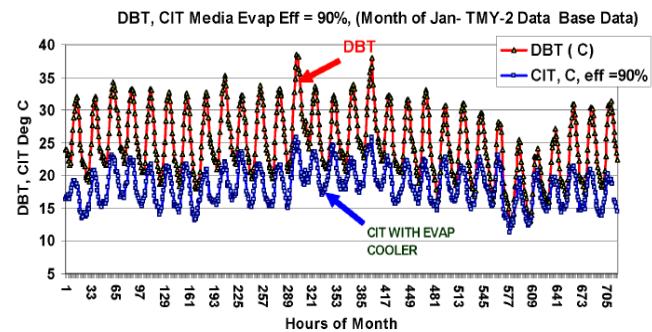


Figure 36. Calculated CIT Due to Evaporative Cooling over a Summer Month. (Courtesy Meher-Homji, et al., 2007)

Future LNG Drivers

There are several interesting potential future gas turbine engines that could be utilized for LNG drives. These include the LM6000 and the LMS100 engine.

The LM6000 depicted in Figure 37 is a 44 MW driver, operating at a pressure ratio of 29:1, with an exhaust mass flow rate of 128 kg/sec. This engine is a two-spool gas turbine with the load driven by the low speed spool. The low speed spool is mounted inside the high speed spool enabling the two spools to turn at different speeds. The output speed of this machine is 3600 rpm.

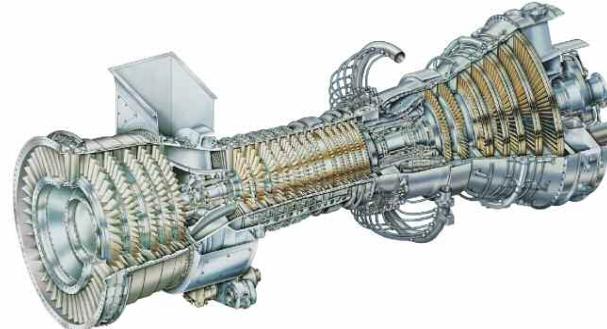


Figure 37. The LM6000 Gas Turbine, Rated at 44 MW, and a Thermal Efficiency of 42 Percent.

The LM6000 gas turbine makes extensive use of variable geometry to achieve a large operating envelope. The variable geometry includes the variable inlet guide vanes, variable bypass valves, and the variable stator vanes in the engine compressor with each system independently controlled. The gas turbine consists of five major components—a five-stage low pressure

compressor, a 14-stage high pressure compressor, an annular combustor, a two-stage high pressure turbine, and a five-stage low pressure turbine. The low pressure turbine drives the low pressure compressor and the load. The engine is available in both water injected and DLE configurations. Details may be found in Montgomery (2003). The engine is qualified as a variable speed mechanical drive unit (50 to 105 percent speed) and has been extensively tested both for variable speed operation and for its startup torque capability.

The LMS 100 (Reale, 2003) is a relatively new engine but has future promise as a large LNG driver with a nominal ISO power output of 100 MW and a thermal efficiency of 44 percent. A layout of the gas turbine is shown in Figure 38. The compressor is an intercooled design, operating at a pressure ratio of 40:1 and an exhaust flow rate of 460 lb/sec. As mentioned, at this time this engine has just recently been introduced into service but has potential for future LNG projects.

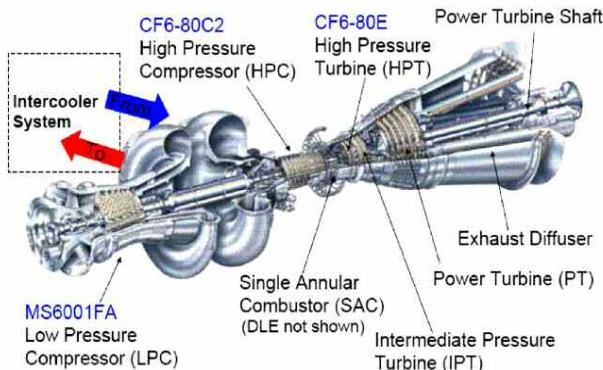


Figure 38. The LMS-100 Gas Turbine, Rated at 100 MW, and a Thermal Efficiency of 44 Percent.

LNG COMPRESSOR TECHNOLOGY

Compressors Used in LNG Service

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. Depending on the process technology used, the refrigerants may be pure components such as propane, ethylene, and methane, or mixed refrigerants, or some combination of these. The propane compressor is the most challenging machine in terms of flow coefficient and inlet relative Mach number. The design complexities, risks, and compromises involved in the selection and design of refrigeration compressors will be covered in this section.

Because of the dynamic nature of the LNG market, with several process design variants being studied and optimized, 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.

Compressor Configurations and Compressor Technology Milestones

Compression of large flows of gas can be accomplished by the use of either axial compressors, which are more suited to lower pressure ratios, or centrifugal compressors that enable higher pressure ratios. Some LNG processes use both axial and centrifugal compressors while others utilize only centrifugal compressors. A multistage axial flow compressor is shown in Figure 39 and a typical horizontally split centrifugal compressor is shown in Figure 40.

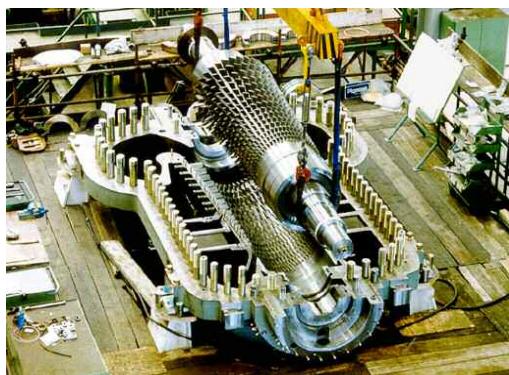


Figure 39. Axial Flow Compressor Used in LNG Service.



Figure 40. Horizontally Split LNG Centrifugal Compressor.

Compressor Technology Milestones

LNG compressors can be extremely large, with centrifugal impeller diameters in the range of 1300 to 2000 mm (51 to 79 inches). A large 1750 mm (5.7 ft) diameter impeller is shown in Figure 41. These 3D shrouded impellers can be produced on five-axis milling machines from a single piece forging as shown in Figure 42. Internal compressor casing diameters can be up to 4000 mm (13.1 ft) for horizontally split compressors and 2600 mm (8.5 ft) for barrel designs. A large horizontally split casing is shown in Figure 43. Inlet flows for centrifugal compressors can reach 300,000 m³/h. Shaft length can be 7400 mm (24.3 ft) with a weight of 7.5 tons, and bearing spans can be on the order of 5.5 m (16.4 ft), which require a shaft of suitable diameter at the bearings and end seals. Dry gas seals can have internal diameters of 350 mm (1.14 ft) and journal bearing sizes can be as high as 320 mm (1.1 ft). A representation of the current technology limits for large LNG compressors is provided in Table 3.



Figure 41. Large 1750 mm (5.7 Ft) Diameter 3D Centrifugal Impeller.



Figure 42. Five Axis Milling of Centrifugal Impeller.



Figure 43. Large Horizontally Split Casing.

Table 3. Current State-of-the-Art Technology Boundary for LNG Compressors.

| CASING | |
|----------------------------|-------------------|
| Weight of Barrel, tons | 120 tons |
| Inner Casing Diameter, mm | 3600 mm |
| SHAFT | |
| Length, mm | More than 7000mm |
| Bearing Span | More than 5800 mm |
| Amplification Factor | Greater than 5 |
| Log Dec | Greater than 0.1 |
| Dry Gas Seal Dia | 350 mm |
| Journal Bearing Dia | 320 mm |
| IMPELLER | |
| Inlet Relative Mach Number | 1.05 model test |
| Inlet Relative Mach Number | 0.94 full scale |
| Machine Mach Number | 1.24 Model test |
| Machiner Mach Number | 1.20 full scale |
| Head Coefficient | Higher than 0.54 |
| Head Per Impeller | More than 4000 m |
| Impeller Diameter | Upto 2000 mm |
| COUPLING | |
| Size | 350m |
| Power Transmitted | 145 MW |

Most of the LNG compressor casings are low pressure horizontally split designs. Per API 617 (2002), when a relief valve is not specified, the maximum allowable working pressure (MAWP) of the casing is 1.25 times the maximum discharge pressure. Sometimes, off-design process conditions can exceed this 1.25 times maximum discharge pressure. This can potentially cause problems with large horizontally split compressor casings and may exceed the supplier's casing joint bolting experience. These off-design process conditions should be reviewed with the compressor designer. A diagram showing typical discharge pressures versus flow rates for different types of compressor configurations is shown in Figure 44. Higher pressure refrigeration

services are accomplished in a radially split (barrel compressor), which has less stringent limitation as regards maximum allowable working pressure. A large barrel compressor is shown in Figure 45.

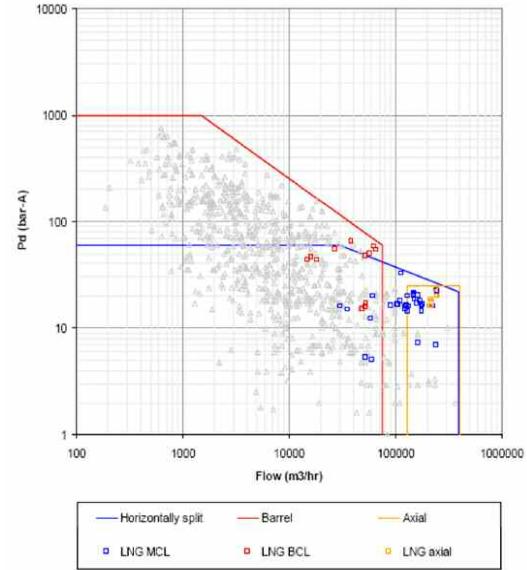


Figure 44. Flow Versus Discharge Pressure for Different Types of Compressors.



Figure 45. Large Barrel Type LNG Compressor.

Horizontally Split and Barrel Designs

Propane (precooling) refrigeration service is usually accommodated in one or two horizontally split compressor casings. The propane compressor is the most challenging machine in terms of flow coefficient and inlet relative Mach number. The low inlet temperature and high molecular weight (mole weight of 44 for propane) combined with large sidestream flow, require a precise fluid dynamic design of the gas path and particular attention to rotordynamics.

Mixed refrigerant service, which handles large capacities at low temperatures, requires the solution of many technical manufacturing and assembly issues. Mixed refrigerant service is typically implemented in two or three pressure levels in two or three compressors (horizontally split and barrel).

Other refrigeration services are often facilitated by one compressor casing in a back-to-back or compound arrangement. Particular attention has to be given to rotordynamics, due to the high pressure ratio and consequent large number of impellers (and long rotor length) generally associated with this.

For refrigeration services that require high pressure ratios, a three casing solution is typically used. The last casing is a barrel type compressor, while the first and second are horizontally split. These compressors do not generally present technical challenges other than those normally related to trains involving multiple large units.

Subcooling service can be accomplished by two or three casings (horizontally split and/or barrel design). The critical aspect of this service is the low temperature material selection requirements.

Double Flow Compressors

As LNG plant capacities increase, centrifugal compressor casings can become very large. At times, to minimize compressor casing size, a double-flow arrangement can be used. The inlet flow enters the casing via two nozzles, passes through each individual section of the compressor, and joins at the final diffuser exiting the casing through a single nozzle. The advantage of the double-flow arrangement is that the same casing size can accommodate a large volume flow. Losses in the flow paths through a double-flow compressor must be identical. In practice, this is difficult to achieve and the sensitivity is a function of the total head level. The lower the head level, the more closely the flow paths in the compressor should be the same. Another considerable advantage of a double flow compressor design is that an optimum speed match can be derived with the next stage casing. Double flow compressors have a limitation on the number of stages and therefore on pressure ratio, and this can lead to an additional casing being required for higher pressure ratios.

A double flow compressor is essentially two parallel compressors in a single compressor casing. Perfectly manufactured impellers are needed to duplicate performance for the two parallel process stages. It is essential that the internal flow paths be identical, and also crucial that the inlet piping be well designed to assure balanced volume flow rates to each section. A double flow compressor layout is shown in Figure 46, with a photograph of the horizontally split casing and impeller being depicted in Figure 47.

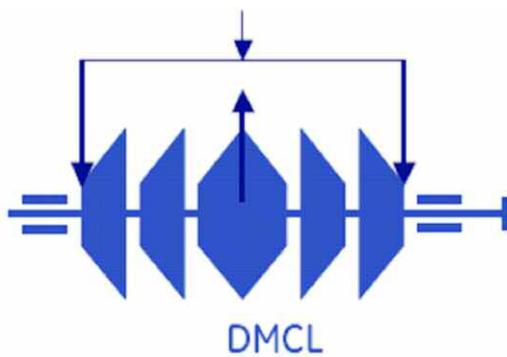


Figure 46. Double Flow Compressor Schematic.



Figure 47. Double Flow Compressor Casing and Rotor.

Back-to-Back LNG Compressor Designs

Another compressor arrangement used for LNG centrifugal compressors is a back-to-back arrangement. The compressor inlet flow enters one end of the compressor and is removed part way through the compressor, is reintroduced at the opposite end, and then exits near the center. A schematic of a back-to-back configuration is shown in Figure 48 and a back-to-back compressor is shown in Figure 49. This arrangement also allows for process cooling during the compression process, thus improving efficiency. It also can reduce the net rotor thrust, which may allow the use of a smaller thrust bearing compared to a conventional arrangement with a balance drum. An evaluation of the cross leakage between the two discharge nozzles separated by a center seal should be compared to the balance drum leakage in the conventional arrangement. If a damper seal is used, then the location of this seal at the rotor midspan is good from a damping perspective.

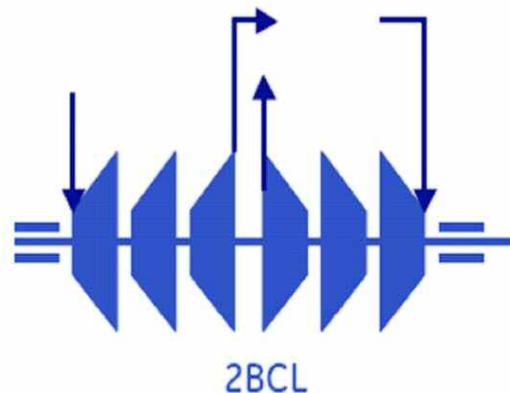


Figure 48. Schematic of a Back-to-Back Compressor.



Figure 49. Back-to-Back Barrel Compressor.

Another area requiring careful attention in a back-to-back compressor design is the rotordynamics. The center seal in this arrangement is at the midspan of the rotor. This seal is typically a labyrinth type design, and the large pressure drop across this seal and the interaction of the gas flow in the small seal clearance spaces produce considerable destabilizing excitation forces. Since these forces are produced at or near the rotor midspan, which coincides with the point of maximum rotor deflection, their effect on rotor stability is maximized (unless a damper seal design is used). For the conventional straight-through compressor arrangement, the balance piston is located closer to a bearing centerline and is often considered to have less of an influence on rotor stability compared to a midspan located back-to-back center seal. If this center seal causes the rotordynamics to be unstable or exhibit a low or negative log decrement, then other types of center seal designs to reduce aerodynamic excitation such as the use of shunt holes or swirl brakes can be investigated. If these designs are insufficient to

suppress an instability then other type seals such as a honeycomb or hole pattern seals may be required. The compressor manufacturer experience with all of these designs should be carefully reviewed. A typical shunt hole design for a back-to-back compressor is shown in Figure 50 and a honeycomb design is shown in Figure 51.

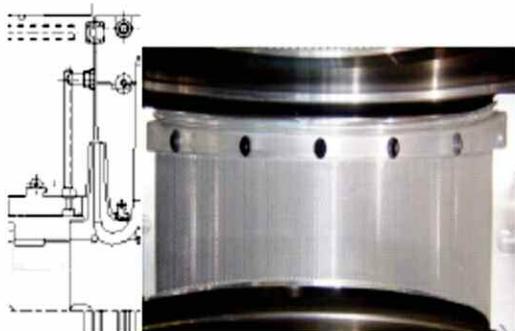


Figure 50. Shunt Holes typical for a Back-to-Back Compressor.



Figure 51. Honeycomb Seals.

Axial Flow Compressors

For large plants, the use of axial compressors provides high efficiency and operational advantages. The number of axial stages range between 14 and 15, to comply with the required pressure ratio. These machines require careful design of rotor and stator blading, accurate analysis of startup procedures, and precise evaluation of off-design operating conditions. Variable guide vanes are typically used to control the compressor. Particular attention should be paid to the linkages of these inlet guide vanes and avoid resonance of the group (blade plus linkage) to extend their operability and maintenance intervals. By its inherent design and the use of IGVs, the axial flow compressor can operate with a large operating range. Maps for an axial flow compressor and centrifugal compressor are shown in Figures 52 and 53, respectively.

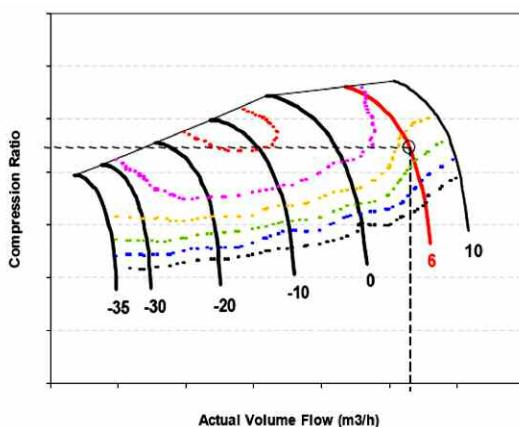


Figure 52. Axial Compressor Map—Fixed Speed with IGV.

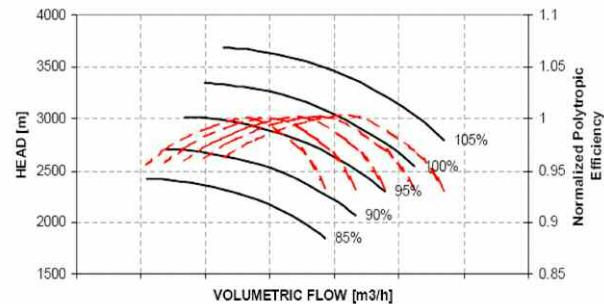


Figure 53. Centrifugal Compressor Map, Variable Speed.

Auxiliary Compressors

Besides the main refrigerant compressors there are other smaller compressors for auxiliary duties in an LNG plant. The number and size of these compressors depend on the LNG process and, in some cases, on feed gas conditions. Some of these compressor services might include feed gas booster compressors, CO₂ compressors, end flash compressors, and boil off gas (BOG) compressors.

The BOG compressors are usually installed close to the LNG tank, in order to recover gas vapors that boil off from the LNG tank. The BOG is usually driven by an electric motor, and the compressor may be a horizontally split machine or an integral compressor design. The most critical aspect of this compressor is the very low operating temperature (-120°/-161°C [-184°/-256°F]) and the design aspects associated with it in terms of materials, dry gas seals, inlet guide vanes, and, in the case of some integral type compressors, diffuser guide vanes.

End flash service can be accommodated with an electric motor driven horizontally split compressor with a gear box and barrel compressor as needed.

Centrifugal Compressor Design and Performance

Design fundamentals and terminology for centrifugal compressors can be found in Aungier (2000), Japikse (1996), and Cumpsty (1989), while application aspects are covered by Brown (1997).

The behavior of a compressor stage can be characterized in terms of parameters that specify its operating conditions as well as its performance. The dimensionless representation makes it possible to disregard the actual dimensions of the machine and its real operating conditions (flow rate and speed of rotation) and is thus more general as compared to the use of dimensional quantities.

Parameters commonly used to describe the performance of centrifugal compressors include:

Machine Mach Number

The Mach number, M_u , is defined as the ratio between the machine tip speed and the velocity of sound at the reference conditions:

$$M_u = \frac{U_2}{a_0} \quad (5)$$

where,

U_2 = Impeller peripheral speed

a_0 = Speed of sound at Inlet conditions

Flow Coefficient

The inlet flow coefficient for a centrifugal compressor is defined as follows:

$$\phi = \frac{4Q}{\pi D^2 U_2} \quad (6)$$

where,

Q = Volume flow

D = Impeller tip diameter

U = Impeller tip speed

Reynolds Number

The Reynolds number, Re , is defined as the ratio between inertial forces and viscous forces, evaluated in relation to assigned reference conditions. For centrifugal machines the following formulation is frequently used:

$$Re = \frac{U_2 b_2 \rho_0}{\mu_0} \quad (7)$$

where,

- b_2 = Impeller exit width
- μ_0 = Viscosity at the inlet conditions
- U_2 = Impeller tip speed.
- ρ_0 = Gas density at inlet conditions

Impeller Work Coefficient

The impeller imparts energy to the working fluid to raise the total enthalpy and the impeller work coefficient is defined as:

$$\tau = \frac{\Delta h}{u_2^2} = \frac{C_p(T_{02} - T_{01})}{u_2^2} \quad (8)$$

where,

- T_{01} and T_{02} = Total inlet and outlet temperature
- C_p = Specific heat
- U_2 = Impeller tip speed

Polytropic Efficiency

The polytropic efficiency is given by:

$$\eta_p = \frac{k-1}{k} \cdot \frac{\log\left(\frac{P_2}{P_1}\right)}{\log\left(\frac{T_2}{T_1}\right)} \quad (9)$$

where,

- k = C_p/C_v
- P_2 and P_1 = Outlet and inlet pressure
- T_2 and T_1 = Outlet and inlet temperature

Head Coefficient

The head coefficient, μ , is defined as:

$$\mu = \tau \eta_p \quad (10)$$

2D and 3 D Impellers

Impeller flow coefficient and geometry are related. Lower flow coefficients correspond to shrouded, two dimensional (2D) impeller geometry. Higher flow coefficients correspond to shrouded, three dimensional (3D) impeller geometry as shown in Figure 54. For very high flow coefficients, an axial flow blade design is utilized. LNG centrifugal compressors typically use a combination of 2D and 3D impellers.

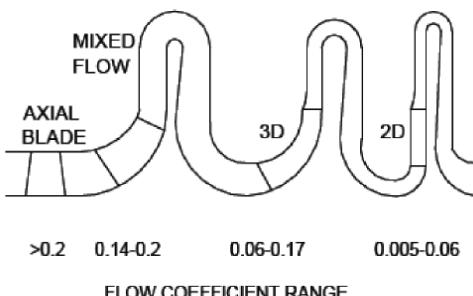


Figure 54. Typical Geometry for Different Impeller Types.

A 2D impeller has a basically constant blade angle from disk to shroud along the leading edge of the blade. It can therefore only match the incidence angle at one point on the leading edge, and has an increasingly mismatched leading edge flow field in moving toward the disk or shroud from that point. However, as 2D impellers are typically used for low flow coefficients (smaller inlet flow areas), this effect is not severe. In a high flow high inlet eye diameter 3D impeller, the blade angles along the leading edge may be varied from disk to shroud to match the incoming flow field and obtain optimum incidence angles. However, this typically occurs at the impeller's design flow rate, and any increase or decrease in flow rate will result in nonoptimal incidence and lower performance. The efficiency of an impeller is a function of the inlet flow coefficient. As the flow coefficient increases, there is an increase in impeller efficiency. While it would seem that only 3D impellers should be used due to their higher efficiency, it generally means a large stage spacing (axial length), which will result in a longer compressor that could increase rotordynamic complexity and cost. Consequently 3D impellers are typically used only for the first or first few impellers unless an incoming sidestream calls for a high flow coefficient stage further down the machine. High flow and high efficiency design staging may require reduced speeds due to higher stress levels encountered. The compressor supplier and designer's experience in selecting various flow coefficient impellers should be carefully reviewed.

Machine Mach Number and Inlet Relative Mach Number

As flow increases, the impeller eye diameter also increases, and because of the high molecular weight of some refrigerants such as propane (MW = 44) and low inlet temperatures, high Mach conditions often exist for these compressors. Mach number is generally defined as the ratio between gas speed and the speed of sound. In centrifugal compressor applications two Mach numbers commonly referred to are the peripheral Mach number (M_u) and the inlet relative Mach number (M_{1rs}). The peripheral Mach number (also known as machine Mach number) is commonly used because it is easily calculated and gives a rough indication about the criticality of the stage, and provides a benchmark for the impeller exit velocity into the diffuser. The inlet relative Mach number is much more important and gives an indication of the gas behavior at the impeller eye. It is based on the inlet relative velocity at the shroud as shown in Figure 55. It is very important to carefully evaluate different manufacturers' calculations of inlet relative Mach numbers, taking into consideration the location of measurement and other details. Some manufacturers use the mean or root-mean-square (rms) diameter, while others use the tip as the reference location. Sorokes and Kopko (2007) have provided a valuable paper covering this important area that helps users understand the issues involved when making comparisons between compressor designs.

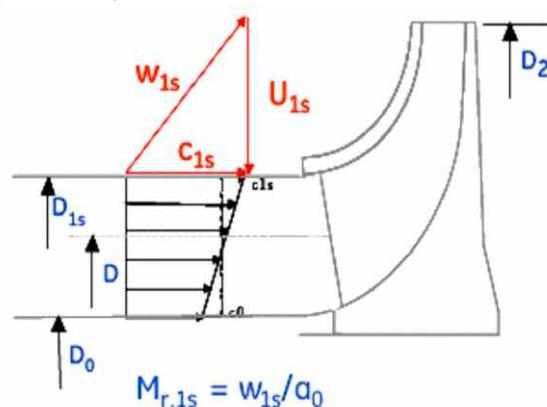


Figure 55. Representation of Velocity Triangle for Inlet Relative Mach Number ($M_{r,1s} = W_{1s}/a_0$).

The change in operating envelope as the Mach numbers change is shown in Figure 56. Shock waves within a compressor impeller are shown in Figure 57. This sonic condition (Mach number = 1) for an impeller designed for subsonic flow constitutes a “stonewall” barrier to any further flow increase, which shortens the operating envelope.

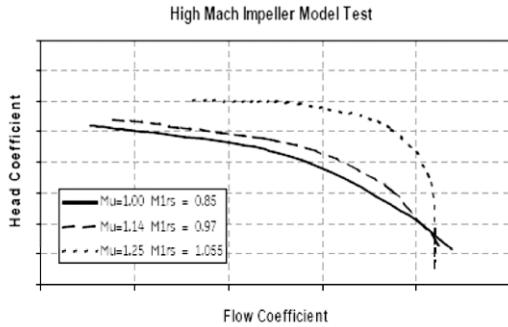


Figure 56. Effect on Operating Range of Different Peripheral and Inlet Relative Mach Numbers.

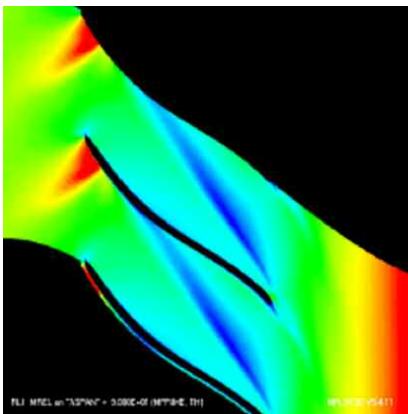


Figure 57. Shock Waves Inside Compressor.

Inlet relative Mach numbers for these type impellers have been validated up to values of 0.95. Model test activities have brought inlet relative Mach number up to 1.06 with good performance at design operating point but with a narrow operating range. This problem can be overcome by introducing IGVs. IGVs can prerotate (swirl) the flow approaching the impeller, thus reducing the relative velocity vector and therefore the inlet relative Mach number. This results in improved efficiency and a wider operating range. Plots of flow coefficient versus peripheral Mach number and inlet relative Mach numbers are shown in Figures 58 and 59, respectively.

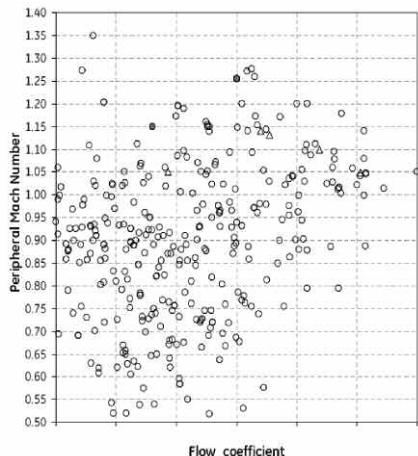


Figure 58. Peripheral Mach Number Versus Flow Coefficient.

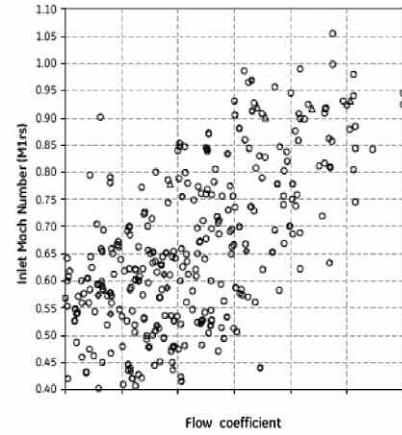


Figure 59. Inlet Relative Mach Number Versus Flow Coefficient.

A ratio of the two Mach numbers can be also used to verify the “stiffness” of the impeller as shown in Figure 60. A high ratio means that the external diameter is close to tip diameter, the gas has no space to develop a smooth radial flow, and it reduces the operating range. The introduction of mixed flow impellers (Figure 61) can overcome this issue.

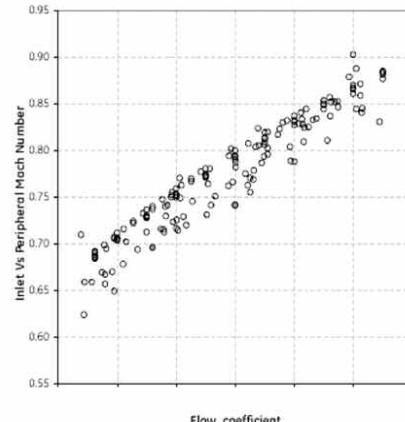


Figure 60. Impeller Stiffness Map (Ratio of M_{1rs}/M_w) on Ordinate.



Figure 61. Mixed Flow Impeller.

The use of splitter vanes, wherein the inlet area is opened up, results in a reduction in the inlet relative Mach number, but arrangement does not allow for future compressor growth. Splitter vanes are partial length vanes between adjacent full blades. The splitter blades reduce the inlet blade metal blockage area at the minimum passage area, and thus allows the impeller to pass a greater flow rate prior to impeller choking. An impeller utilizing splitter vanes is shown in Figure 62.



Figure 62. Centrifugal Compressor with Splitter Vanes.

Impeller Head Per Stage

For centrifugal compressors operating with high molecular weights and low temperature gases, users and manufacturers have traditionally tended to limit the head per impeller within the range of 3200 to 3400 m (10,500 to 11,155 ft) based on considerations that stages with high head could have a low pressure rise to surge and a narrow operating range between the design point and surge. Current experience shows it is possible to design centrifugal impellers with heads in the range of 5000 to 6000 m (16,400 to 19,685 ft) keeping satisfactory characteristics of flow stability and operating range. High head is attained with the combination of high tip speeds (allowed by higher strength materials) and high head coefficients above 0.54. These head levels must be carefully evaluated prior to implementation. Head per impeller and head coefficients in LNG service are shown in Figures 63 and 64, respectively.

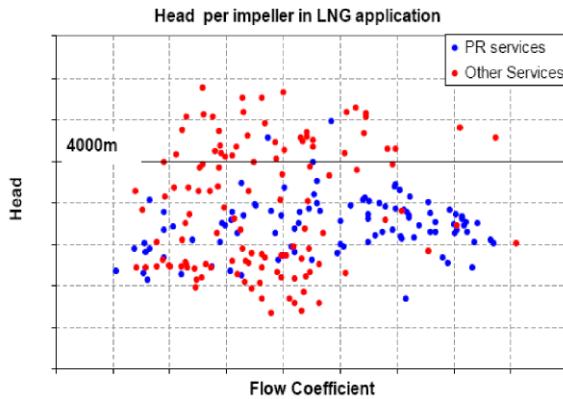


Figure 63. Head (in Meters) per Impeller in LNG Applications Versus Flow Coefficient. PR Represents Propane Service.

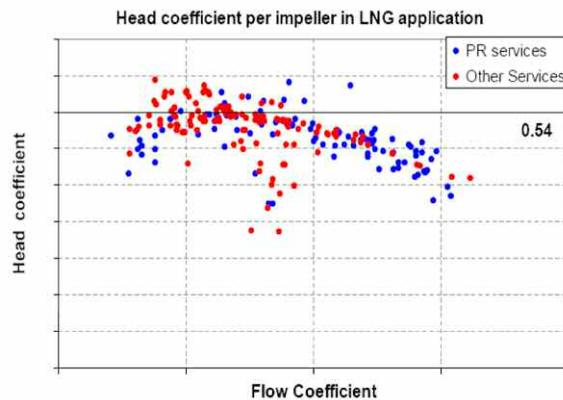


Figure 64. Head Coefficient per Impeller in LNG Applications Versus Flow Coefficient. PR Represents Propane Service.

Design tradeoffs come into play here as lower head per impeller means more impellers resulting in an increase in bearing span, which could lead to rotordynamics issues. Higher head per impeller requires increased rotational speed resulting in operation at higher Mach number levels, which as stated earlier can restrict the operating range and reduce efficiency. High head coefficient stages provide a narrower operating range and lower rise-to-surge. Low head impellers provide much greater head rise-to-surge are easier to control and provide a wider operating range.

High efficiency of large LNG compressors is an important parameter. The efficiency of a centrifugal stage is a function of capacity and head, and most impellers in LNG service fall in the polytropic efficiency range of 82 to 86 percent. Further improvements in efficiency over the short-term can be attained by reducing internal impeller flow recirculation by utilizing low-clearance abradable seals. The use of vaned diffusers on the discharge of the centrifugal compressors in place of the vaneless (free-vortex) can also improve efficiency. For axial compressors, efficiency improvement can be attained by reducing the radial clearances on the rotor blade tips by the utilization of special seals.

As more impellers are added to the compressor, "off-design" operation gets affected and stability inherently reduced. This occurs because the individual impeller performance maps dictate the overall surge point. The situation is further complicated by the large side load flows and the complexities of uniform mixing at the sidestream mixing plane. Impeller selection directly affects the compressor's overall operating range. Careful staging will allow the designer to take advantage of each impeller's best operating range. A typical curve and limits for a centrifugal compressor stage are shown in Figure 65.

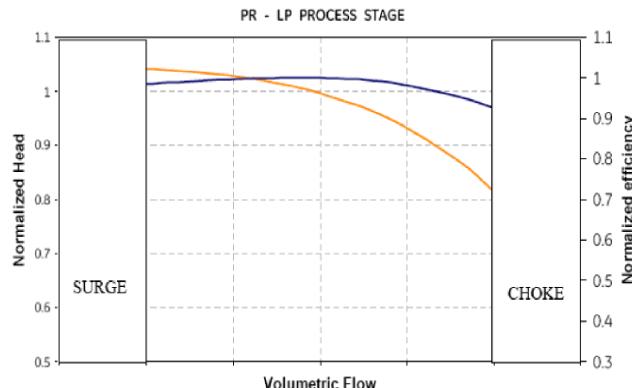


Figure 65. Typical Curve and Limits for a Centrifugal Compressor Stage. Normalized Head and Efficiency Shown.

Aerodynamic Efficiency—Range Considerations

The total stage efficiency is a function of impeller efficiency, stator component efficiency, return channel efficiency, and exit scroll efficiency, and all of these plays a part in the optimization of the machine. In the case of sidestream machines that are typical in LNG service, the way in which the side load flow is introduced and the induced flow and temperature distributed are also very important and the subject of much analytical and experimental study. The injection volute design can be optimized for sidestream mixing by going with radial injection rather than the traditional tangential injection. The radial injection has much lower pressure loss coefficient (approximately 0.6 instead of 2.7). The advantage of tangential injection is the reduced space required and therefore the reduced bearing span. Typical configurations for sidestream injection are shown in Figure 66. The configuration on the left is the tangential sidestream approach and the currently used radial configuration is shown on the right.

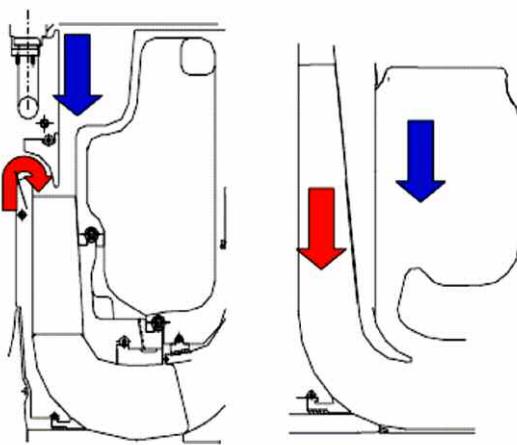


Figure 66. Tangential (Left) and Radial (Right) Sidestream Injection Discharge Volute Configurations.

Compressor discharge systems can be optimized by changing from constant section collectors to variable section collectors, which have a much lower loss coefficient compared to constant section collectors. The two designs are shown in Figure 67, and a variable section discharge volute used in a model test is depicted in Figure 68.

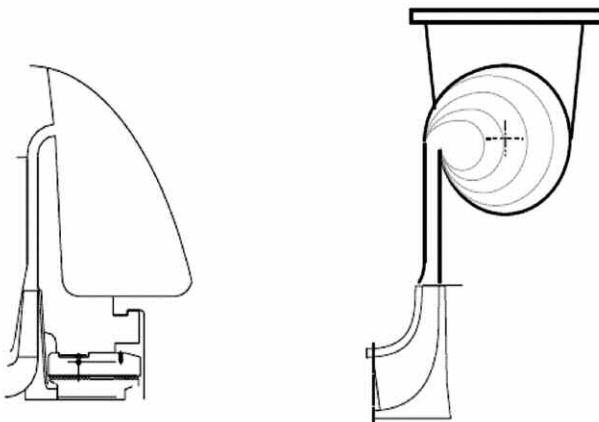


Figure 67. Constant Section (Left) and Variable Section (Right) Cross Sections.

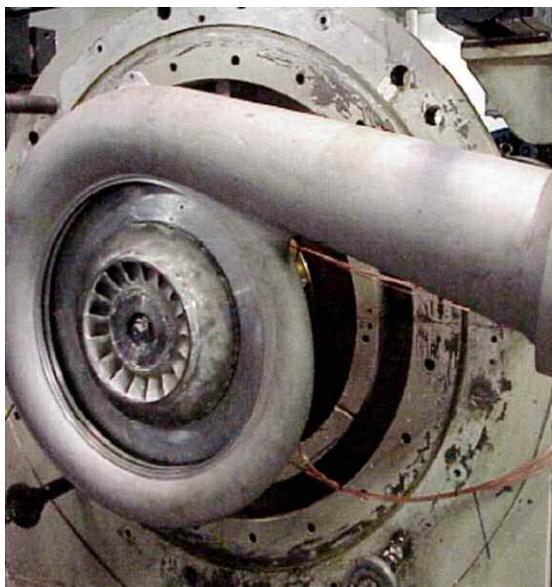


Figure 68. Variable Section Discharge Volute for Model Testing.

Computational Fluid Dynamics Studies

Centrifugal compressors flow fields are among the most complex in turbomachinery. They are characterized by strong 3D effects, heavy viscous phenomena (producing separation and wakes), and high secondary flows. Until around 1995 semiempirical methods or simplified fluid dynamic models were the most sophisticated tools available. Powerful computational fluid dynamics (CFD) tools are available today to attack these problems with complex models, but calculation accuracy is still an open issue. Well-calibrated CFD tools can help during detailed engineering to better predict nondimensional parameters and help to reduce manufacturer design margins. CFD is also used to verify and increase operating range. Details on CFD analysis may be found in Bonaiuti, et al. (2003).

Specific aerodynamic CFD studies can be carried out on items such as the impeller and return channel and on stator parts such as the discharge scroll, inlet plenums, and sidestreams. Typically the use of CFD allows the verification of predicted envelopes and performance and also examination of blade loadings and the flow separation. CFD can be done on single stages or on multiple stages. A typical impeller grid is shown in Figure 69. Particle traces along an impeller are shown in Figure 70, and indications of compressor impeller stall are shown in Figure 71. Impeller choke is shown in Figure 72.

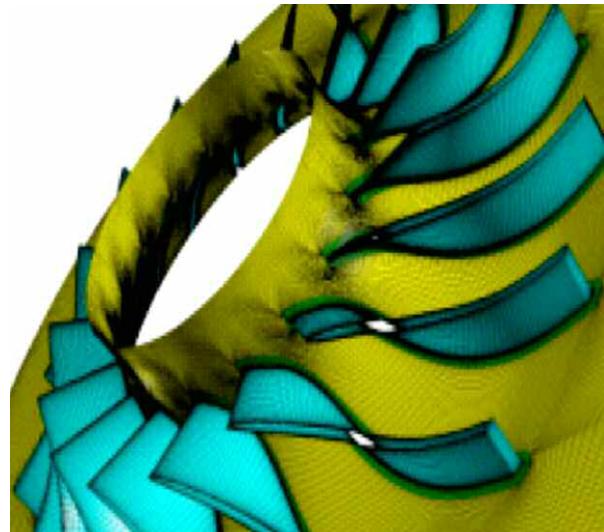


Figure 69. Impeller Grid for CFD.

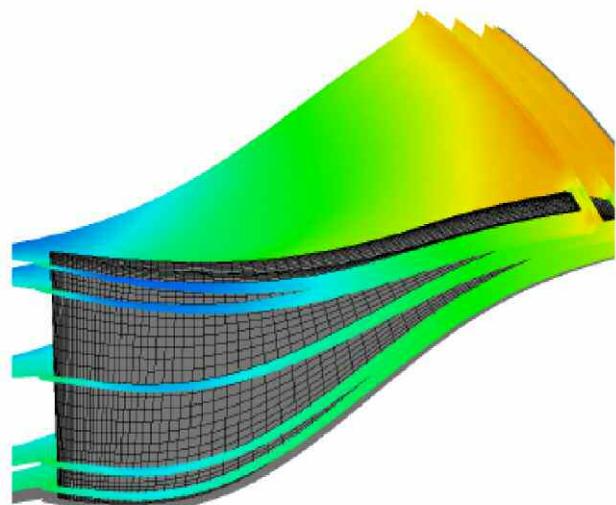


Figure 70. Particle Traces along Impeller.

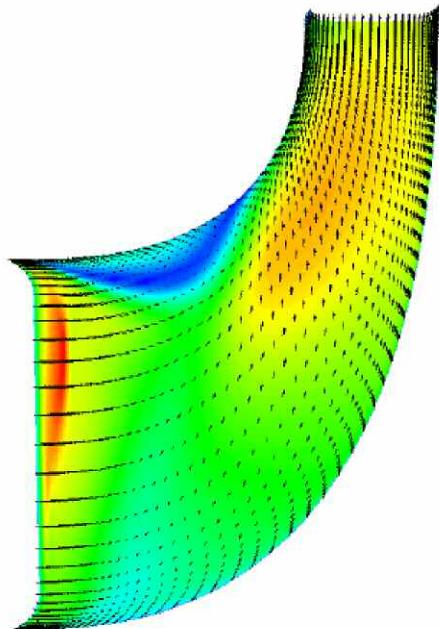


Figure 71. Impeller Stall.

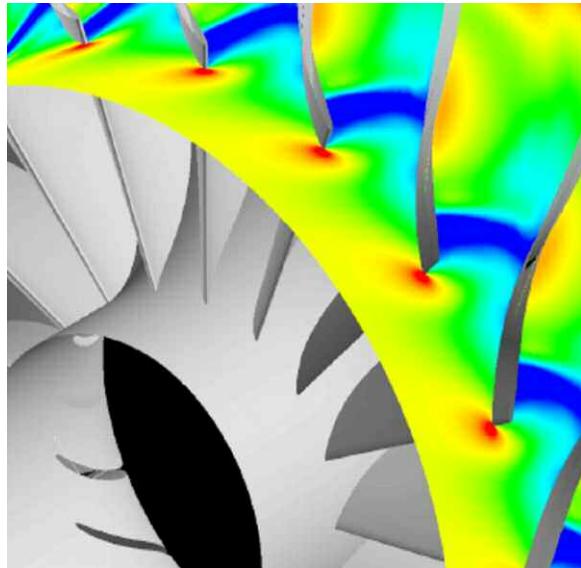


Figure 72. Impeller Choke.

It is important to note that a strong correlation with testing and experience is still needed. Performance can be assessed through CFD on a subcomponent basis, then coupled together by a “stacking code” to obtain overall performance.

Scale Model Studies

Scale model testing is very useful for performance estimation and validation of any computational tool. The geometry of the stage is reproduced and the dynamic similarity is achieved as far as it concerns the flow coefficient, the Mach number, and the density ratio/volume reduction. Dimensionless coefficients are used for design because they depend only on dimensionless parameters and allow a generalized analysis.

A constant flow coefficient implies that relative flow angles are the same. Mach number similarity means that perturbation velocity effects are the same. A constant density ratio implies that the specific volumes are the same resulting in similar velocity triangles. For large machines, the Reynolds number is not critical.

However, a constant Reynolds number would imply that viscosity effects are the same. A comprehensive explanation of dynamic similitude can be found in Key (1989).

Compressor Design Tradeoffs and Compromises

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. 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). Another excellent reference is chapter six of Japikse (1996), 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. A discussion of parameters relating to refrigeration compressors is also made by Peters (1981).

Head Rise to Surge and

Operating Range (Surge to Choke Margin)

A positive continuous head rise to the compressor surge point at constant speed is crucial to stable compressor operation and reliable responsive process control. The achievable rise to surge is directly influenced by the required compressor operating range. The head rise to surge (HRTS) is a measure of the rise in the head (at constant speed) when the operating point goes from the operating point to the surge line. Other common terms related to HRTS include pressure rise to surge and pressure ratio rise to surge. In some cases, specifying a hard number such as 5 percent may result in a reduced efficiency and also a reduced “overload” capacity. Selecting a lower HRTS may result in a higher efficiency and an increased operating margin. With refrigeration machines, therefore, specification of excessive HRTS and surge margin criteria may result in a moderate overload capacity.

The refrigeration process requires operation at nearly constant discharge pressure due to condensing temperature. Either the driver speed or suction pressure must reduce as flow moves toward surge. Reduction of suction pressure below atmospheric pressure is usually not desirable. Due to various aerodynamic laws, the surge to stonewall/choke flow range for refrigeration service is less than that seen on various other types of applications. Hence, imposition of an excessive HRTS and/or surge margin criteria may result in only minimal overload capacity. A recent application has shown a compressor working well with HRTS below 2 percent on the guarantee point and lower on alternate operating points.

Head per Section

This is an area where there is an interaction between mechanical design and aerodynamic design. A larger number of impellers may cause an excessively long shaft and thus a high critical speed ratio. Fewer impellers per section would result in higher head per stage. This, in turn, calls for higher speed operation at higher Mach numbers. Operation at higher Mach numbers will cause a restriction in operating range and cause the head rise curve to flatten, and may reduce the efficiency. Further, as the flow reduces, either the driver speed or suction pressure must be reduced.

Aerodynamic Mismatching of Stages

As more impellers are added to the compressor, “off-design” operation gets effected and stability inherently reduces. This occurs because the individual section performance maps dictate the

overall surge point. The situation is further complicated by the large sideload flows and the complexities of uniform mixing at the sidestream mixing plane. Impellers must be carefully selected in concert with the other impellers of the compressor section, and interactions between different compressor stages must be well understood to ensure that the overall compressor has an acceptable operating range under both design and off-design conditions. Careful staging will allow the designer to take advantage of each impeller's best operating range.

Rotordynamic Considerations

Rotordynamics play a very important role for large LNG centrifugal compressors where the rotational speeds may be low but where bearing spans can be very large (up to 5850 mm [19.2 ft]). In general these large rotors are quite rigid and have good separation margins. API 617, Seventh Edition, (2002) stability analysis (Level 2) must be always carried out and cross-checked with manufacturer experience.

Rotor displacement, deflection, amplification factors, and separation margins must be evaluated. Many LNG compressors have "drive through" arrangements with large couplings on both ends of the machine, and these overhung weights would affect the rotordynamic model.

LNG compressor trains can include several compressor bodies, the driver, possibly gears, and helper/starter motors resulting in long strings with a multiple number of seals, bearings, and couplings. A train torsional analysis should be performed in accordance with the requirements of API 617 (2002).

Per API, when lateral or torsional natural frequencies cannot be avoided by the necessary separation margins, then a stress analysis should be performed to demonstrate that the resonances have no adverse effect on the complete train. When synchronous motors or motors with variable speed drives are included in the train, additional steady-state and transient analysis should be performed. The supplier's experience with this type analysis should be carefully reviewed based on previous jobs with compressors running in the field. To minimize stress concentration, an integral flange on the motor may be specified. A torsiograph connection can also be included on the appropriate equipment in the train to enable checking the calculated torsional speeds during a string test.

In addition to the traditional rotor bearing system forcing functions that can cause vibration, there are a host of aerodynamically induced destabilizing forces that can cause rotor instability typically for high density fluid services. Gas forces act on the rotor either as a forced excitation where the unsteadiness and nonuniformity of the gas flow can create problems of rotating stall or incipient surge. This can occur in either the diffuser (vaned or vaneless) or at times in the inducer of the impeller. It typically originates in the diffuser section. These can set up a host of subsynchronous vibrations that can, under certain conditions, be potentially unstable.

There are self-induced excitations that are the result of the pressure distribution generated by the motion of the rotor itself inside the close clearance areas (such as the internal seals). When these forces overcome the damping, the rotor can become unstable creating a serious operating problem. The balance piston seal is often a contributor to this problem and the issue can become more severe for back-to-back compressor configurations.

The risk of rotor instability can be evaluated by using several empirical approaches where experience with operating machines can be compared. There are curves available to evaluate the propensity of the rotor to go unstable, and this evaluation would be part of the design review conducted by the mechanical department.

Recent work done in the area of rotordynamic stability has been presented by Bernocchi (2006). Some values derived by new tools in terms of prediction are presented in Figure 73 for the log decrement and Figure 74 for the amplification factor.

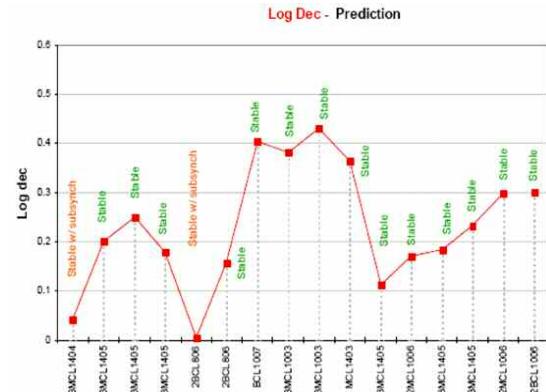


Figure 73. Stability Analysis of LNG Compressors.

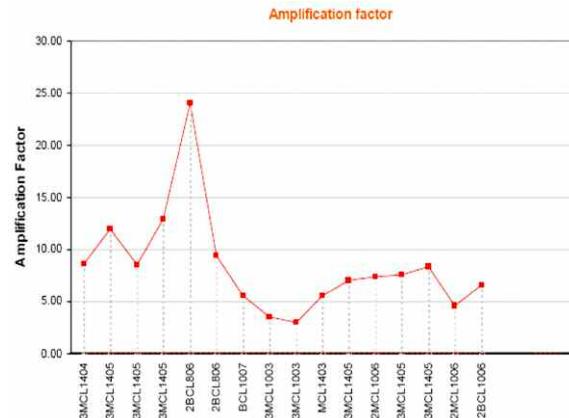


Figure 74. Amplification Factors of LNG Compressors.

Large single shaft gas turbine drivers such as the Frame 7 and Frame 9 or electric motors for motor driven LNG plants (some rated at 65 MW) require the installation of large couplings and hubs. Careful attention must be paid to hub hydraulic fitting, procedure, and tools, in order to avoid any damage on the shaft ends.

Casing Stresses and Design

Casing stresses and sealing of horizontally split flanges, etc., are very important considerations, especially for large LNG compressors, often calling for finite element analysis as shown in Figure 75. There are several designs available to deal with higher working pressure requirements including the use of flat end casings. If needed finite element analysis (FEA) can be used to:

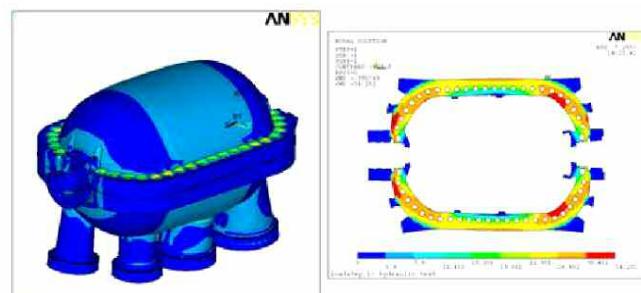


Figure 75. Casing Finite Element Method (FEM) Analysis.

- Verify equivalent stresses within the casing.
- Evaluate contact pressures on horizontally split flanges.
- Evaluate stresses at bolt hole locations.

It is also possible to verify and calibrate FEA models by strain gauging casings during hydro tests.

Dynamic Simulation

As reported by Valappil, et al. (2004a), 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. 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 76. The operability and profitability of the plant during its life depend 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.

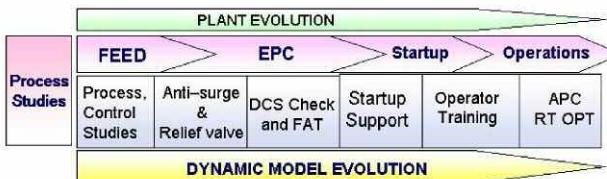


Figure 76. Use of Dynamic Simulation Through the Design and Operations of an LNG Plant.

There are several benefits to be realized by using the dynamic simulation in the various stages of an LNG project. These include both design and operational benefits. On the process side, it is an important tool for evaluating the antisurge 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 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 also during transient conditions such as trips and startup. Startup of LNG strings with single shaft drivers often needs simulation to check the motor sizing and dynamic behavior of the string. Dynamic simulation is also helpful in defining antisurge philosophy, determining the antisurge valve size, and defining safe shutdown schemes and procedures. Details may be found in Valappil, et al. (2004a, 2004b). Some typical dynamic simulation results are shown in Figure 77.

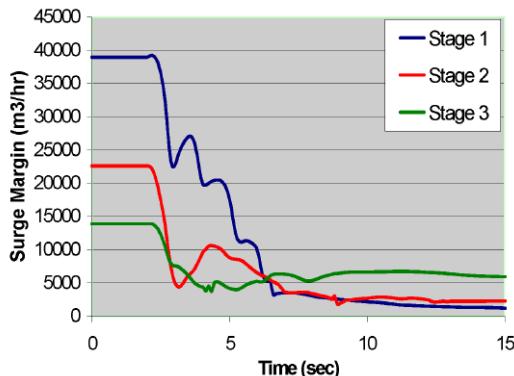


Figure 77. Response of a Refrigeration Compressor in a Trip. Surge Margins for the Three Stages Are Shown as the Machine Speed Drops. (Courtesy Valappil, et al., 2004b)

Piping and Nozzle Considerations

Compressor nozzle velocities should be carefully reviewed. For new compressors, nozzle velocities should be limited to 34 to 37 m/sec (110 to 120 ft/sec) or less (depending on the molecular weight) for any specified operating range. The higher the molecular weight and the lower the temperature, the more these values should be reduced. Higher nozzle velocities can translate into higher pressure drops and more noise. Details on specifying and evaluating parameter for compressors may be found in Blahovec, et al. (1998), and piping related issues may be found in Gresh (1992).

If the inlet piping for double flow machines is not well designed, poor performance and premature surging will occur since each section will be operating on a different performance curve flow point, yielding different heads/discharge pressures and different efficiencies. Keeping the inlet piping to each section of a double-flow compressor identical can be accomplished with different arrangements. Experience has shown that the preferred arrangement uses a drum to split the flow. While a "Y" design with a proper upstream straight run of pipe may be an acceptable design, it should be noted that even a small discontinuity in the piping upstream of the "Y" can cause the flow to shift to one leg of the "Y." A lower velocity will minimize this effect and result in an improved flow distribution.

ELECTRIC DRIVE LNG

Interest in All Electric LNG Plants

There is increasing interest in the LNG marketplace for using electric motors to drive LNG plant compressors as opposed to the more traditional practice of using gas turbine drivers. This area has been discussed in several papers including Martinez, et al. (2005), Kleiner, et al., (2003), Hori and Hitoshi (2004), and Chellini (2004). Electric LNG plants have some advantages but are more expensive in terms of capital expense. The advantages include:

- Potential for higher plant availability and lower compressor driver life cycle costs.
- Improved plant operational flexibility (by using full-rated power variable frequency drives that allow compressor speed variation and startup without depressurization).
- Decoupling of plant production and ambient temperature.
- Reduced maintenance costs and downtime for the motors and the VFDs (as compared to gas turbines).
- Possible reduction in plant air emissions (by replacement of the gas turbine drivers with motors). However this would *only hold if a combined cycle were used for power generation.*
- Enhanced safety in the process area due to the absence of fuel gas.

The choice and cost of the associated power generation solution can, however, significantly impact the economic viability of an electric drive LNG plant, and an evaluation would have to be made on a project specific basis.

Electric Motor Design and VFD Suppliers

The Frame 7EA ISO power rating is approximately 83 MW. If one considers a 12 to 15 MW starter-helper motor, the size of a single electric motor required to replace the gas turbine starter-helper configuration can range from 95 to 100 MW. Given the vast experiences in building large generators well over 100 MW, motors of this size are not considered new technology. In essence, these motors can be built without employing new designs or materials.

Suppliers indicate that two-pole motors equipped with variable frequency drives are the most attractive and economical solution for power ranges needed for large LNG drives. Motor suppliers state that the two-pole motors can be built in general accordance with API 546 (1997) standards and operate at high efficiencies.

Issues to be addressed include:

- The startup of the compressor train (ensuring adequate torque-speed capability to allow appropriate acceleration with minimal dwell time at the critical speeds).
- Restart of the compressor train after a motor trip (time taken).
- Range of operability (efficient turndown)—there are limitations generally imposed on the speed range of the motor and these are typically rotordynamic constraints.
- Torsional analysis of the full compressor train under startup conditions, compressor transients such as surge, and electrical transients such as power dips.
- Interaction of torsional and lateral vibration. This could be accentuated with the presence of a gearbox that may be needed to optimize compressor operation.
- Sensitivity of the motor to excitation of its critical speeds.
- Effects imposed on the power system resulting from motor trips (this is by far the most critical issue affecting the electrical system stability), switch gear, and power distribution failure.
- VFD
 - Availability and efficiency.
 - Induced low frequency harmonics.
 - Stable operation during a power disturbance.
 - Rapid speed response and range of variable speed (*this is usually constrained by the rotordynamics of the motor and this requirement is linked very much to the process*).
- Operation at degraded levels of the VFD may present problems.

VFD configurations utilizing both load commutated inverter (LCI) and gate commutated turn off thyristor (GCT) technology are available.

The LCI is a conventional drive system that uses a low switching (on/off) frequency device known as a thyristor. The GCT is a more advanced drive system that uses the latest high switching frequency device. The VFD provides a high degree of “ride-through” capability when there is a supply voltage dip. The use of 12, 24, or 30 pulse VFDs results in lower harmonics induced within the network. A project specific evaluation must be made on a case-by-case basis, with respect to the use of harmonic filters. Further, the low torque ripple would help in the torsional design of the motor-compressor string. Although a “starter VFD” would result in a lower total installed cost, a “Full VFD” provides maximum operating flexibility during start up, trips, electrical transients, ambient swings, turndown, and is the generally accepted solution.

Supporting Power Plant Solution Considerations

Power plant configuration and sizing require a detailed analysis in selecting a solution that best fits the need of the project. This is perhaps the most critical aspect of implementing an all electric solution, and a very detailed study would need to be done of transient characteristics of the overall electrical system.

It is important that in evaluating the overall availability, the availability of the electric supply system (grid or self generation) be taken into account and built into the overall project economic evaluation.

With a few exceptions, a long-term reliable and risk-free electric supply in most of the countries where LNG plants may be built would not be available for the quantum of power demand by an LNG plant. Further, there may be considerable political risk and uncertainty with regard to the cost of tariff and energy pricing over the life of the LNG plant. Options for power plants might include:

- Simple cycle power plants—careful consideration of the robustness of the solution and a careful consideration of spared capacity must be made taking into account DLN operability.
- Combined cycle self generation—again careful tradeoffs must be made regarding the configuration, redundancy, and transient operability.

- “Across the fence” utility electric supply (preferably from a reliable and economical supplier).

An “inside the fence” simple cycle power plant solution is the least complicated to operate, but it has the shortest installation time and lowest overall total installed cost (TIC) when compared to a combined cycle plant. A combined cycle power plant is more complicated to operate (high pressure steam system ranging from 800 to 1500 psi), requires greater attention to maintenance, requires a larger plot plan and is greater in TIC than the simple cycle solution. However, the combined cycle solution does benefit from a much greater thermal efficiency (48 percent versus 33 percent) and comes in larger blocks of power.

To summarize, power plant reliability and availability are very important to the success of an electrically driven LNG plant. Consolidation of power can impact LNG plant availability because of power instabilities due to generator trip, mechanical failures, electrical faults, and transient events. These influences can be mitigated by selecting a power plant that consists of multiple power generation packages. A thorough investigation of the power plant solution must be performed on a case-by-case basis, taking into account: capital cost, operating cost and the stability of the system to transient upsets. A comparison of simple and combined cycle solutions is provided in Table 4.

Table 4. Comparisons of Simple Cycle and Combined Cycle Power Plant Solutions.

| Simple Cycle | Combined Cycle |
|---------------------------------|---------------------------------|
| • Lower thermal efficiency | • Higher thermal efficiency |
| • Lower initial CAPEX | • Higher Initial CAPEX |
| • Lower OPEX | • Higher OPEX |
| • Higher stability | • Lower Stability |
| • Small real estate requirement | • Large real estate requirement |
| • Simple to operate | • Complex to operate |
| • Shorter delivery | • Longer delivery |

Issues to be Considered for an All Electric Solution

While an electric motor driven LNG plant is technically feasible, careful attention is required when evaluating the project economics. Key issues that have to be considered are:

- Sensitivity of the project to capital expenditures (CAPEX).
- Value being placed on the fuel gas.
- CO₂ emission considerations and possible credits.
- Real estate availability for the LNG plant and power plant solution.
- Export power considerations.
- Load shedding philosophy in the event that power is also exported.

Implementation of an All Electric Solution

As reported by Hortig and Buerkle (2006), the first implementation of an all electric LNG plant is at Snøhvit, Norway. The train will have three refrigeration compressors driven by variable speed motors. This solution was supported by power generation from five LM6000 gas turbines. All of the equipment was extensively string tested including the three main refrigeration strings and the power plant. The string tests comprised the electric equipment including the high velocity switch gear, transformers, and controls.

TESTING OF LNG TURBOMACHINERY

Overview of Testing

Testing capabilities are a very important requirement in the LNG business. Not only is it important to verify the performance of the compressors, but also to verify the rotordynamic and aerodynamic stability of the large and high horsepower compressors, and the proper operation of the overall train of gas turbines, compressors, and, where applicable, gears, and/or helper/starter motors.

Consideration should be given to a high speed balance of the compressor rotors per API 617 (2002). LNG compressor impellers

are usually large, and sometimes these large impellers will not take a proper set on the shaft until they reach operating speed. If these impellers are only low speed balanced, there is a possibility that, when the impellers reach operating speed during the mechanical test, they can "move" and change the balance of the rotor. This could require rebalancing and possible schedule delays.

Testing can be broken into the following elements:

- Scale model tests for aerodynamic performance
- Mechanical run tests
- ASME PTC-10 (1997) type tests for aerodynamic performance verification—may be Type 2 or Type 1
- Full density testing to validate rotordynamic stability under load
- String tests—for overall validation of the string and job components
- In manufacture testing—this would include numerous tests and checks as the components are manufactured and built as standard part of the OEMs procedures.

The most beneficial test regime has to be selected based on factors such as how proven the respective design configuration is, evaluation of aerodynamic and rotordynamic risks, and by evaluating and trading off performance risks and corrective actions with project schedule.

Scale Model Tests for Compressor Aerothermal Behavior

Scale model testing is very useful for performance estimation and validation of any computational tool. The geometry of the stage is reproduced and dynamic similarity is achieved in terms of key aerodynamic parameters such as the Mach number and flow parameters. Some parameters such as the Reynolds number and specific details, such as surface roughness effects, end wall flows, and seal clearances, cannot be scaled but corrections can be applied to deal with this. The scale model tests provide a good insurance as to how the compressor will behave, and provides an opportunity to modify design during the design process (the model test is performed prior to impeller fabrication) as opposed to making changes after noting problems during the PTC performance test.

Scaled model test activities can be carried out to better predict the critical impeller stage that may be at the boundary of the manufacturer's experience. Model tests can be applied to the front stage, intermediate stages (with or without sidestreams), or the last stage. Typical arrangements for an intermediate stage model test are shown in Figure 78. Several measurement sections are positioned inside the channel equipped with pressure probes (Kiel, Cobra, five-holes, static), temperature probes, and dynamic pressure probes to identify stall/surge. Scaled model tests are very practical; any performance issue can be easily identified and corrected.

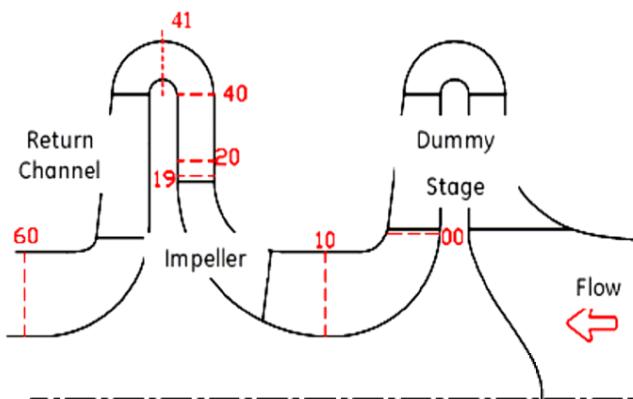


Figure 78. Typical Arrangement for Intermediate Stage Model Test.

Wind tunnel activities (Figure 79) are also very important and very easy to perform. The purpose is to test a stator part such as volutes, sidestreams, or return channel.



Figure 79. Wind Tunnel Test.

Mechanical Run Tests

These are API 617 (2002) test specified and are essentially a four hour endurance test run at full speed in a no-load condition. This is done under vacuum with the inlet and discharge flanges blanked off. Acceptable vibration characteristics including the mechanical and electrical runout levels should be defined before the equipment is purchased based on the recommendations of API 617 (2002) and the suppliers experience. Generally, the larger the compressor, the lower the operating speed, and therefore the higher the allowable peak-to-peak unfiltered vibration and the higher the allowable electrical and mechanical runout. An unbalanced rotor response test should be performed as part of the mechanical running test per API 617 (2002). This unbalanced response test should only be performed on the first rotor tested. The results of the mechanical run including the unbalance response verification test should be compared with those from the analytical model. Spare compressor rotors should also be given a mechanical run-test. Testing with the contract couplings is preferred. If this is not practical, the mechanical running test should be performed with couplings or simulators that have overhung moments within 10 percent of the contract couplings. Consideration should be given to using the contract coupling guards if supplier experience is problematic or coupling guard temperature calculations appear high. API 671 (2002) recommends maximum coupling guard temperatures of 71°C (160°F).

This is a test that has to be done on all compressor bodies regardless of them being identical, as the rotordynamic/vibration behavior is a function of the close clearances and assembly procedures including balancing.

Performance Tests

The PTC type tests can be of several classes, but the commonly applied tests are what are known as similitude tests, (ASME PTC-10, 1977, Type 2) where the compressor is driven by a shop driver (typically a steam turbine or a variable speed motor), and then aerodynamic similitude is derived between the test conditions and the site conditions. Details on shop testing are provided in Colby (2005). This is a test always specified (unless the compressor is identical to a pretested design). It is typically a test conducted on one compressor of a specific service used in a project. A typical indoor test bed is shown in Figure 80.



Figure 80. Indoor Test Stand.

For LNG compressors, the test will be a Type 2 test on a substitute gas. For performance testing a multistage compressor, volume reduction is a critical parameter. The test volume reduction must match the design volume reduction to be representative of the compressor performance on the specified gas. The head coefficient and efficiency of an impeller vary with the machine Mach number. Therefore, performance testing is required to be conducted at a Mach number relatively close to the design Mach number for each stage. The Reynolds number relates to the boundary layer and frictional losses of the medium in the flow path. Type 2 in-shop performance tests are almost always conducted at a lower Reynolds number than under specified conditions. The lower the Reynolds number, the greater the frictional losses and lower the efficiency. Most users do not allow the manufacturer to correct the test data for the differences in Reynolds number. Minimum straight lengths of inlet pipe are required for an accurate performance test, due to establishing laminar flow into the impeller and depending on the type of pressure, temperature, and flow devices. Equivalent arrangements can be studied by the compressor manufacturer and reduce the straight lengths. A minimum of five points, including surge and overload, shall be taken at normal speed. For variable-speed machines, additional points may be specified for alternate process operating conditions, and two alternate speeds to define the slope of the surge line and help the supplier in determining where, in which stage, surge initiated. If the three surge points are very close in flow coefficient, surge is initiated early in the compressor, the first or second stage. A greater separation between the three surge points would indicate surge to be initiated in one of the later stages.

Compressors with intermediate specified process pressures (i.e., side-loads) should have individual sectional head (pressure) tolerances as mutually agreed. Many compressor suppliers have a standard tolerance of + or - 2 percent on side-load head (pressure), so if the owner's process would prefer that a side-load pressure (temperature) be lower or higher than design, this direction should be furnished to the supplier upfront before they begin their detailed impeller design. Performance testing is especially important where there are two process services on the same train, since any speed change to improve the performance issue of one compressor service will also impact the performance of the other compressor service. Performance testing is also very important when there is a constant speed driver, since the compressor speed cannot be increased or decreased to make up for performance problems. This performance testing will be conducted typically using only one contract rotor.

For some large LNG compressors, in-shop performance testing is not possible, due to the size of available shop drivers. For this situation, performance testing in a string test configuration is employed. A disadvantage of this arrangement is that any performance problems discovered during the string test may cause schedule

problems with the compressor and the string test stand since the compressor may need to be sent back to the shop for impeller or other modifications. Performance testing in the string test configuration is also conducted in accordance with ASME There are two methods for determining the efficiency of the compressor. A torquemeter may be used to measure the input into the compressor. The second and most common method is the heat balance method where the work input is measured via the enthalpy rise from inlet to discharge.

String Tests

String tests are specified when there is concern about the overall performance in terms of the turbomachinery train under consideration. The test, if conducted under full load, full speed conditions, can replicate actual operating conditions. As the job driver and control system is used, many of the field startup problems can be minimized. The string test verifies that the entire job components are correctly designed and manufactured. The string test can be an expensive and time-consuming test, adding several weeks (10 to 15 weeks). Cook (1985) has covered the advantages of turbomachinery shop testing compared to field corrections that might have to be made if problems are discovered in the field.

This testing would include any applicable compressors, gas turbines, gear units, gas seals, and associated buffer gas systems, lube oil systems, and helper/starter motors. For large helper/starter motors and variable frequency drives, where two identical motors are purchased, consideration should be given to a back-to-back motor test at the motor supplier's facility. Once again, any problems discovered during the back-to-back test can be solved more efficiently and timely at the supplier's factory.

The primary advantage of the string test is the ability to perform a full load mechanical running test, and measure equipment vibration and bearing temperatures at full load and full speed. Machinery vibrations can be investigated in both the steady-state and transient conditions. Equipment train alignment capability can also be established. Besides the equipment, most auxiliaries are used during the string test including inlet filters/ducting, lube systems, couplings, coupling guards, control systems, electrical motor systems, exhaust systems, dry gas seal buffer systems, torquemeters, and vibration monitoring systems. Any fit up or assembly problems can be resolved during the string test, which is preferred over correcting these issues in remote field locations. Auxiliary and control systems can be tuned, thus minimizing these activities in the field. For extremely long compressor trains, with several compressor bodies, a gas turbine driver, and large VFD helper/starter motor, it can also be possible to measure torsional critical speeds during the string test if a torsograph connection has been provided.

A large scale string test arrangement for a Frame 7 driven LNG compressor is shown in Figure 81. A setup for a Frame 9 driven string test is shown in Figure 82.



Figure 81. Frame 7—LNG Compressor String Test Setup.



Figure 82. Frame 9—LNG Compressor String Test Setup.

INTEGRATION ASPECTS

All the turbomachinery components and the associated auxiliaries must work together in context of an integrated LNG plant. This means that aspects such as control philosophy, maintenance philosophy, and design of auxiliary systems are best considered during the design phases, involving the turbomachinery supplier.

Some of the key points that must be considered include:

- Integration of turbomachinery design into plant configuration—careful integration of process design with the machinery design. This should be done at an early stage of the design to ensure that a good aerodynamic and rotordynamic solution is derived taking all process cases into consideration.
- Lube oil systems/dry gas seal systems/air filtration and auxiliary systems. Careful attention to these systems is critically important to ensure good operational reliability.
- Distributed control systems (DCS) and control systems. These must be carefully integrated into the overall system.
- Antisurge valves and dynamic simulation—early definition of control schemes, control philosophy, compressor load sharing, startup issues, and transient issues must all be considered.
- Emission testing.
- Noise issues. These are best addressed at an early stage of the design.
- Condition monitoring. This is an important area for plant operational reliability. Modern condition monitoring tools should be considered to allow operators to have the best indication of operational health in terms of both aerothermal performance and vibration and other mechanical parameters.
- Fuel variability, gas turbine issues, and DLN/DLE considerations.
- Power plant integration.

SUMMARY

The dynamic growth of the LNG industry has resulted in the use of very complex and high technology turbomachinery for refrigeration service. This paper has provided an overview of the turbomachinery involved, covering both gas turbines and compressors. The propane compressor represents one of the most challenging designs.

From the perspective of compressor selection, design, and testing, close designer-user interaction and good communication are 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 nonoptimal 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.

APPENDIX A— LNG CONVERSIONS

LNG conversions can be found in Table A-1.

Table A-1. LNG Conversions.

| TO CONVERT FROM | \rightarrow | M^3 of LNG | Metric Ton of LNG | M^3 of Gas | Ft^3 of Gas | Milli Btu of Gas | Gigajou les | Kilowa Hour | Barrel Crude |
|--------------------------|---------------|-----------------|-------------------------|-----------------|------------------|------------------------|----------------|----------------|-----------------|
| MULTIPLY BY | | | | | | | | | |
| 1 M^3 of LNG | | 1 | 0.405 | 584 | 20,631 | 21 | 04 | 22.19 | 6173 |
| 1 Metric Ton of LNG | | 2.47 | 1 | 1379 | 48,690 | 52 | 54.8 | 15222 | 9.41 |
| 1 M ³ of Gas | | .00171 | .000725 | 1 | 35.3 | .036 | .038 | 10.54 | .006 |
| 1 Ft ³ of Gas | | .00005 | .00002 | .0283 | 1 | 00102 | .00108 | .299 | .00019 |
| 1 Million cu 048 | | .0192 | .0192 | 27.8 | 981 | 1 | 1.054 | 292.7 | .182 |
| 1 Gigajoule | | .045 | .018 | 26.3 | 930 | .095 | 1 | 277.5 | .173 |
| 1 Kilowatt Hour | | .000162 | .000065 | .0949 | 3.3 | 0.003415 | .0036 | 1 | .0062 |
| 1 Barrel Crude | | .261 | .106 | 153 | 5390 | 5.5 | 5.79 | 1.610 | |

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