



48TH TURBOMACHINERY & 35TH PUMP SYMPOSIA
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COMPRESSOR SELECTION FOR LNG LIQUEFACTION PLANTS

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ABSTRACT

The international LNG market is anticipated to expand in the coming years as world demand for LNG is expected to exceed the current supply as early as the mid 2020's. LNG trade volumes in 2017 were approximately 284 MTPA (Million Tons Per Annum), representing a 26 MT increase over 2016, and the fourth consecutive year of incremental growth. There is an increased interest in developing new LNG liquefaction plants, with many projects being proposed worldwide by both major oil and gas companies and independent developers to meet the forecasted growth rate.

As facility designs and configurations are being selected to match the gas supplies, sales contracts, and other logistical and economic constraints specific to the projects, many LNG train designs being proposed fall within the 4-6 MTPA range, with some proposed trains having an even larger capacity.

One of the most critical components of an LNG liquefaction facility are the refrigeration compressors which have a significant influence on overall plant performance and production efficiency. Refrigeration compressors for the train sizes referenced above are challenging to design due to large volume flows, high Mach numbers, low inlet temperatures, and complex sidestream flows. Compressor drivers for these facilities include both heavy duty industrial and aero-derivative gas turbines that range in size from 30 MW to 120 MW, and several recently proposed projects are considering the use of large electric motor drivers up to 75 MW.

This tutorial covers considerations encountered in the design, application, and installation of refrigeration compressors in large LNG facilities. The paper does not focus on any specific LNG process technology but addresses turbomachinery design and application aspects that are common to most processes. Topics cover key technical design issues, refrigeration cycles, compressor designs and configurations and risks involved in compressor selection and aeromechanical design. Practical design compromises that must be made to obtain a robust compressor solution will be covered. While the focus of this tutorial is compressor selection, the topic of selecting and matching drivers to refrigeration compressors will also be covered.

INTRODUCTION

The international Liquefied Natural Gas (LNG) trade is expanding at a rapid pace with LNG accounting for approximately 9-10% of global gas supply. Table 1 provides some salient parameters of the current LNG market (Data from IGU World LNG Report, 2017 https://www.igu.org/sites/default/files/103419-World_IGU_Report_no%20crops.pdf).

Table 1. Current LNG Market Data.

Global LNG Trade in 2016	258 MT
Global Nominal Liquefaction Capacity	340 MTPA
LNG Shipping fleet	439 ships
Share of LNG in Global Gas Supply	9.8%

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 $1/600^{\text{th}}$ of its standard 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 BTUs of gas (54.8 GJ). The refrigeration is accomplished using several commercially available processes that typically involve the successive chilling of the feed gas to lower and lower temperatures using mechanical refrigeration systems utilizing large refrigeration compressors. The compressors are typically driven by gas turbines ranging from 30-120 MW. Recently, motor driven LNG facilities are also being developed. Figure 1 shows the growth in LNG train size over the past five decades. The size range of 4-6 MTPA is currently popular for large trains and most of the current LNG facilities are of this size range but as pointed out by Caswell



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(2019), there is much interest in smaller scale LNG in the region of 1-2 MTPA.

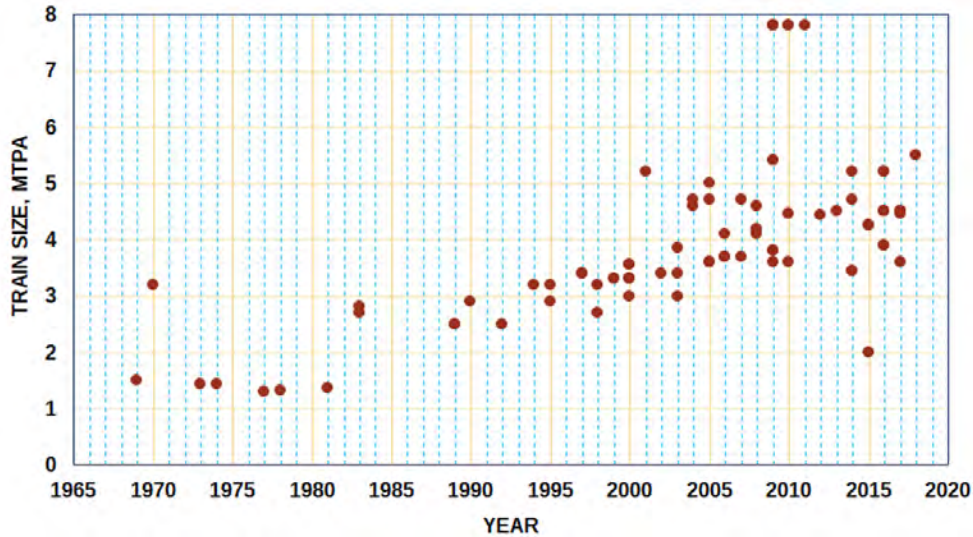


Figure 1. Train Size Growth of LNG Liquefaction Plants in the Last Five Decades.

In the LNG market “Take or Pay” contracts currently dominate the marketplace (approximately 70%) but there is an increasing number of short term contracts in play and also use of the spot market. In the past developers have typically been IOCs (international oil companies) or consortia of IOCs. Several smaller developers are currently developing LNG liquefaction projects. The LNG market has typically been a very conservative market, with novelty (new drivers, new compressor designs etc.) not readily being utilized. This conservative approach has been used because of the risk averseness of investors, LNG purchasers, LNG institutions and end users. Recently, due to market pressures and the pursuit of economy of scale more innovative solutions and technology have been deployed.

Some general trends in LNG liquefaction include:

- Desire for higher plant efficiency and higher plant production efficiency
- Need for cost reduction (\$/ton)
- Focus on greenhouse gas emissions reduction and reduction of other pollutants
- Focus on time to market (e.g. delivery of turbomachinery) which is typically on the project critical path along with the LNG storage tank
- Desire to minimize site work (modularization) especially in challenging remote locations or regions where labor is constrained or expensive
- Need for smooth and flawless startup implying the need for special planning prior to startup

Some useful rules of thumb are provided below to provide a high level understanding of LNG liquefaction parameters and should be applied with care because they may vary on a project by project basis:

- Approximately 1.2 TCF (trillion cubic feet) of reserves are required per MTPA of LNG for expected 20 year life
- Enthalpy reduction to liquefy methane is approximately 370 BTU/lbm (861 kJ/kg)
- Specific power to liquefy typical feed gas composition is approximately 17.4 hp per ton/day (13 kW per ton/day)
- Impact of 1 % power difference results in approximately 0.8 to 1% of LNG production
- A 1% increase in refrigerant condenser UA (heat transfer coefficient) results in an approximate 0.3% increase in LNG



production

- Typical inlet feed gas pressure to an LNG plant is 60-70 bar. Increasing the inlet pressure results in more LNG being produced. A rough rule of thumb is that a 20 bar increase in feed pressure results in a 2% increase in LNG.
- Boil-off gas losses (BOG) are typically 2-3%
- Fuel gas consumption is 6-10% of feed and is a strong function of the driver efficiency
- CO₂ emissions are approximately 0.2 tons per ton of LNG
- Typical LNG ship loading rate is 353,000 to 424,000 ft³/hr (10,000 to 12,000 m³/hr)

A generic block diagram overview of an LNG liquefaction plant is shown in Figure 2. The facility can be broken up into a gas treatment section, a liquefaction section and a LNG storage and loading system. The liquefaction block includes the refrigeration compressors to chill and liquify the natural gas. These large and complex compressors are the focus of this paper.

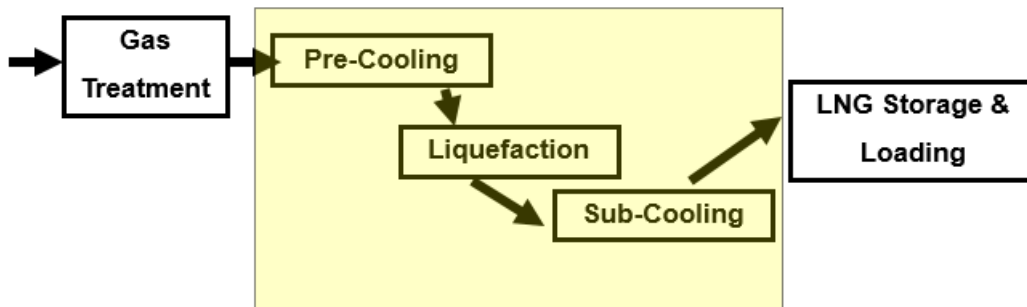


Figure 2. Block Diagram of a LNG Liquefaction Facility.

The gas treatment block normally includes:

- Acid gas (CO₂ and H₂S) removal is typically down to 50 ppmv of CO₂ and a max of 4 ppmv of H₂S. Acid gas may result in corrosion and may freeze, creating solids in the cryogenic section of the liquefaction process
- Dehydration: water can freeze in the cryogenic process and hence has to be removed down to 1 ppmw
- Mercury Removal: Mercury can cause severe corrosion, especially in brazed aluminum heat exchangers, and hence has to be reduced to 0.01 microgram/Nm³

LNG storage and loading consists of the LNG Tank, and loading arms which facilitate loading of LNG onto LNG Tankers.

Specific Power, Auto Consumption and LNG Plant Thermal Efficiency

Some common terms used in natural gas liquefaction plants are specific power, auto consumption and thermal efficiency. The specific power is a measure of process efficiency and is defined as:

$$\text{specific power} = \frac{\text{power consumed for liquefaction}}{\text{LNG production rate}} \quad (1)$$

The specific power is a function of compressor efficiency and also process conditions such as the inlet pressure to the liquefaction section, pressure losses, and temperature approaches (effectiveness) of heat exchangers.

Auto consumption is the percent of feed fuel which does not end up included in the LNG or the natural gas liquids (NGLs) produced. It is essentially the percentage of the feed energy that is “used” (hence the term auto consumption) to provide thermal or mechanical



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energy for the process.

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

- The thermal efficiency of the refrigeration turbines and compressors (major contributor)
- The efficiency of power generation turbines if power generation is “inside the fence”
- Other fuel use for process heating, such as hot oil heaters, etc.

The choice of driver type will impact auto consumption. The traditional E class industrial gas turbines used in liquefaction service (rated firing temperatures near 2055F) have efficiencies around 32 percent. Aero-derivative engines commonly used as liquefaction compressor drivers have thermal efficiencies of approximately 40 percent. It should be noted that some modern, large two-shaft heavy duty engines have been introduced with thermal efficiencies approaching 38 percent which are close to aero-derivative engines. A large hybrid engine in the 100MW class has even higher thermal efficiency due to its intercooled design. This engine has recently been selected for a large scale LNG Facility in Western Canada.

The thermal efficiency of an LNG facility is defined as:

$$\eta_{plant\ thermal} = \frac{LNG\ Heat\ Content + NGL\ Heat\ Content}{Feed\ Gas\ Heat\ Content} \quad (2)$$

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

$$Auto\ Consumption = 1 - \eta_{plant\ thermal} \quad (3)$$

Thermal efficiency considerations for LNG plants is provided in Yates (2002).

LNG LIQUEFACTION PROCESSES

Several liquefaction technologies are available for application in larger LNG projects and include:

- Air products and Chemicals (APCI) C3-MR Process
- APCI AP-X Process
- Conoco Phillips Optimized Cascade Process
- Shell Dual Mixed Refrigerant Process
- Linde Dual Mixed Refrigerant Process

Small to mid-range capacity processes (base load, peak shavers, and transportation fuels) utilize:

- Black & Veatch’s PRICO Process
- Single Mixed Refrigerant (SMR) processes
- Gas phase, Brayton refrigeration cycles (nitrogen cycle)
- LNG Limited Optimized Single Mixed Refrigerant (OSMR) Process

In almost all processes, the feed gas liquefaction duty is done in two or three steps. Precooling is predominantly done by propane refrigerant loops, or in some cases, using mixed refrigerants. Liquefaction is done by mixed refrigerants or in some processes, utilizing ethylene refrigeration. An overview of LNG processes is found in Shukri, (2004).



C3-MR Process Overview

The APCI C3-MR process uses propane as a pre-cooling medium and a mixed refrigerant (a mix of nitrogen, methane, ethane and propane) as the liquefaction medium. This process has been historically the most applied in industry. While a wide range of drivers can be used, the GE-7EA industrial gas turbine is the most predominant. In this process, mixed refrigerant is partially condensed against air and four stages of propane cooling. The vapor and liquid refrigerant fractions are subsequently auto-cooled through expansion, in order to approach matching of the feed gas cooling curves in a spiral-wound main cryogenic heat exchanger (MCHE). Natural gas is liquefied in this heat exchanger. A four-section propane refrigeration cycle provides the pre-cooling for the MR and the feed natural gas. Quite commonly, an end-flash compressor is used at the end of the liquefaction process which provides pressure boosting of fuel gas to the gas turbines. An overview of the APCI C3-MR cycle is shown in Figure 3. Primary cooling is provided by either water or air coolers.

Propane-precooled Mixed Refrigerant Process (C3MR)
 (Licensor: Air Products and Chemicals Inc.)

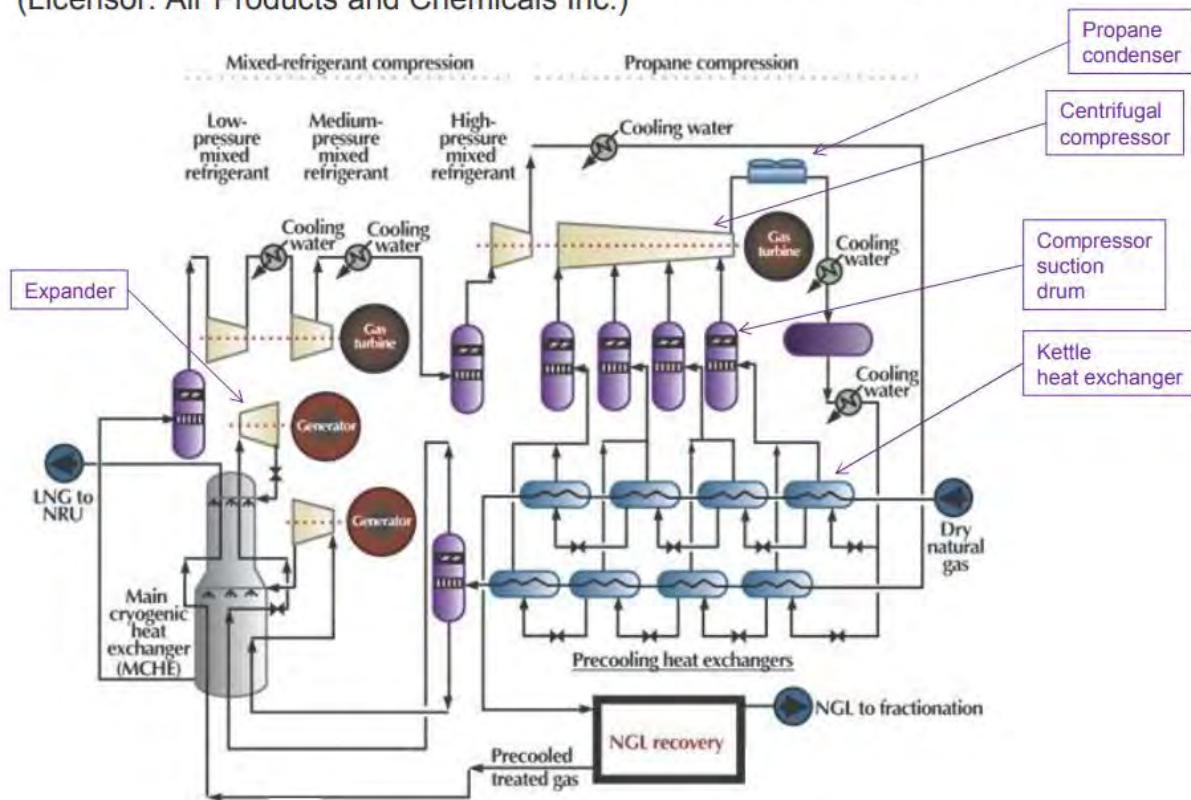


Figure 3. APCI C3-MR Liquefaction Process.

Conoco-Phillips Optimized Cascade Overview

The Optimized Cascade process is a multiple, single component refrigerant system wherein the lowest boiling temperature stage of each refrigerant is used in turn to condense the next refrigerant. The consecutive cooling steps are done by three section propane (pre-cooling) and subsequently by two section ethylene and three section methane compressors. Core-in-kettle type heat exchangers and plate-fin heat exchangers are used for cooling of the feed natural gas. The use of these exchanger types accommodates low temperature approaches.



The inlet pressure of the lowest pressure section of the methane compressor is maintained above atmospheric conditions and runs flashing into the LNG tank. Hence boil-off gas compression forms part of the methane cycle. In order to prevent the accumulation of nitrogen in the methane cycle system, gas may be extracted from the methane compressor process loop and utilized for gas turbine fuel. A schematic of the Conoco-Phillips Optimized Cascade Cycle is provided in Figure 4.

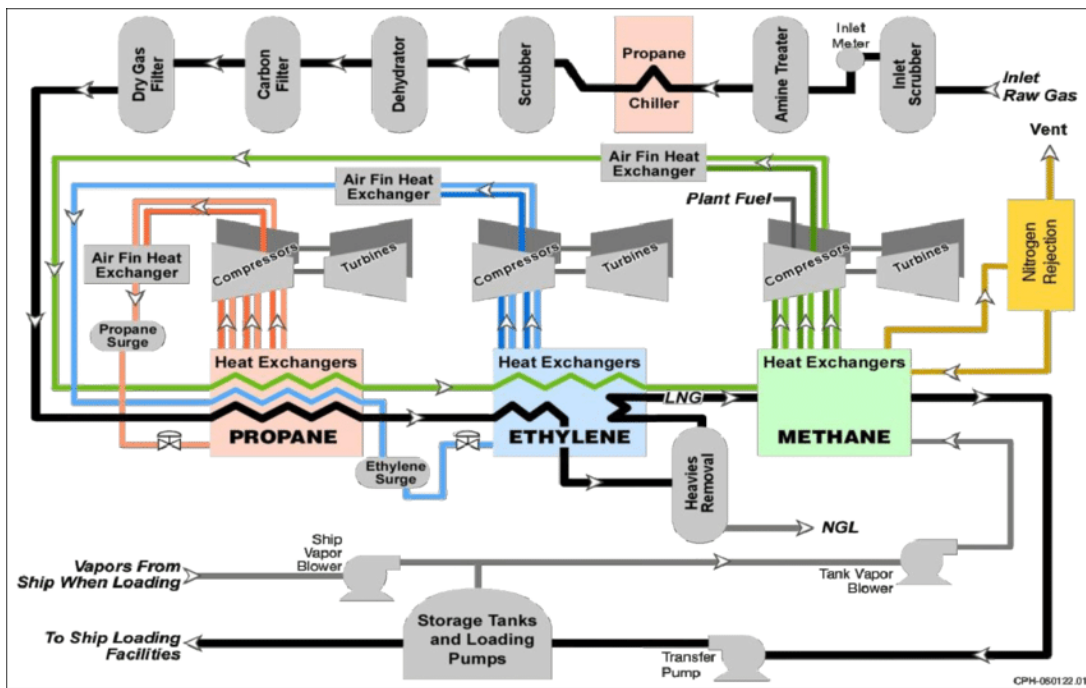


Figure 4. CoP Optimized Cascade Liquefaction Process.

Dual Mixed Refrigerant Cycle (DMR) Overview

This liquefaction cycle is like the C3-MR cycle except that mixed refrigerant is also used for the precooling cycle. The Dual Mixed Refrigerant process uses a mixture of methane, ethane, propane and butane as precooling medium. The compressed mixture is condensed against air and subsequently auto-cooled and expanded to provide refrigeration. The expansion can be performed at one, two or three pressure levels. DMR liquefaction processes are offered by more than a single licensor.

A typical DMR Cycle is shown in Figure 5.

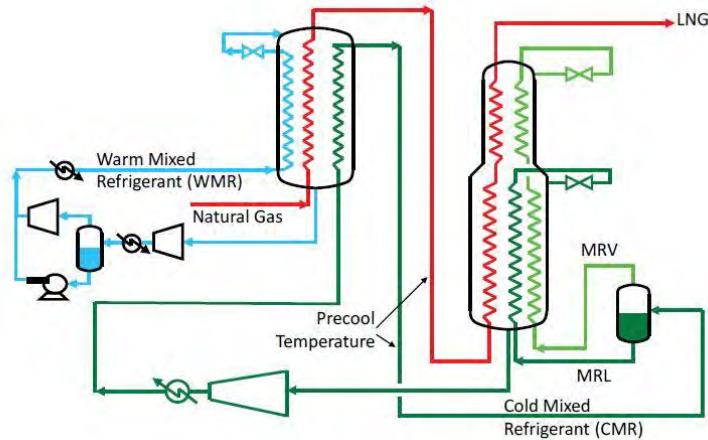


Figure 5. Typical DMR Liquefaction Process.

VAPOR COMPRESSION CYCLES FOR LNG LIQUEFACTION

A majority of existing commercial natural gas liquefaction processes are based upon the use of vapor compression refrigeration cycles. This fundamental thermodynamic cycle is the same as used in commercial and residential air conditioning and refrigeration services. It may be characterized in four separate steps. First, a vapor at or near saturated vapor conditions is compressed to an elevated pressure. This higher temperature (due to the heat of compression) gas is then cooled, condensed, and sometimes sub-cooled to the liquid phase through heat exchange to a lower temperature heat sink (typically ambient air). The condensed liquid is then expanded to a lower pressure and temperature liquid through expansion, such as a throttling valve or other expansion device. Some amount of flashing (partial vaporization) will typically occur during this expansion step. Finally, the lower pressure liquid is used to extract heat from another source, evaporating the liquid to a saturated vapor at the existing lower pressure. A simple vapor compression refrigeration cycle schematic is provided in Figure 6 below:

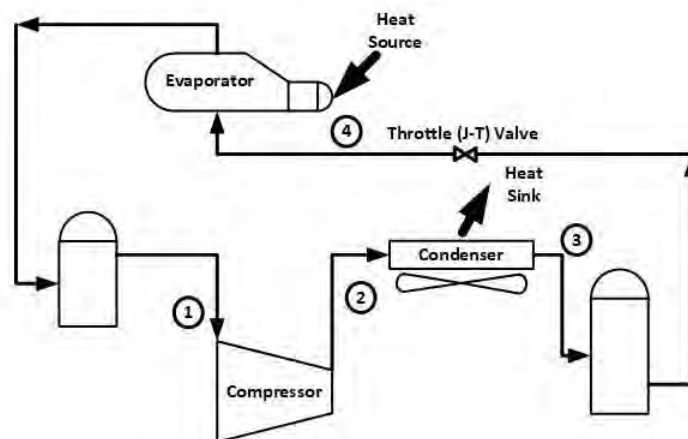


Figure 6. Simple Vapor Compression Refrigeration Cycle Schematic.



The refrigeration duty afforded by the cycle represented above is a function of the closed cycle mass flow rate and the latent heat of vaporization of the liquid phase of the refrigerant at the compressor suction pressure. Compression power required is related to the ratio of existing condensing pressure to the evaporation pressure, the closed cycle mass flow rate and the compressor efficiency. The required capacity of the condensing heat exchangers is a function of the compressor discharge temperature, condensing temperature and latent heat, and the closed cycle mass flow rate. The condensing heat exchanger, typically rejecting heat to ambient air (alternatively a water cooling system), must be sized to first remove the sensible superheat from compressor discharge to the condensing temperature, the latent heat to condense the refrigerant, and any sub-cooling that may be possible. Figure 7 presents a pressure-enthalpy (P-H) diagram of a simple vapor compression refrigeration cycle.

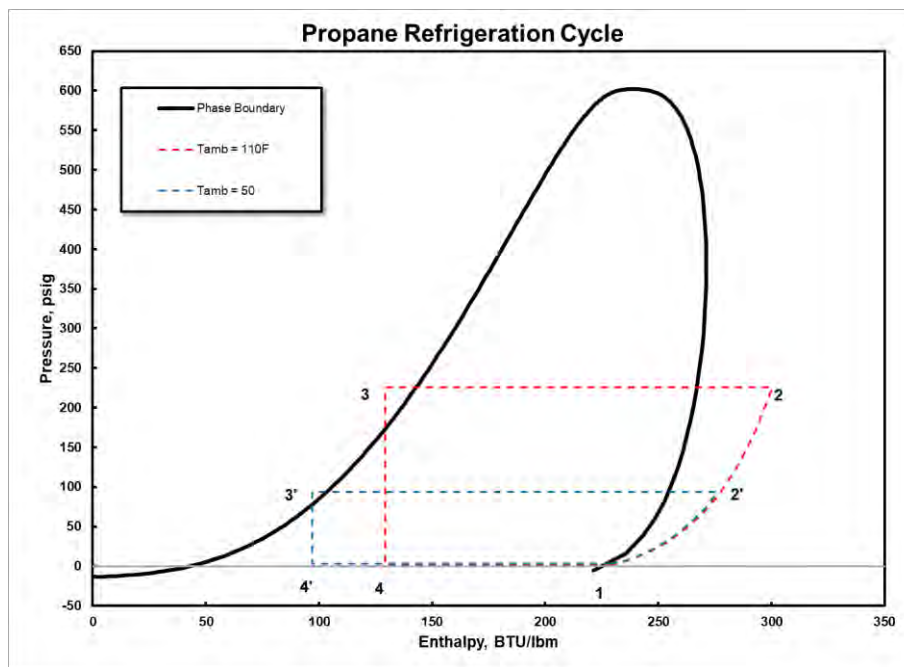


Figure 7. Propane Refrigeration Cycle Pressure-Enthalpy Diagram.

This diagram presents two distinct closed cycle paths, one as a red dashed characteristic and the other defined with a blue dashed characteristic. The black curve defines the phase envelope for the propane refrigerant with a mixed liquid-vapor phase existing within the bounds of the phase envelope. Numbers adjacent to portions of each path correspond to points identified on the cycle schematic presented in Figure 6. The primed numbers are specific to the blue dashed process path. The red dashed path approximates a vapor compression process using propane as a refrigerant with condensation occurring for an ambient air temperature of 110°F (43°C) and an approach temperature of 10°F (5.5°C).

The process path from point 1 to 2 is based upon a polytropic compression path with a constant value of polytropic efficiency. The discharge pressure is set by the condensing pressure of propane at existing ambient conditions. The path from point 2 to point 3 represents constant pressure heat exchange occurring within the condenser and associated piping and equipment. It is recognized that some amount of pressure drop will be observed in a real system, but the error incurred will be relatively small. Heat exchange to the right side of the phase envelope is characterized as a sensible heat change with a reduction of temperature of the vapor phase of the refrigerant. Once the boundary of the phase envelope (saturated vapor) is reached, additional heat exchange results in partial condensation of vapor to the liquid phase. This continues at a constant temperature until the left side of the phase envelope (saturated liquid) is reached, where the refrigerant exists totally in the liquid phase. Further heat exchange results in sensible heat sub-cooling of the liquid with additional reduction of the temperature. While it may be possible to sub-cool the liquid with heat exchange between the refrigerant and ambient



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conditions, another method to provide sub-cooling involves additional heat exchange with the refrigerant flow downstream of the expansion device.

An expansion process is the next step in the cycle which is represented by the path from point 3 to point 4. The path included in Figure 7 represents an isenthalpic expansion process which is achieved by throttling the flow from the condensing pressure to a lower pressure through a restriction such as a valve. It is evident that the path progresses into the phase envelope. At the endpoint of the process, the refrigerant exists as a combination of both liquid and vapor phases with the percentage of vapor related to the fractional length of that point from the saturated liquid boundary to the total length of the latent heat (saturated liquid to saturated vapor) line. An alternative process that is employed in some select cycles replaces the throttling valve with an expansion turbine that extracts energy from the higher pressure stream approaching an isentropic path. This moves the terminal point on the phase envelope closer to the saturated liquid line resulting in a lower fraction of the total stream being flashed to the vapor phase and a greater amount remaining as a liquid to provide additional refrigeration duty. As noted previously, this vapor-liquid stream downstream of the expansion device may also be cross exchanged with itself upstream of the expansion to further sub-cool the flow prior to expansion. Although this results in some increased vaporization of the downstream flow, it is offset by reduced flashing to vapor during the pressure reduction.

The final step of the closed vapor compression cycle involves heat input into the refrigerant and evaporation of the liquid portion of the flow back to the vapor phase. The combined vapor-liquid flow created through the expansion device is directed to the evaporator where additional liquid is changed to the vapor phase. The closed cycle flow rate is set by the rate of vapor generation unless it is limited by the condensing capacity of the associated cycle equipment or compression power available. The evaporator operating pressure is normally maintained at a slight positive pressure during steady state conditions to avoid potential ingress of air in the case of containment loss. Notwithstanding the safety concerns if the refrigerant is hydrocarbon based, the introduction of air into the closed loop may result in the presence of non-condensable components that reduce the effectiveness of the refrigeration cycle. It is recognized that some pressure loss will occur in this equipment, although this is considered to be minimal and of limited impact to the overall results. The vapor phase flowing into the evaporator along with additional vapor generated from heat exchange is then directed to the suction of the compressor where the cycle repeats.

Existing condenser heat sink (typically ambient air) conditions significantly affect the cycle operating conditions. The vapor pressure of the refrigerant is directly related to the temperature. Reference to Figure 7 illustrates the impact of ambient temperature on an air cooled condenser cycle with a 10°F (5.5°C) approach temperature for two ambient temperatures of 110°F (43°C) and 50°F (10°C). The 110°F (43°C) based cycle conditions are illustrated with the red dashed characteristics while the 50°F (10°C) cycle is provided as a blue dashed characteristic. It is obvious that as the ambient temperature is reduced, the operating pressure of the condenser is also reduced. This may result in increased specific cooling duty required of the condenser along with increased refrigeration capacity due to reduced expansion flashing fraction and additional refrigerant evaporation latent heat availability, as illustrated. The lower condensing pressure coincident with the lower ambient temperature also results in a reduced pressure ratio and specific power required from the compressor. These variations in required duties and operating conditions demonstrate the challenges that may accompany the sizing and selection of heat exchange and compression equipment that become more prevalent as the ambient temperature extremes existing for a natural gas liquefaction plant location deviate. Impacts of significant variations in ambient temperature may also influence available power from some compression drivers that will need to be evaluated, as well.

The selection of an appropriate refrigerant for a specific application is based upon a number of factors. A primary consideration is the ability to condense the vapor phase refrigerant against whatever heat sink is employed. In the case of ambient air, this involves examining the range of temperatures expected during seasonal and over historical periods. A valuable graphical tool that may be utilized is a plot of different refrigerant vapor pressures versus temperatures. Such a plot is provided in Figure 8. Several commonly used industrial hydrocarbon based refrigerants are included in this figure along with nitrogen and R-134a (tetrafluoroethane), a current commercial and residential refrigeration and air conditioning refrigerant. Vertical dashed and dashed-dot black lines are also included on this graph, denoting selected temperatures. The intersection of one of these lines of constant temperature and a vapor pressure line determines the condensing pressure of that refrigerant at that temperature. Vapor pressure curves that do not intersect with these lines demonstrates that the refrigerant will not condense against that temperature at any pressure covered by the graph, or possibly not at all.



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A horizontal solid black line denotes ambient (0 psig) pressure. Intersection of this solid line and a selected vapor pressure curve denotes the evaporating temperature of that refrigerant at ambient pressure conditions. Recalling that it is established good practice to maintain the minimum steady state operating pressure of the evaporator slightly above ambient to avoid leakage of air into the closed refrigeration loop. Accordingly, it is possible to determine condensing pressure, evaporation temperature, and required compression pressure ratio with consideration also included for expected heat exchanger approach temperatures from this simple diagram.

Other refrigerant selection factors include the relative cost and availability of each refrigerant, latent heats and flash vaporization rates at condensing and evaporation conditions, and, of course, safety and handling concerns. For example, while R-134a might be a reasonable alternative to the commonly used propane for natural gas liquefaction, it's significant higher cost and relative lower availability (propane can be generated on site from most feed gas compositions), offsets it's increased safety in handling.

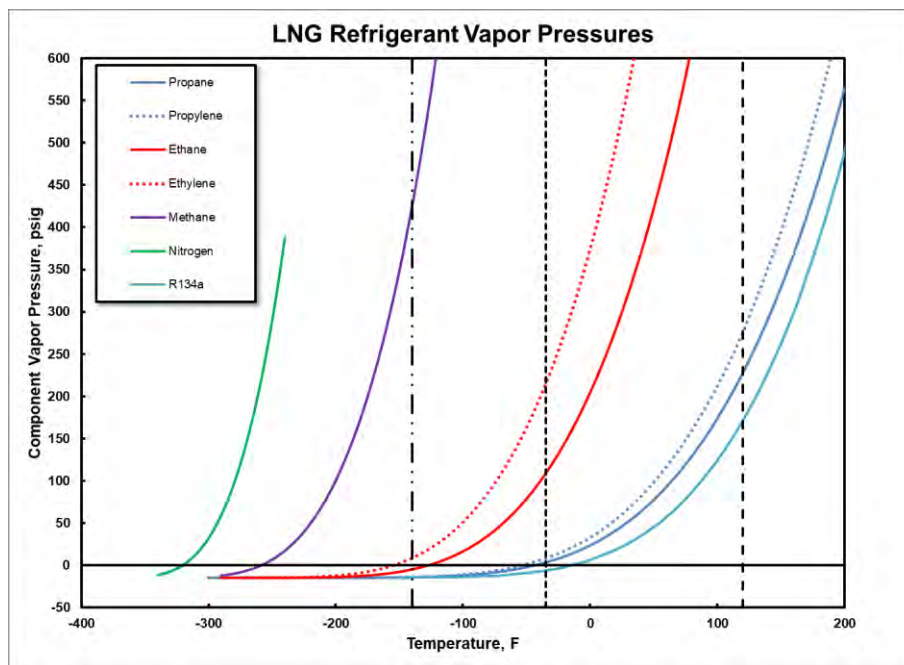


Figure 8. Selected Refrigerant Vapor Pressure Curves.

Another significant observation provided by Figure 8 is associated with the minimum evaporating temperature available for a given refrigerant. This minimum temperature that is a characteristic of individual refrigerants at ambient pressure conditions represents the minimum temperature, along with allowances for slight positive pressure and temperature differential (approach temperature) allowed for heat transfer, attainable by the refrigeration heat source. In the case of propane the minimum temperature is approximately -40°F (-40°C), so any source that is refrigerated by a propane loop will not be able to be cooled below this temperature plus any aforementioned allowances. This creates a limitation for each type of refrigerant. Additional cooling capacity to a lower temperature may only be achieved by cascading separate vapor cycle loops, using the evaporation temperature of a higher temperature range loop to act as the condensation source of a lower temperature range refrigerant loop. A schematic of cascaded loops is illustrated in Figure 9.

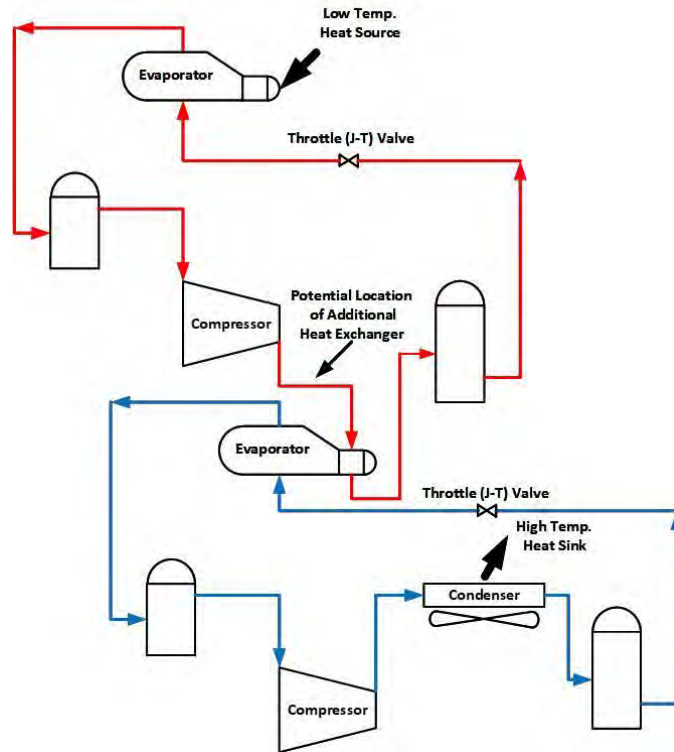


Figure 9. Cascaded Vapor Compression Refrigeration Loops.

Two or more refrigeration loops arranged in a cascaded configuration afford a much wider temperature range between the highest temperature heat rejection condenser and the lowest temperature evaporator providing the ultimate duty required of the refrigeration system. It is obvious that the relative flow rates, cooling duties (latent heats) and pressure ranges are established by the properties of the individual refrigerants. Should the discharge temperature of any compressor within cascaded loops be above ambient, additional heat rejection to the environment is possible by adding an exchanger as noted in the diagram. If cascaded propane-ethylene loops are utilized for a given refrigeration service, since propane represents the higher temperature range refrigerant loop, it will act as the primary refrigerant loop that rejects heat to the environment. Assuming that this heat rejection is through air-cooled heat exchangers, the condensing pressure is set by the existing ambient temperature plus some approach temperature. Further, assuming that the evaporator operating pressure and resulting temperature are then cross exchanged with the ethylene loop, the ethylene condensing temperature and pressure are established. These can be approximated by the two dashed vertical lines included in Figure 8. The third, dot-dashed vertical line represents the temperature that could result in the ethylene evaporator. This significant reduction in ultimate cooling source temperature demonstrates the substantial value in cascading vapor compression refrigerant loops to achieve refrigeration to relatively low temperatures.

The liquefaction of natural gas requires a significant amount of heat extraction and accompanying temperature change from initial conditions. This process normally occurs at an elevated natural gas pressure, where sensible heat, condensation and some amount of sub-cooling takes place prior to an expansion to near atmospheric pressure and a temperature of approximately -260°F (-160°C). Figure 10 provides relevant information of a “typical” 20 molecular weight natural gas (feed gas) mixture initially at 600 psig (41.4 barg) and 100°F (37.8°C) that is cooled to a state prior to expansion where it can be stored and transported.

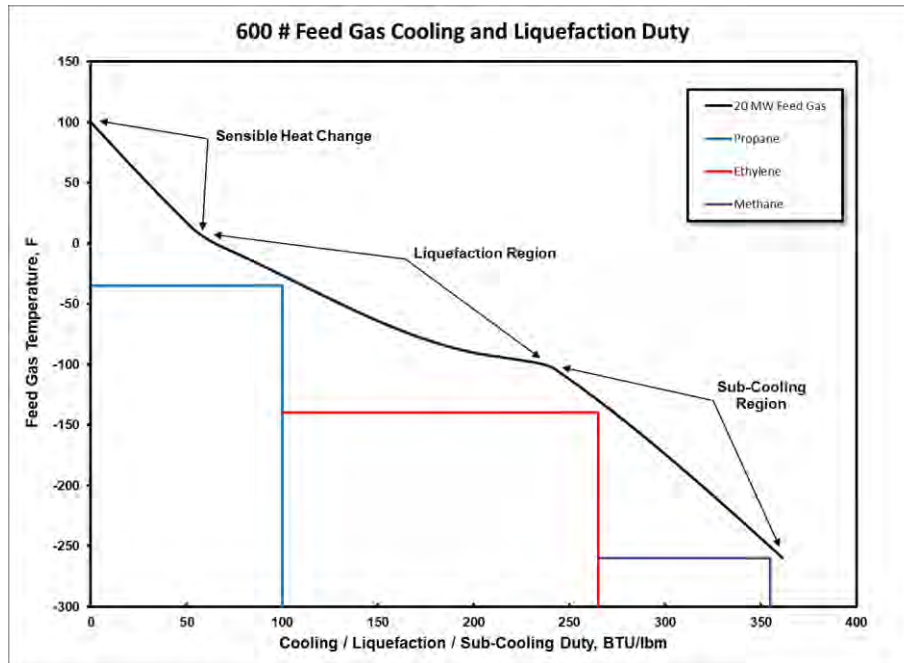


Figure 10. Typical Feed Gas Cooling and Liquefaction Curve.

Prior to entering the liquefaction portion of an LNG facility, the feed gas is processed to remove any acid gas (carbon dioxide and hydrogen sulfide) components and dehydration. Mercury removal is also accomplished, if necessary. As the gas proceeds to the pre-cooling phase, sensible heat is removed from the feed gas resulting in a relatively linear relation between cooling duty and temperature change. Once the dew point of some of the heavier hydrocarbon components are reached, the cooling curve changes slope and assumes a more curved characteristic. The curve shape is a function of the relative amounts of components and their respective latent heats. Temperature of the feed gas continues to decrease as more heat is removed, but at a reduced rate. Once all of the hydrocarbon mixture components are condensed, the cooling curve assumes a near linear relationship between cooling duty and temperature again. It is possible that some non-condensable components remain (e.g. nitrogen), but those are generally removed before or at the time of the expansion flash to storage tank conditions. It is beneficial to sub-cool to the extent possible to minimize the amount of feed gas evaporated during the expansion flash. The flashed LNG and any non-condensables are either recycled to the inlet of the plant or utilized as fuel gas.

Single component refrigerant evaporation curves are superimposed on the feed gas cooling curve in Figure 10. They represent a possible liquefaction process including the three identified refrigerants. The three evaporation curves appear as horizontal lines of constant temperature for constant respective evaporator operating pressures. A comparison of the feed gas cooling curve and the refrigerant evaporation curves shows that a varying temperature differential exists between each of the two streams. A higher average temperature differential, while resulting in a smaller heat exchanger, represents a more substantial requirement of power to compress and condense all of the refrigerant to the common evaporator pressure. Ideally, a constant, minimum temperature differential between the feed gas and refrigerants would exist to minimize the refrigerant compression power.

A reduction in required compression power can be achieved by providing one or more of the distinct refrigerants in multiple evaporators operating at different pressures. Higher evaporator operating pressure results in a higher equivalent vapor temperature. Thus, by exchanging a higher pressure refrigerant with the higher temperature portion of the feed gas cooling curve, a reduced average temperature differential exists. A schematic of a two section refrigerant loop is illustrated in Figure 11.

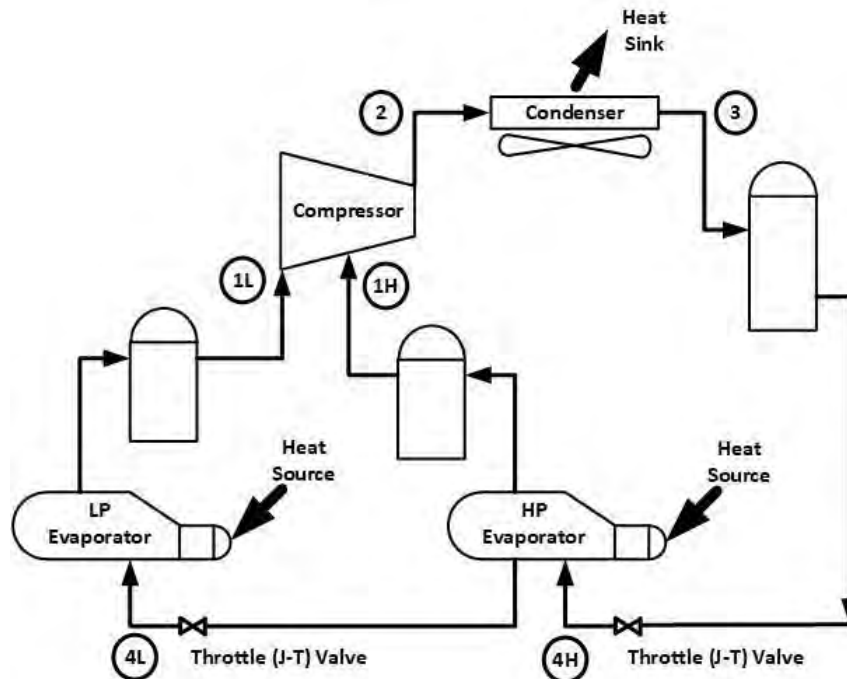


Figure 11. Two Section Vapor Compression Refrigeration Cycle.

This diagram shows that the high pressure, liquid refrigerant flow from the condenser is directed to a high pressure evaporator which has an operating pressure somewhere between the condenser pressure and the low pressure evaporator. After the high pressure condenser flow is expanded through an expansion device, it enters the high pressure evaporator where it transfers heat and is further partially evaporated. A portion of the flow is extracted and directed through another expansion device where it is throttled to a lower pressure and flows into a low pressure evaporator. The relative amount of refrigerant vaporized in each of the evaporators is a function of the combined cooling duty and the total mass flow through the refrigeration loop. The refrigerant mass flow rate in the high pressure portion of the loop is subject to a lower pressure ratio, requiring a lower specific power than the remainder of the total flow rate that is compressed over the entire pressure ratio of the loop.

A P-H diagram for a two-section, propane refrigeration loop is presented in Figure 12. The numbers included on the process path correlate with the circled numbers included on the cycle schematic of Figure 11. Starting with the refrigerant condensing portion of the cycle (station 2 to 3), the entire cycle mass flow rate is condensed into the liquid phase at vapor pressure conditions set by the heat sink temperature with some allowance for heat exchanger temperature approach. The condensed refrigerant then flows to the high pressure evaporator (station 4H) through an expansion device operating at a pressure between the low pressure evaporator pressure and condenser pressure. During this expansion process, some of the liquid flow is flashed back into the vapor phase. The isenthalpic path displayed in Figure 12 demonstrates this partial vaporization with the linear portion of the constant pressure path relative to the total length between saturated liquid and saturated vapor states (quality) representing the amount of liquid fraction existing. Some of the liquid phase refrigerant entering the high pressure evaporator is further vaporized with the existing cooling duty. This combined vapor flow is then introduced to the second section inlet of the compressor (station 1H). The remainder of the liquid flow rate is further expanded through a second expansion device to the lowest operating pressure of the loop in the low pressure separator. There will be additional flashing of this liquid stream (station 4L) through the expansion device prior to entering the low pressure evaporator that will be related to the quality difference between states 4H and 4L in Figure 12. Again, an isenthalpic expansion process is assumed in this diagram, but other expansion devices may result in a path that is between an isenthalpic and isentropic expansion. Further vaporization is generated with the cooling duty associated with the low pressure evaporator. This combined vapor flow is directed into the low pressure section inlet



of the compressor (station 1L). The low pressure vapor flow is compressed to the high pressure evaporator operating pressure via a constant efficiency polytropic process to station 1H. Here it is mixed with the high pressure evaporator vapor flow. The mixed vapor flow results in a mixture enthalpy that is dependent upon relative mass flows and enthalpies of the two streams. Generally, the resulting mixed stream enthalpy falls between the first stage discharge enthalpy and second stage saturated vapor as shown. The combined mass flow rate is then compressed in the high pressure compressor section to the condenser operating pressure where the preceding described cycle starts once more. The respective section mass flow rates and their relative proportion can change due to variations in cooling duties and ambient conditions.

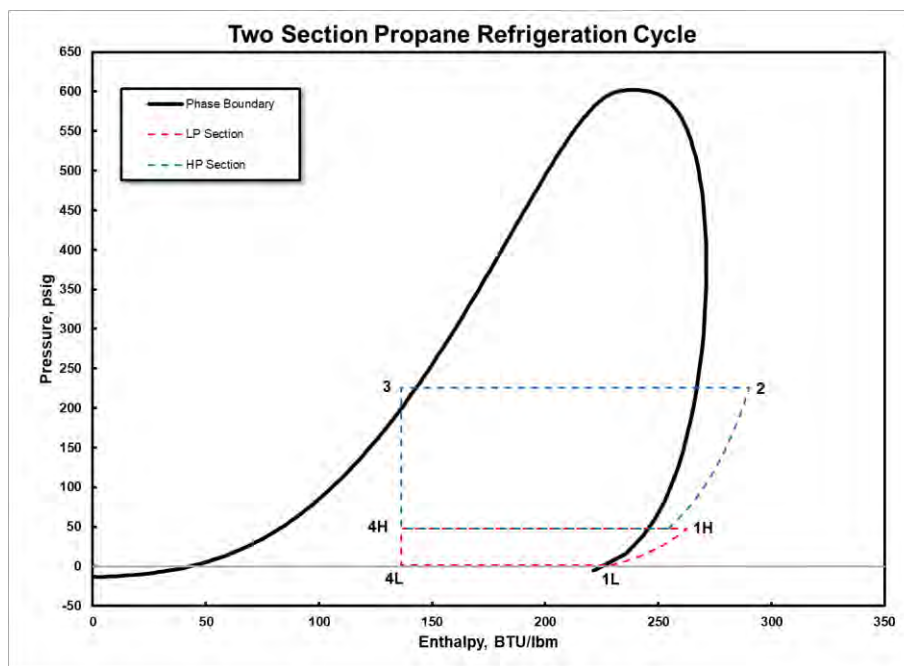


Figure 12. Two-Section Propane Refrigeration Cycle Pressure-Enthalpy Diagram.

Figure 13 provides a section of the feed gas cooling curve presented in Figure 10 that is associated with the propane cooling portion of the overall curve. In this case, instead of a single stage propane refrigeration cycle, the approximated operating curve of an arbitrary two section propane refrigeration loop is inserted. The resulting refrigerant curves demonstrate a stepped characteristic that represents a lower average temperature differential between the feed gas and the refrigerant. This lower average differential temperature results in a lower refrigeration power requirement for the same cooling duty. It is relatively common practice to have two to four pressure stages and accompanying compressor sections in industrial refrigeration loops. Ideally, an infinite number of stages would result in a constant temperature differential between the feed gas and refrigerant, but this is not practical.



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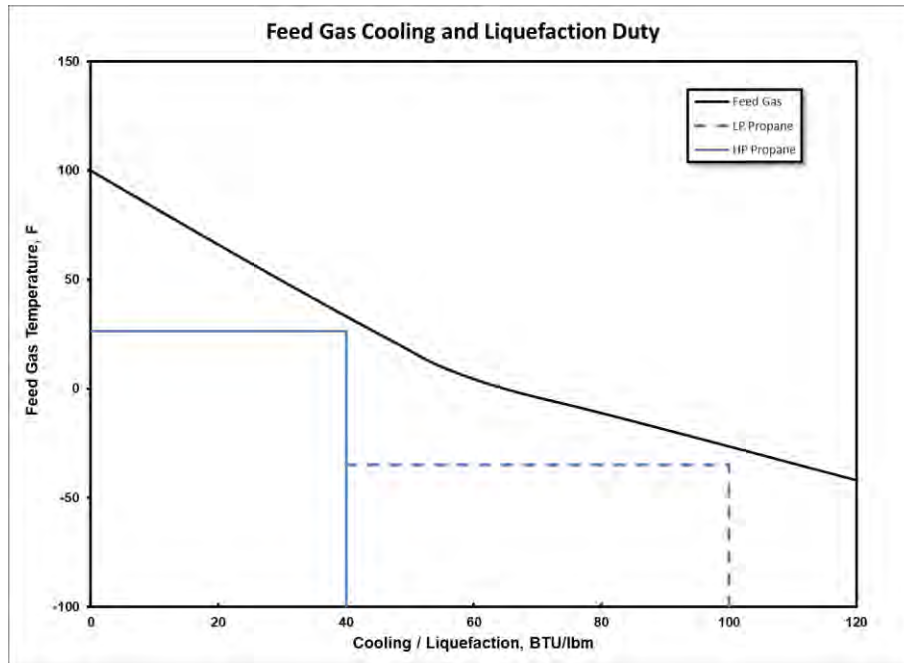


Figure 13. Typical Feed Gas Cooling Curve with Two-Section Propane Refrigeration Loop.

The previous descriptions and analyses of variations in vapor compression refrigeration cycles have been limited to single component refrigerants. It is also possible and common practice to replace single component cycles with refrigerants composed of two or more components. A primary difference in utilizing mixed refrigerants is the fact that condensation and evaporation of the refrigerant takes place at a varying temperature during constant pressure heat exchange. This allows development of mixed composition refrigerants whose evaporation characteristics more closely follow feed gas cooling and liquefaction curves with a smaller, more consistent temperature differential. It also normally results in lower specific power requirements and more efficient heat exchange for a defined refrigeration duty.

Virtually all large scale existing natural gas liquefaction processes include one or more of these fundamental vapor compression cycle configurations and the use of multiple single component and mixed refrigerant compositions. This includes the predominantly utilized Air Products and Chemicals C3-MR and Conoco-Phillips Optimized Cascade processes. An alternative cycle which is similar to those described above but maintains the refrigerant only in the vapor phase is the Brayton refrigeration cycle. The absence of phase change limits heat transfer to only sensible heat exchange in the cycle. Due to lower thermodynamic efficiency, its use has mostly been limited to small or medium scale LNG applications or where other considerations exist. Variations in separate project requirements and considerations such as specific power efficiency, safety, refrigerant availability, environmental conditions, and equipment configurations can have a significant impact on the evaluation and selection of these different available processes.

COMMON COMPRESSOR TYPES AND CONFIGURATIONS IN LNG REFRIGERATION

The design of LNG compressors involves large casing sizes, high flow coefficient impeller designs, high inlet relative Mach numbers, complex impeller flows and potential sidestream mixing. 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, which is typically used to pre-cool the gas, is usually the most technically challenging machine in terms of flow coefficient and inlet relative Mach number. Details of LNG gas turbines and compressor technology have been previously presented by Meher-Homji, et



al. (2011) and some of the material is drawn from this reference.

Process optimization during compressor selection should be completed 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 supplier, the process licensor and the project engineering team. Details of compressor machinery and process configurations may be found in Wehrman et al, (2011).

Compressor Casing Designs for Typical LNG Process Configurations

Common type of compressors used for the major processes are shown in Table 2.

Table 2. Compressor Casing Designs Used in Different Process Configurations.

Liquefaction Process	Service	Casing Design	Notes
APCI C3-MR	Propane Precooling	Horizontally Split	Propane is used for pre-cooling the gas.
APCI C3-MR	LP MR	Horizontally Split	LP, MP and HP MR is used for liquefaction and sub-cooling
APCI C3-MR	MPMR HPMR	Vertically Split (Barrel) Compressor Barrel Compressor	Note MP/HP can be combined in a back to back barrel compressor
APCI APX	Propane Precooling LP and HP C3	Horizontally Split	
APCI APX	LP MR	Horizontally Split	
APCI APX	MP/HPMR	Barrel Compressor	
APCI APX	LPN2	Horizontally Split	Sub-cooling is provided by a separate N2 refrigeration cycle
APCI APX	HP N2	Barrel Compressor	
CoP Optimized Cascade	Propane	Horizontally Split	
CoP Optimized Cascade	Ethylene	Horizontally Split	
CoP Optimized Cascade	Methane	Horizontal and Vertical Split	HP casing is a Vertically Split casing
DMR	Similar to C3-MR		One mixed refrigerant for pre-cooling and the second for liquefaction and sub-cooling

Some current typical train layouts are shown below:

- APCI C3-MR (SplitMR process): currently the workhorse driver for this configuration is the Frame 7EA (See Figure 14)
- APCI 2X50% solution: in this solution, two parallel trains are utilized each consisting of C3, LP and HP MR casings. (See Figure 15)
- CoP Optimized Cascade Process: drivers have been predominantly two shaft or multi-spool aero-derivative engines and each service is split into two parallel trains. (See Figure 16)



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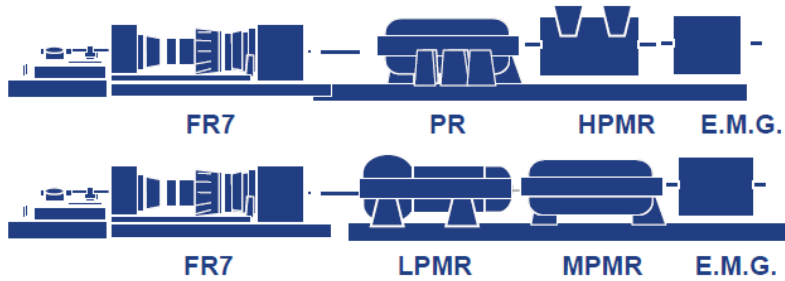


Figure 14. APCI SplitMR Configuration (Pelagotti and Baldassarre, 2014).

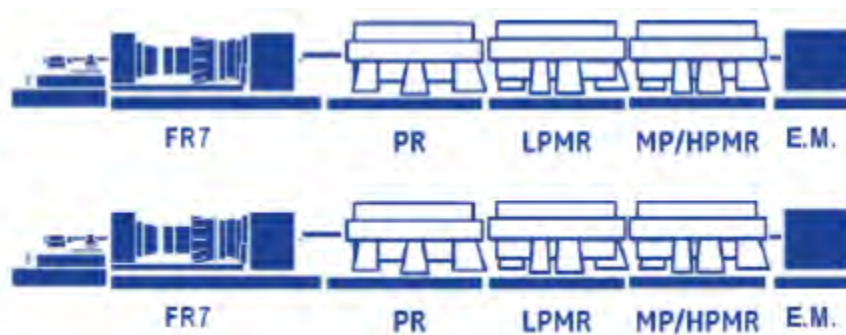


Figure 15. APCI 2x50% Configuration (Pelagotti and Baldassarre, 2014).

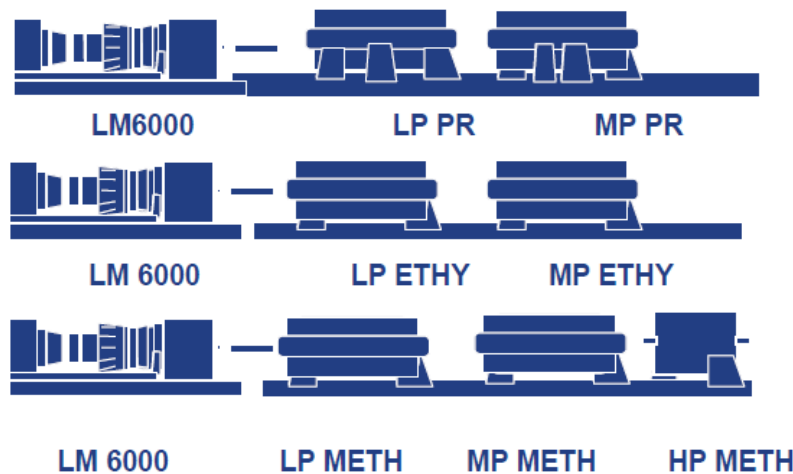


Figure 16. CoP Optimized Cascade Configuration, Total 6 Strings (Pelagotti and Baldassarre, 2014).

Compressor Types and Parameters

Compression of large volumetric flows of gas can be accomplished using either axial compressors, which are more suited to lower overall pressure ratios, or centrifugal compressors that enable higher pressure ratios. Some LNG processes use both axial and centrifugal



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compressors while others utilize only centrifugal compressors. Axial flow compressors have been used for low pressure MR service in the past but centrifugal compressors have continued to also be popular for this service. A multistage axial flow compressor is shown in Figure 17 and a typical horizontally split centrifugal compressor is shown in Figure 18. There is a trend towards the use of centrifugal machines due to their inherent robustness and increasing efficiencies approaching those of axial machines.

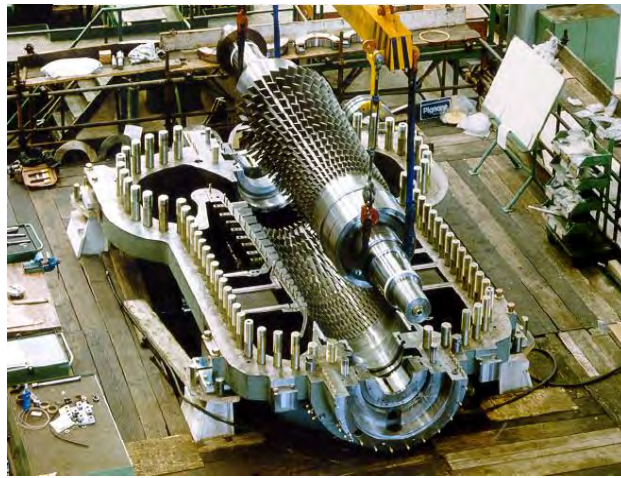


Figure 17. Axial Flow Compressor Used in LNG Service.



Figure 18. Horizontally Split LNG Centrifugal Compressor.

LNG refrigeration compressors can be extremely large, with centrifugal impeller diameters in the range of 1300-2000 mm (51-79 inches). A large 1750 mm (68.9 in.) diameter impeller is shown in Figure 19. 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. Inlet flows for centrifugal compressors can reach 300,000 m³/h (177,000 ft³/min). 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 (18 ft.) which require a shaft of suitable diameter at the bearings and end seals. Dry gas seals can have internal



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diameters of 350 mm (13.8 in.) and journal bearing diameters can be as high as 320mm (12.6 in.). A representation of the current technology limits for large LNG compressors is provided in Table 3.



Figure 19. Large 1750 mm (68.9 in.) Diameter Centrifugal Impeller.

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

CASING	
Weight of Barrel, tons	120 tons
Power in single casing	100 MW
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 Diameter	350 mm
Journal Bearing Diameter	320 mm
IMPELLER	
Inlet Relative Mach Number	1.05 model test
Inlet Relative Mach Number	0.975 full scale
Machine Mach Number	1.24 Model test
Machine Mach Number	1.20 full scale
Head Coefficient	Higher than 0.54
Head Per Impeller	More than 4000 m
Impeller Diameter	Up to 2000 mm
COUPLING	
Size	350mm
Power Transmitted	145 MW



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Most of the LNG refrigerant compressor casings are low pressure, horizontally split designs. Per API 617, when a relief valve is not specified, the maximum allowable working pressure (MAWP) of the casing must be 1.25 times the maximum discharge pressure. Sometimes, off-design process conditions can exceed 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. Higher pressure refrigeration services are accomplished in a radially split (barrel) compressors, which have higher limitations in maximum allowable working pressure. A large barrel LNG compressor is shown in Figure 20.



Figure 20. Large Barrel Type LNG Compressor.

Horizontally Split and Barrel Designs

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

Mixed refrigerant service which also includes 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 suction pressure levels in two or three compressors (horizontally split and barrel).

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

Methane refrigeration services require the development of a relatively high compression ratio that is generally often achieved utilizing a three casing solution. The last casing is a barrel type compressor while the first and second are horizontally split. These compressors do not generally present unique technical challenges other than those normally related to trains involving multiple large units.



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Double Flow Compressors

As LNG plant capacities have continued to increase, centrifugal compressor volumetric flow rates and casing sizes have become very large. A double-flow casing configuration can be used to reduce the required casing size. A double flow compressor is essentially two parallel compressors in a single compressor casing. The inlet flow is externally divided and enters the casing via two nozzles and passes through each individual section of the compressor and is recombined at the final diffuser exiting the casing through a single nozzle. The advantage of the double flow configuration is that the same casing size can accommodate twice the volume flow. Losses in the flow paths through a double flow compressor should ideally be identical. In practice, this is difficult to achieve and some minor deviation in performance between the two sections is likely. Double flow compressors have a limitation on the number of stages per section and therefore on pressure ratio which can lead to one or more additional casings being required for higher overall pressure ratios. A double flow compressor layout is shown in Figure 21, with a photograph of the horizontally split casing and impeller being depicted in Figure 22.

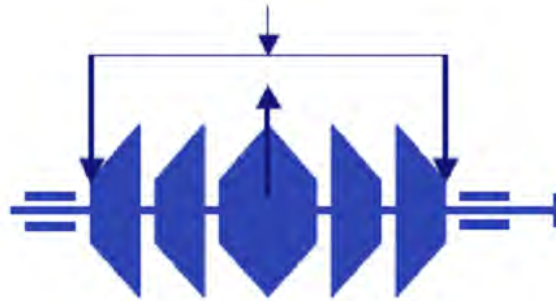


Figure 21. Double Flow Compressor Schematic.



Figure 22. Double Flow Compressor Casing and Rotor.



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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 23. This arrangement also allows for inter-cooling during the compression process thus improving efficiency. It will also 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 mid-span is good from a damping perspective.

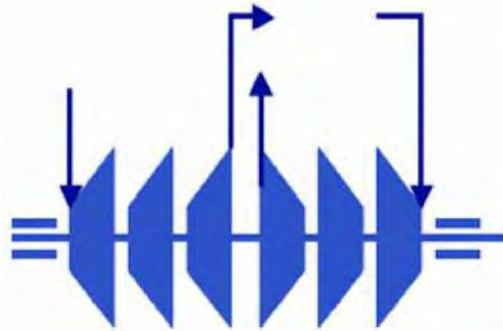


Figure 23. Schematic of a Back-to-Back Compressor.

Another area requiring careful attention in a back-to-back compressor design is the rotordynamics. The center seal in this arrangement is at the mid-span 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 mid-span, 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 configuration, the balance piston is located closer to a radial bearing and is often considered to have less of an influence on rotor stability in comparison. 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 breaks 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 these designs should be carefully reviewed

Sidestream Considerations

Sideload compressors represent a unique configuration that is found in many LNG refrigeration compressor applications, particularly propane and some dual mixed refrigerant (DMR) services. These multi-section machines include one or more sidestreams that are introduced to the compressor at intermediate pressures between the first section suction and final section discharge pressures. The flow from lower pressure sections does not leave the compressor casing, but instead is mixed internally with a higher pressure sidestream, resulting in the absence of any intercoolers. This is illustrated in Figure 24 below for a two section compressor.

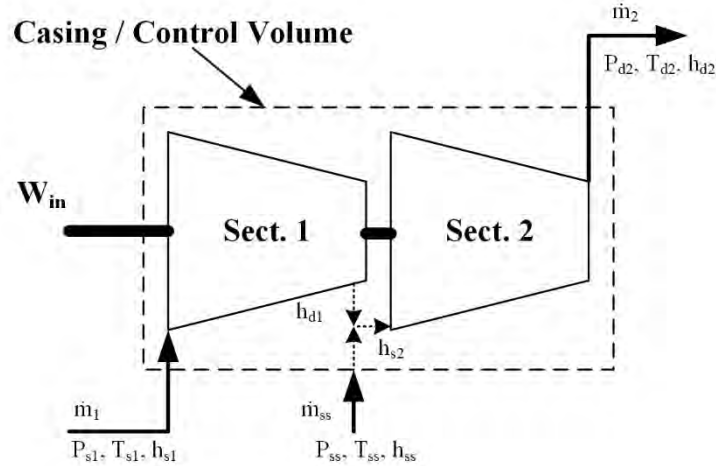


Figure 24. Sideload Compressor Schematic.

The primary difference in performance evaluation between this type of design and other multi-section compressors is associated with the inability to measure the discharge conditions of all but the last section and the inlet conditions of all but the first section. While attempts to estimate these conditions during factory acceptance testing are temporarily provided with test instrumentation, longer term operation and monitoring of these conditions must be approximated. An overall First Law of Thermodynamics energy balance of the multiple sections may be completed by constructing a control volume coincident with the compressor casing. The resulting governing equation is given by:

$$W_{in} = \dot{m}_2 h_{d2} - \dot{m}_1 h_{s1} - \dot{m}_{ss} h_{ss} \quad (4)$$

Or, more generally,

$$W_{in} = \dot{m}_{total} h_{d_{nsect}} - \dot{m}_1 h_{s1} - \sum_{i=1}^{nsect-1} \dot{m}_{ss_i} h_{ss_i} \quad (5)$$

where,

$$\dot{m}_{total} = \dot{m}_1 + \sum_{i=1}^{nsect-1} \dot{m}_{ss_i} \quad (6)$$

Although these relations allow the total amount of energy (and power) and the changes in this total to be predicted and calculated during longer term operation, they do not provide insight into the initial distribution of power for each section or changes in the absorbed power per section during continued operation. Additional relations are necessary to determine the performance of individual sections, however, it is apparent that these equations are based upon assumptions of the values of pressures and temperatures between sections. The energy absorbed by each section is given by:

$$W_i = \dot{m}_i (h_{di} - h_{si}) \quad (7)$$

and,

$$W_{in} = \sum_{i=1}^{nsect} W_i \quad (8)$$

It is evident from Figure 24 that the value of the discharge enthalpy of section 1 and the suction enthalpy of section 2 are unknown values that must be estimated. These values are related through a mixing calculation that is given as:



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$$h_{s2} = \frac{\dot{m}_1 h_{d1} + \dot{m}_{ss} h_{ss}}{\dot{m}_1 + \dot{m}_{ss}} \quad (9)$$

Or once again, in more general terms,

$$h_{s(i+1)} = \frac{\dot{m}_i h_{di} + \dot{m}_{ssi} h_{ssi}}{\dot{m}_i + \dot{m}_{ssi}} \quad (10)$$

These mixing relationships can be converted into values of pressure and temperature. The ASME PTC 10, “Performance Test Code on Compressors and Exhausters”, and common industry practice assumes that the discharge pressure of the preceding section and suction pressure of the subsequent section are common and equal to the sidestream nozzle total pressure. The associated temperatures are dependent upon the relative values of their related enthalpy and a further assumption that the two streams are completely and uniformly mixed. Of course, the individual stream total pressures and pressure differentials are not equal and temperature profiles are likely stratified and not uniformly mixed.

Individual predicted section performance characteristics are typically provided using these assumptions to allow actual operating evaluation to be completed with the understanding that some amount of error will be incurred in the calculations. These uncertainties inherent in the assumption of inter-section operating parameters may limit the ability to accurately determine performance degradation of any particular section during continued operation.

Different LNG processes also have different sidestream to core mass flow ratios as shown in Figure 25. These differences in mass flow ratios between sidestreams and core flows typically vary from section to section in multiple section machines and are also impacted by feed gas and mixed refrigerant compositions where the refrigerant flow, such as propane, is utilized to cool or condense the other fluid.

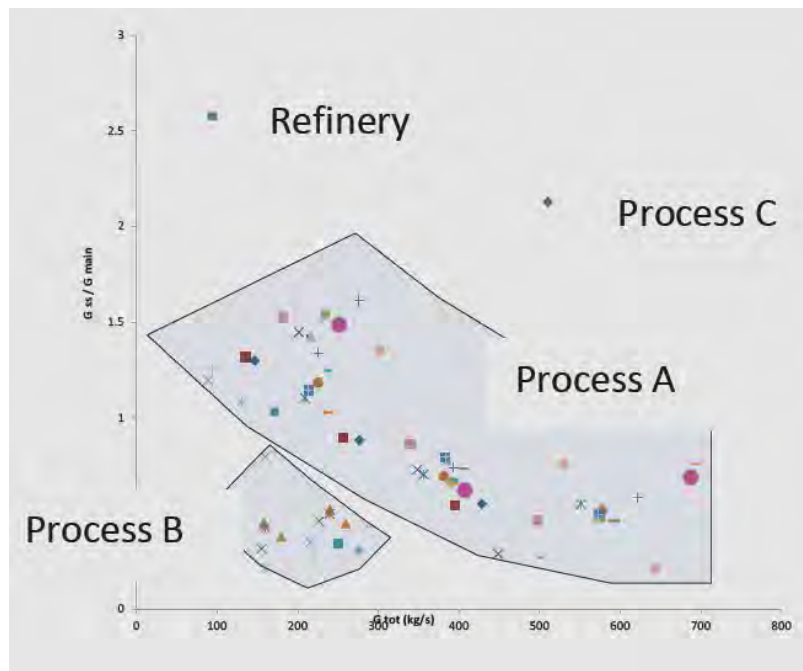


Figure 25. Typical Sidestream to Core Ratios for Different LNG Processes, (Pelagotti and Baldassarre, 2014).



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COMPRESSOR AERO/THERMODYNAMIC SELECTION AND DESIGN

The design and selection of refrigeration compressors in natural gas liquefaction (LNG) services represents one of the more challenging applications for centrifugal compressors. Several factors contribute to the unique issues created by these refrigeration applications that do not exist in others. These include high Mach numbers associated with the relatively high molecular weight gases involved and substantial volumetric flow rates that have evolved as plant capacities have increased over time. Accepted sizing guidelines and physical design limits have been tested as a result.

A number of design parameters that are controlled by environmental factors remain constant across the wide range of refrigeration applications. It has been noted previously that compressor discharge pressures are established by the condensing pressure of the refrigerant which is related to the cooling source supply temperature that is either ambient air conditions or an alternate coolant (e.g. water) likely influenced by ambient temperature. Compressor suction pressure is also normally maintained slightly above atmospheric conditions during steady state operation in order to maximize refrigeration duty. Accordingly, given these relatively constant inlet and discharge pressure and temperature conditions existing for a specific location, the only parameter remaining that can vary the refrigeration loop cooling duty is the cycle refrigerant mass flow rate. In the simple case of a single stage vapor compression loop, this variation in mass circulation rate is directly related to the compressor inlet volumetric flow rate since the suction pressure and temperature remain relatively constant. This relationship is also valid for more complex multi-stage and cascaded refrigeration loops. Thus the ever increasing desired capacities of LNG process trains has resulted in higher inlet volumetric flow rates imposed on refrigeration compressor designs.

Sandberg (2016) provided guidelines and a methodology that allows independent sizing, selection and verification of equipment supplier compressor selections. Although general in nature, the information and methods presented also apply to refrigeration compressor applications. Fundamental performance parameters related to centrifugal compressor design and selection include the inlet volumetric flow rate, compression specific work (head), efficiency, and specific power (work input). While there are a few thermodynamic models that may be used to calculate and describe these parameters, the most predominantly used in process and large-scale refrigeration compressors is the polytropic model which will be utilized herein. The polytropic head may be derived from measurable compressor operating parameters as shown below:

$$H_p = C_1 \frac{Z_s T_s}{MW} \frac{n}{n-1} \left[\left(\frac{P_d}{P_s} \right)^{\frac{n-1}{n}} - 1 \right] \quad (11)$$

$$\text{where: } n = \frac{\ln \left(\frac{P_d}{P_s} \right)}{\ln \left(\frac{v_s}{v_d} \right)} \approx \frac{\gamma \eta_p}{\gamma (\eta_p - 1) + 1}$$

Inspection of this relation demonstrates that the polytropic head is dependent upon the pressure ratio across the compression stage, suction conditions of temperature and compressibility, and the molecular weight and additional properties of the refrigerant. As previously noted, these values tend to remain constant or vary over a limited range with changes in ambient temperature for a specific location. This yields a relatively narrow range of stage or section polytropic head for refrigeration applications. Sandberg (2016) noted that the average generated head per stage of compression varied over a range of approximately 6000 to 14,000 ft-lbf/lbm (17.93 to 41.84 kJ/kg). Figure 26 provides data on head per stage for a significant number of actual stages proposed or manufactured for refrigeration service including LNG liquefaction. This data represents compression staging from multiple equipment manufacturers. Obviously, there are data points that fall outside this range, but the majority of the data falls within the listed range.



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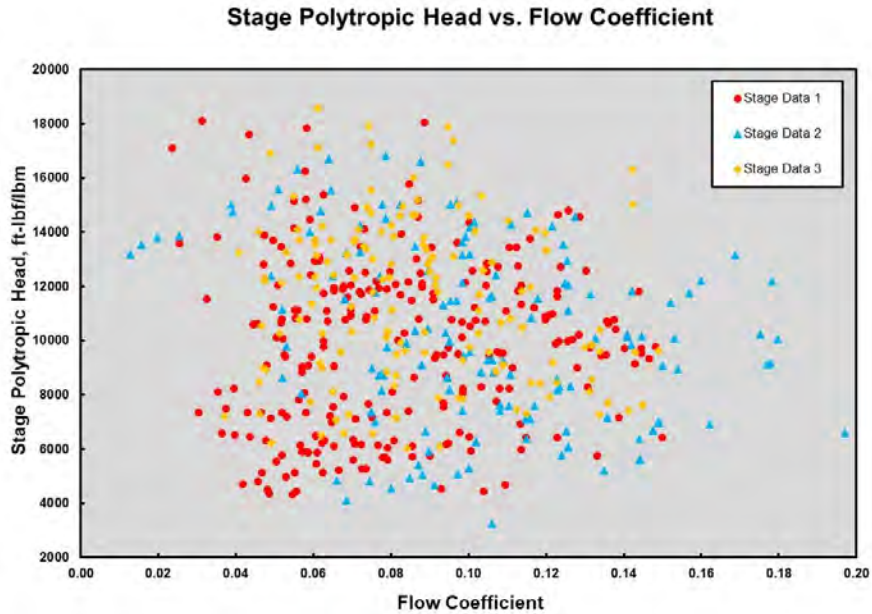


Figure 26. Stage Polytopic Head for Refrigeration Applications.

Polytropic head is also related to the square of the impeller tip speed as can be demonstrated through the use of impeller velocity diagrams. This will become an important point in further developments. The tip speed is given by the equation:

$$U_{ip} = C_2 D_2 N \quad (12)$$

While a plot of impeller tip speed versus the flow coefficient can be presented, it displays a comparable amount of scatter similar to the plot of stage polytopic head versus flow coefficient above. A typical range of impeller tip speeds encountered in actual applications is 650 to 900 ft/sec (198 to 274 m/sec), although it is possible for some applications to fall outside of this range. Low molecular weight gases are primarily limited by impeller material stress considerations, so higher tip speeds in these applications are necessary to generate moderate pressure ratios for significant generation of head. In contrast, high molecular weight gases will generate substantial pressure ratios for lower amounts of generated polytopic head and the associated lower tip speeds.

Machine Mach Number

Another parameter that is important in the compression of higher molecular weight gases is the machine Mach number (also sometimes referenced as the peripheral Mach number). It is defined as the ratio of the impeller tip speed to the acoustic velocity at suction conditions and is given by the relation:

$$M_m = \frac{U_{ip}}{a_s} \quad (13)$$

The machine Mach number does not represent an actual physical quantity since the flow field in the tip region of the impeller is multi-dimensional and the local acoustic velocity is different from that of the inlet, however, this parameter is indicative of other performance characteristics. Higher molecular weight gases (e.g. many refrigerants) will typically have lower acoustic velocities than lighter gases encountered in industrial applications. This results in higher machine Mach numbers for the normal range of tip speeds. Machine Mach



numbers for a large set of refrigeration related stages are presented in Figure 27.

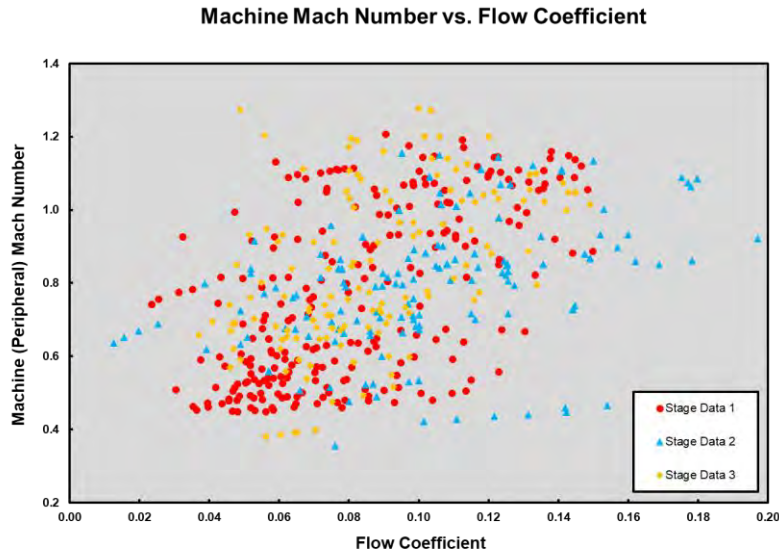


Figure 27. Machine Mach Numbers for Refrigeration Stages.

A significant number of data points in the figure above have machine Mach numbers with values above unity. Again, since the machine Mach number does not represent an actual physical quantity, the local absolute velocities are not necessarily greater than the local acoustic velocity. One utility of the machine Mach number is associated with its relationship to stage performance characteristic curves. This is directionally illustrated in Figure 28.

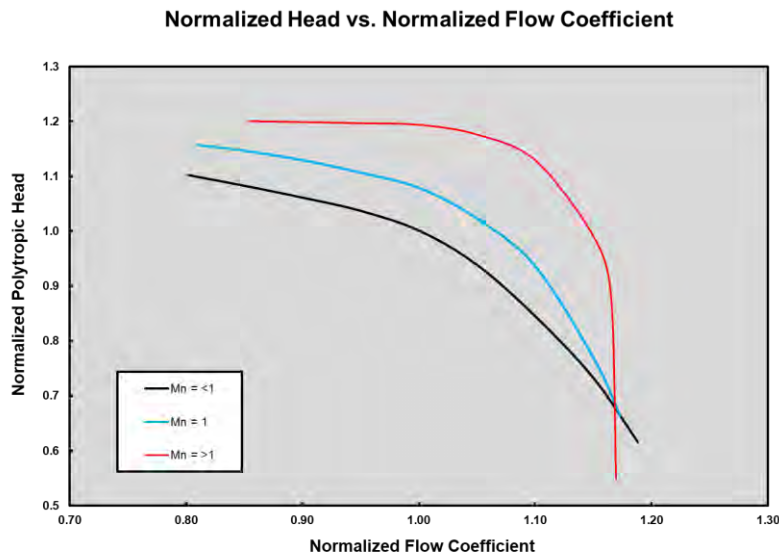


Figure 28. Machine Mach Number Performance Curve Influence.



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As the machine Mach number approaches and exceeds a value of one, a number of impacts in stage performance characteristics become evident. The amount of head produced at specific values of flow coefficient increases with the increase in rotational and tip speed, assuming constant suction conditions and acoustic velocity. Another obvious impact is the reduction of flow stability and range as the machine Mach number is increased. Stability in this case is defined as the difference in flow between the surge and design points along a constant speed compressor characteristic. Range may either be defined as the difference in flow from the surge point or design point to the flow rate at overload (choking) conditions. Finally, as machine Mach numbers increase above unity, the slope of the head characteristic curve reduces significantly between surge and overload. This, coupled with the reduction of range, may create issues with control and operation of the compressor.

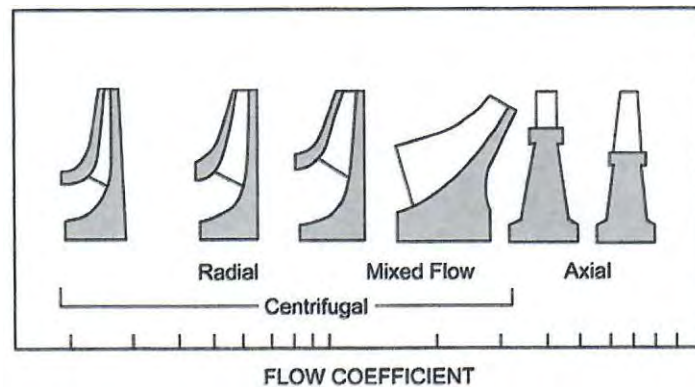
Comparison of performance parameters between different compressor designs, services and conditions provides insight into benefits and limitations of various designs. Although dimensional forms of these parameters provide some information, the evaluation of dimensionless forms of fundamental compressor performance parameters allows substantially more insight into compressor performance across a broad range of applications, handled gases and operating conditions. Machine Mach number is such an example where impeller tip speed is compared against compressed fluid acoustic velocity. Tip speeds limiting compressor operating conditions in high molecular weight, relatively low acoustic velocity services may not be as restrictive with other gases when the machine Mach number is used as a basis.

Flow Coefficient

The flow coefficient has already been included as an independent variable in the previous comparisons of operating parameters. It is a valuable, dimensionless parameter in that it allows comparisons of different stage inlet flow rates, rotational speeds, inlet gas densities and impeller physical dimensions. The equation for the flow coefficient is given as:

$$\phi \approx \frac{Q_{sa}}{U_{tip} A_{char}} = C_3 \frac{Q_{sa}}{ND_2^3} \tag{14}$$

Compressor stage flow coefficient represents an important independent compressor performance parameter that when paired with other dimensionless parameters, allows more detailed evaluation and comparisons between different compressor designs and selections. Another benefit of knowing the flow coefficient of a specific compression application is associated with the stage impeller geometry. Figure 29 presents a comparison of the flow coefficient’s impact on impeller relative dimensions and geometries.



Courtesy of Solar Turbines, Inc.

Figure 29. Flow Coefficient Impact on Impeller Geometry.

As the inlet volumetric flow and associated flow coefficient increase, the inlet eye area of radial impellers grows larger and the shroud eye diameter approaches the outer tip diameter of the impeller. In order to accommodate a smooth transition from an axial to radial flow



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direction, the axial length of the impeller also gets larger. The amount of head produced by the impeller is known to be a function of the square of the tip speed (actually the difference of the squares of the outer diameter and eye shroud tip diameter), so it follows that the amount of head produced at a constant rotational speed may be reduced as larger flow coefficients and impeller eye diameters are encountered.

Significant increase in flow coefficient results in an impeller design that no longer discharges flow perpendicular to the axial direction, but at an angle less than 90 degrees from axial. These mixed flow designs accommodate higher volumetric flows, but produce lower head at constant rotational speed. It is also evident that since the discharge flow is not perpendicular to the axis, the associated stationary components included in the stage will also require increased axial length beyond radial design impellers. Although the mixed flow impeller shown in the diagram above is not supplied with a shroud, both shrouded and unshrouded mixed flow impeller designs are available.

Additional increases in required flow coefficient result in the transition from mixed flow impellers to a purely axial compressor design as is illustrated in Figure 29. Recent larger LNG liquefaction train capacities have resulted in many refrigeration compressor applications moving from radial to mixed flow designs. There are even some examples of mixed refrigerant compressors being supplied as axial designs. Figure 30 provides an additional illustration of the impact of increasing flow coefficient on the physical dimensions of two impellers of the same tip diameter.



Courtesy of Dresser-Rand: A Siemens Business

Figure 30. Comparison of Low and High Flow Coefficient Radial Impellers.

The impeller on the left in the photograph above is obviously of a lower flow coefficient design. Impeller eye area and outer tip width are substantially smaller than the high flow coefficient wheel design on the right. The tip-to-eye shroud diameter ratio is also greater on the low flow coefficient wheel. Another important observation afforded by this photograph is the relative difference in axial length of the two designs with the high flow coefficient impeller requiring a much greater axial length. Finally, vanes extending well into the eye area of the high flow coefficient impeller, along with tip vanes not purely parallel to the impeller axis, are also evident. This three-dimensional design may not be required of lower flow coefficient wheels where the vanes do not extend into the eye and are of a more two-dimensional nature. The outer diameter to impeller bore ratio is another illustrated and important design parameter related to lateral rotordynamics and represents a compromise between aerodynamic and rotordynamic interests that the equipment designer must balance.

Fundamental Dimensionless Parameters

A number of other dimensionless compressor performance parameters exist that allow comparison and evaluation of compressor selections and designs across a wide variation of operating conditions including gas compositions, densities, flow rates, rotational speeds



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and physical dimensions. One of the more fundamental dimensionless parameters used during design, testing and evaluation of designs is the head coefficient. It is provided in the equation below and is the ratio of the head generated in a compression stage divided by the impeller tip speed squared.

$$\mu_p = C_4 \frac{H_p}{U_{tip}^2} = C_5 \frac{H_p}{(D_2 N)^2} \quad (15)$$

Aungier (1995, 2000) proposed a relationship between polytropic head coefficient and flow coefficients that represented existing optimal design practice. These relations act as a locus of head coefficients at design point flow coefficients for a broad family of stages at assumed optimum efficiencies. It does not represent a complete performance curve for a single stage or section. Aungier's design polytropic head coefficients for both vaned and vaneless stationary diffuser stages and sections are included in Figure 31.

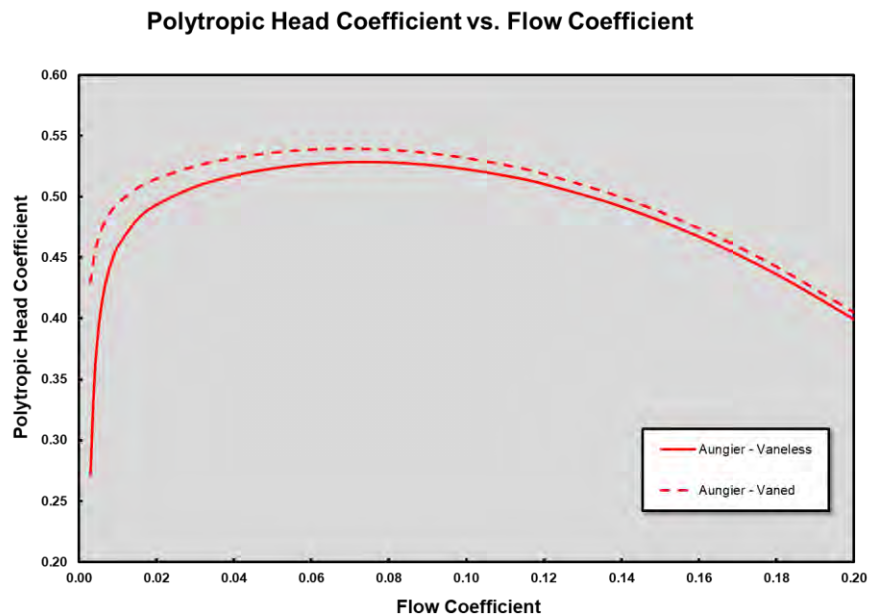


Figure 31. Aungier Polytropic Head Coefficient Characteristic.

An examination of the above figure shows that the polytropic head coefficient has a maximum value region for flow coefficients ranging from approximately 0.02 to 0.11. This optimum range represents impeller designs that are primarily radial in nature. Impeller designs at flow coefficients below the lower limit of this range are influenced by losses attributed to their relatively narrow flow passages and boundary layer effects. Impeller designs with flow coefficients above this optimum range represent impellers that are approaching or are of mixed flow designs. Recall that differences in inlet shroud and impeller outer diameters are diminishing at these higher flow coefficients, resulting in lower head generating ability.



Polytropic Efficiency vs. Flow Coefficient

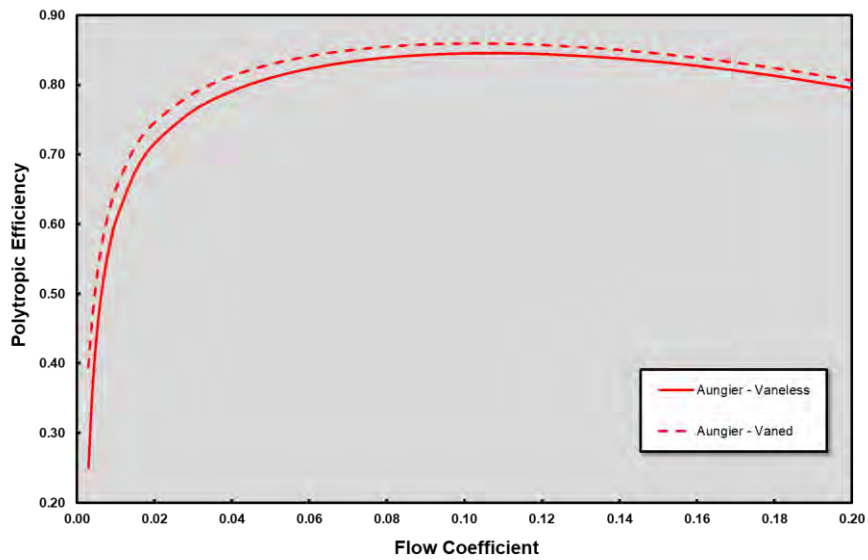


Figure 32. Aungier Polytropic Efficiency Characteristic.

Aungier (1995, 2000) also provided an estimate of optimum polytropic stage efficiency, η_p , as a function of design flow coefficient which is another important dimensionless parameter. This formulation is plotted in Figure 32 for both vaneless and vaned diffusers. When the polytropic head coefficient is divided by the polytropic efficiency, the work input coefficient is obtained. The work input represents the total specific energy absorbed by the compressor stage and is given by the following equation.

$$\tau = \frac{\mu_p}{\eta_p} \approx \frac{h_d - h_s}{U_{ip}^2} \quad (16)$$

Utilizing the above relationships for polytropic head and efficiency, a plot of work input versus design flow coefficient can be derived. It is provided in Figure 33 below.



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Work Input vs. Flow Coefficient

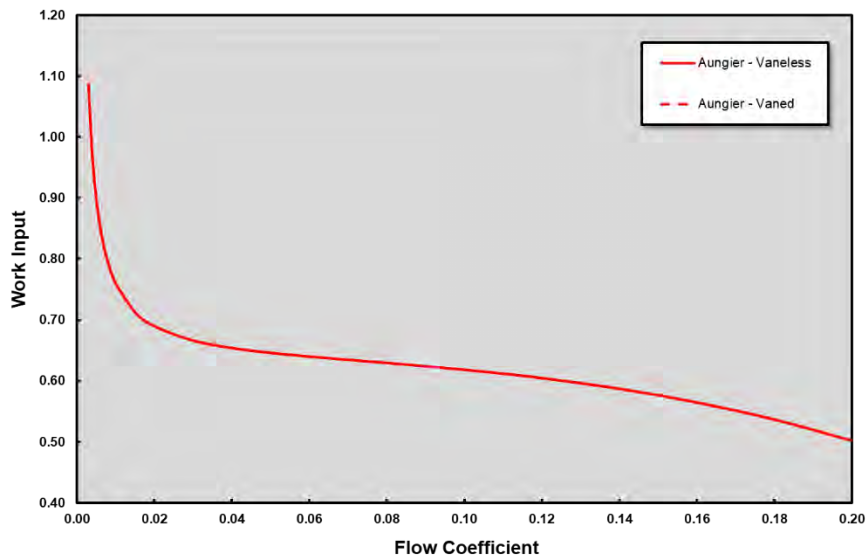


Figure 33. Aungier Work Input Characteristic.

Reference to this figure provides some additional insight and confirmation of previously mentioned points. Three distinct regions are evident with the work input characteristic. At the low end of the flow coefficient range, the slope of the work input curve is relatively steep. This shows that the specific work absorbed decreases rapidly as the flow coefficient increases. Once again, stage performance in this region is substantially influenced by losses due to narrow flow passages. A second region exists over a considerable middle range of flow coefficients (approximately 0.02 to 0.11) with a very shallow reduction of work input with increasing flow coefficient. This correlates closely with the optimum ranges of maximum polytropic head coefficient and efficiency. It represents a preferred design flow coefficient range for compressor staging. The third region represents staging designs approaching and within the mixed flow regime. Although the efficiency of the staging remains relatively high, the polytropic head coefficient decreases due to the reduced head making ability of these impeller configurations. This reduced ability to generate work and produce compression ratio for a comparable tip speed is represented.

Additional Dimensionless Similarity Parameters

Sandberg (2016) presented two additional dimensionless similarity parameters that improve the ability to evaluate and estimate alternative and optimal compressor design parameters based upon required available dimensional performance parameters of volumetric flow rate and work (polytropic head). Although it is possible to perform such analyses with the previously defined dimensionless flow and head coefficients, the complication is that both of these similarity parameters are based upon the impeller tip speed which has been shown to be a function of both rotational speed and impeller outer diameter. An infinite number of combinations of impeller diameter and rotational speed can be envisioned for any specific value of the tip speed. These additional two parameters allow the separation of diameter and speed, resulting in more optimum selections or comprehensive evaluation of supplier selections.

Specific speed is the first of these two additional similarity parameters which is given in dimensionless form in the equation below. A valuable relation between the specific speed and the flow and head coefficients that can be derived is also provided.



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$$ns = C_6 \frac{N\sqrt{Q_{sa}}}{H_p^{0.75}} = \sqrt{\pi} \frac{\sqrt{\phi}}{\mu_p^{0.75}} \quad (17)$$

The second similarity parameter is the specific diameter in the equations given with both dimensional and dimensionless variable forms as follows.

$$ds = C_7 \frac{D_2 H_p^{0.25}}{\sqrt{Q_{sa}}} = \frac{2}{\sqrt{\pi}} \frac{\mu_p^{0.25}}{\sqrt{\phi}} \quad (18)$$

It follows that the product of these two parameters is:

$$nsds = \frac{2}{\sqrt{\mu_p}} \quad (19)$$

The utility and benefit of these two additional similarity parameters becomes more apparent when they are evaluated for a number of different cases. First of all, using the Aungier (1995, 2000) derived functional relationship between flow and polytropic head coefficient, the dimensionless specific speed and specific diameter may be calculated for specific values of the flow coefficient and related head coefficient. Casey, et al. (2010) presented some additional development and more accurately extended the Aungier relations into the mixed flow regime with respect to the Cordier (1955) characteristic. Sandberg (2016) superimposed actual compressor sectional performance parameters onto plots of these functional relations to demonstrate their correlation to actual compressor selections.

While good agreement justified the value of using these alternative parameters in evaluation of sectional compressor performance across a range of different applications, they should also be valid for individual stage performance in refrigeration related services. In order to validate the extension of Sandberg's (2016) results to individual stage performance in refrigeration services, a large number of manufactured and proposed stage data has been evaluated. Three different data sets are included, representing stage selections from multiple compressor suppliers. More than 600 data points have been included in these three data sets.

A plot of dimensionless specific diameter versus flow coefficient is provided in Figure 34. Continuous functions derived from the Aungier (1995, 2000) relations for both vaned and vaneless diffusers and that recommended by Casey, et al. (2010) are included along with the significant number of points provided from the actual stage selection data sets.



Dimensionless Specific Diameter vs. Flow Coefficient

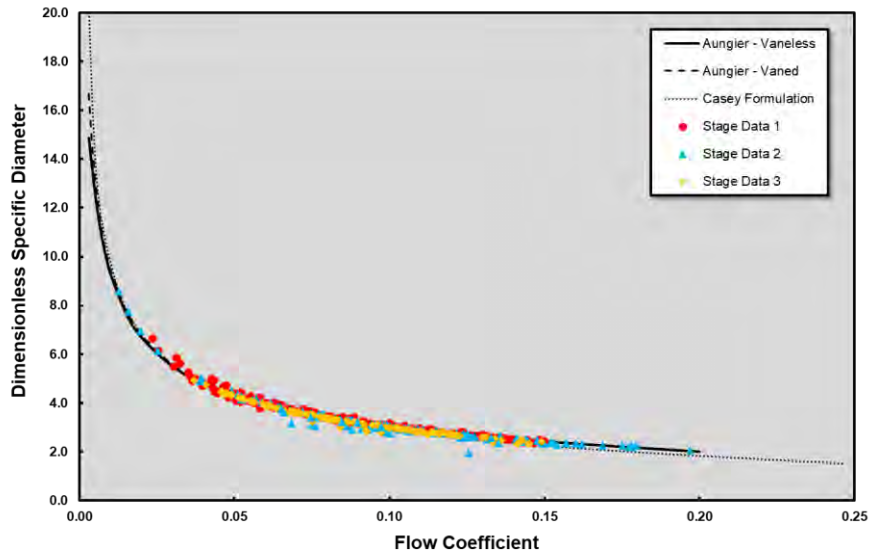


Figure 34. Refrigeration Staging Specific Diameter versus Flow Coefficient.

A similar plot of dimensionless specific diameter versus specific speed (the Cordier characteristic) is presented as Figure 35.

**Dimensionless Specific Diameter vs. Specific Speed
(Cordier Characteristic)**

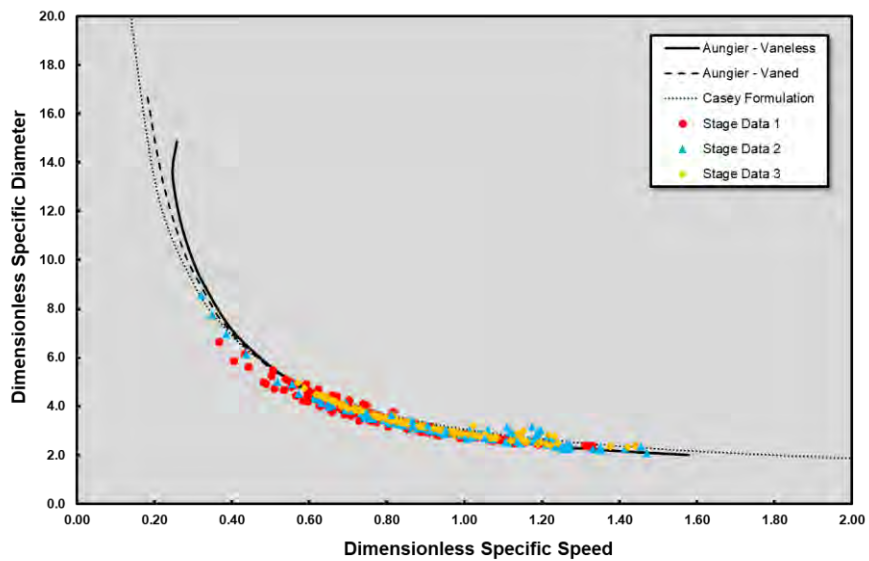


Figure 35. Refrigeration Staging Specific Diameter versus Specific Speed.



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Close agreement displayed between the individual stage data and the prediction functions, especially at higher flow coefficients where many current LNG refrigeration applications are occurring, supports the use of these functions for independent design, selection and evaluation. The challenge is associated with the methodology used in applying these results. Sandberg (2016) outlined logic and calculation steps that may be utilized in cases of variable speed, defined impeller diameter, and constant speed cases to utilize this information. Those steps are repeated below.

Selection Methodology

The information developed in this section allows for preliminary compressor selections to be completed independently from the equipment supplier. They will probably be slightly different from that ultimately provided by the equipment supplier, but they do offer the user and purchaser the ability to evaluate one or more potential configurations prior to engaging a supplier. This methodology will also provide additional tools and criteria for evaluation of equipment supplier selections. The following three methods of selection and evaluation are based upon five dimensionless parameters; namely the flow coefficient, polytropic head coefficient, polytropic efficiency, dimensionless specific speed, and dimensionless specific diameter. Depending upon the actual application, one or more of these evaluation procedures may be utilized to produce a preliminary equipment selection or evaluation of a proposed selection for one or more compression stages.

Flow Coefficient

A primary and flexible approach to making a selection is through an assumption of the flow coefficient. If reference is given to the Aungier plots of polytropic head coefficient and efficiency, it is apparent that the highest efficiency and value of head coefficient occur across a flow coefficient range from a value of approximately 0.05 to 0.11. Optimum selections should, if possible, lie within this range of flow coefficients, however it is recognized that progressively higher volumetric flow rates associated with current LNG processes may make this impossible for one or more services. The solution steps are:

1. Calculate required polytropic head and inlet volumetric flow rate from provided process data. Provide an initial estimate of the polytropic efficiency (reference to the Aungier plot of polytropic efficiency is a good start).
2. Select an assumed value or range of values of the flow coefficient (Note: optimum values normally lie between values of 0.05 to 0.11).
3. Determine the value(s) of the specific diameter from the flow coefficient versus specific diameter relationship [Figure 34]. Calculate the impeller diameter(s) from the specific diameter equation [Equation 18].
4. Determine the value(s) of the specific speed from the specific speed versus specific diameter relationship [Figure 35]. Calculate the design operating speed(s) from the specific speed equation [Equation 17].
5. Determine the polytropic head coefficient(s) using the product of specific speed and specific diameter and the related equation [Equation 19].
6. Determine the range of polytropic efficiencies for each flow coefficient using the Aungier relationship (suggest 100% and 95% of predicted average efficiencies) [Figure 32].
7. Iterate on these steps with resulting efficiencies until efficiency value convergence if improved accuracy is desired.

Situations exist where the maximum diameter derived at the minimum flow coefficient of the range is smaller than that available from a given manufacturer's design range. It should be noted that impeller sizes will be larger at the lower flow coefficients and rotational speed higher at the higher flow coefficients. In the case where the resulting impeller diameter is smaller than is actually available, the selected flow coefficient will need to be lowered to derive larger impeller diameter solutions. Alternatively, the fixed diameter solution method presented below may be utilized with a given minimum impeller diameter provided by an equipment supplier.

Conversely, the resulting rotational speed at the highest flow coefficient in the optimum range may fall below that needed for a specific driver or application. In this case, a higher flow coefficient will need to be evaluated, or the fixed rotational speed method may be utilized if the required speed is known. Speed increasing or reducing gears may also be included before or between casings,



but mechanical efficiency losses attributed to gears and added train complexity should be weighed against compressor efficiency reductions.

Fixed Impeller Diameter

In the case where impeller diameter is known or selectable from a choice of discrete sizes, the fixed impeller diameter methodology provides a direct sizing approach that could only be achieved through iterations with the flow coefficient method. Here the impeller diameter becomes the independent parameter and all other performance parameters, including the inlet flow coefficient, are obtained from the calculation procedure. The solution steps are:

1. Calculate required polytropic head and inlet volumetric flow rate from provided process data. Provide an initial estimate of the polytropic efficiency (reference to the Aungier plot of polytropic efficiency is a good start).
2. Select one or more assumed value(s) of the impeller diameter. Actual equipment supplier information may be useful.
3. Determine the value(s) of the dimensionless specific diameter(s) from the specific diameter equation [Equation 18].
4. Determine the value(s) of the specific speed from the specific speed versus specific diameter relationship [Figure 35]. Calculate the design operating speed(s) from the specific speed equation [Equation 17].
5. Calculate the resulting flow coefficient(s) from the resulting value(s) of impeller diameter and rotational speed utilizing the equation for flow coefficient [Equation 14].
6. Determine the polytropic head coefficient(s) using the product of specific speed and specific diameter and the related equation [Equation 19].
7. Determine the range of polytropic efficiencies for each flow coefficient using the Aungier relationship (suggest 100% and 95% of predicted average efficiencies) [Figure 32].
8. Iterate on these steps with resulting efficiencies until efficiency value convergence if improved accuracy is desired.

Fixed Rotational Speed

The final method to be presented is based upon the assumption of a constant rotational speed where it becomes the independent variable. All other design parameters are derived from the various relations. This method is most useful in applications such as direct drives utilizing fixed speed electric motors or limited speed range drives such as single shaft gas turbines.

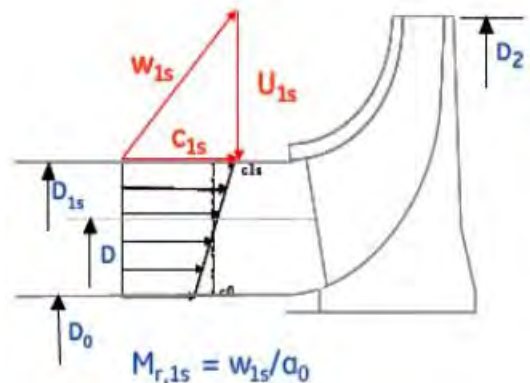
1. Calculate required polytropic head and inlet volumetric flow rate from provided process data. Provide an initial estimate of the polytropic efficiency (reference to the Aungier plot of polytropic efficiency is a good start).
2. Select one or more assumed value(s) of the rotational speed. This may be dictated by the driver involved.
3. Determine the value(s) of the dimensionless specific speed(s) from the specific speed equation [Equation 17].
4. Determine the value(s) of the specific diameter from the specific speed versus specific diameter relationship [Figure 35]. Calculate the impeller diameter(s) from the specific diameter equation [Equation 18].
5. Calculate the resulting flow coefficient(s) from the resulting value(s) of impeller diameter and rotational speed(s) utilizing the equation for flow coefficient [Equation 14].
6. Determine the polytropic head coefficient(s) using the product of specific speed and specific diameter and the related equation [Equation 19].
7. Determine the range of polytropic efficiencies for each flow coefficient using the Aungier relationship (suggest 100% and 95% of predicted average efficiencies) [Figure 32].
8. Iterate on these steps with resulting efficiencies until efficiency value convergence if improved accuracy is desired.

Utilization of one or more of the aforementioned methods can result in reasonably accurate compressor stage selections. While these selections provide an independent design for a given application, it must be recognized that selections provided by the equipment supplier must ultimately be used. The expectation is that the equipment supplier's design must govern since it is assumed that the supplier will support their selection with some form of performance guarantee. Accordingly, the selections derived from these methods will provide reasonably accurate estimates of potential selections provided or allow additional evaluation of supplier proposed selections and configurations.



Inlet Relative Mach Number

Another impact of the increased volumetric flows experienced with recent refrigeration compressor applications is connected with the average inlet velocity and associated inlet relative Mach number at the impeller inlet eye. A number of recent publications, including those by Sorokes and Kopko (2007), Sorokes, et al. (2009), Meyer-Homji, et al. (2007), Ghizawi, et al (2012), and Valente, et al (2018), have investigated and attempted to outline the challenges associated with impeller designs having inlet relative Mach numbers approaching and traversing sonic conditions. The flow in the eye area of the impeller is very complex and highly non-uniform. Figure 36 acts as a reference to a more complete discussion of some of these complexities.



Courtesy of BHGE

Figure 36. Impeller Inlet Eye Diagram.

The axial velocity component of the inlet volumetric flow rate is depicted as black vectors and demonstrate a linearly increasing variation from the eye hub (c_0) to the eye shroud (c_{1s}). This velocity must then be vectorially added to the tangential component of the impeller rotation, which also varies in amplitude from the hub to shroud, to derive the impeller gas absolute velocity distribution at the impeller eye region. The red vector diagram illustrates the determination of the absolute gas velocity at the shroud portion of the impeller eye (w_{1s}). A further complication arises if the inlet gas flow is three dimensional instead of the relatively simple linear, one dimensional variation shown above, and is normally the case in actual applications. Inlet guide vanes can impose pre-whirl into the gas velocity that has a tangential component that is either in the same or opposite direction of impeller rotation. Even without including such potential additional variations, it is obvious that the gas velocity field at the impeller eye is highly non-uniform with maximum absolute values likely occurring at or near the shroud region of the impeller eye area.

If the inlet gas velocity is referenced to the local acoustic velocity, the resulting value is named the inlet relative Mach number, M_r . It is apparent that should sonic flow conditions occur, they will initially be observed in the shroud area of the impeller eye. Analyses and experimental investigations have confirmed that Mach numbers of unity will initially be experienced around the leading edge of the impeller vane at the shroud. The presence of sonic flow conditions is usually accompanied by some form of shock structure or region of flow separation that represents a local increase of entropy and losses, effectively reducing the efficiency of the impeller. Obviously, as the volumetric flow rate is increased more, additional portions of the inlet eye will reach sonic conditions with total choked flow conditions ultimately existing across the eye area of the impeller.

Unfortunately, information necessary to establish the conditions at the impeller eye, and particularly the inlet relative Mach number, is typically only available to the equipment designer. Additionally, the methods and assumptions utilized to calculate these conditions are



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subject to variations and not consistent among equipment suppliers. This normally results in the inlet relative Mach number being supplied by the equipment supplier with a reasonable amount of uncertainty associated with its value. It is also important to confirm that the Mach number being supplied by the equipment supplier is calculated at the eye shroud and not elsewhere in the eye region. The impeller inlet relative Mach number represents another limiting critical performance parameter that is related to high flow, high molecular weight refrigeration compressor applications. An industry consistent method of calculating and reporting this parameter does not currently exist and remains a source of considerable confusion.

Acoustic Specific Speed

Although the methods and parameters introduced thus far allow more detailed evaluation and independent selection of staging for a majority of compression applications, some high molecular weight, high volumetric flow rate examples may not be adequately addressed with these parameters alone. This is the evolving case with LNG liquefaction refrigeration compressors where train refrigerant mass flow and corresponding volumetric flow rates have continued to grow with increasing process train design capacities. These increases in inlet volumetric flow have pushed flow coefficients to higher levels and from radial to mixed flow stage designs. Inlet relative Mach numbers have also been realized that are beyond previous experience limits.

Wislicenus (1965, 1974, 1986) proposed another similarity parameter for compressible flow turbomachines that he compared to the suction specific speed that has been widely adopted to predict suction cavitation conditions in centrifugal pump applications. He referred to this additional similarity parameter as the compressibility specific speed. Casey, et al. (2010) also mentioned this similarity parameter in their evaluation of mixed flow compressor designs, but referred to it as the acoustic specific speed. The equation for the acoustic specific speed using both dimensionless and dimensional parameters is given as:

$$na = \sqrt{\pi} \sqrt{\phi} Mn^{1.5} \approx \frac{N \sqrt{Q_{sa}}}{a_s^{3/2}} \quad (20)$$

The dimensionless form of the acoustic specific speed is a function of the flow coefficient and the machine (peripheral) Mach number. Its relevance becomes obvious when it is compared to the inlet relative Mach number reported for a number of actual stages in refrigeration applications. Figure 37 provides such a plot.



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**Impeller Inlet Relative Mach Number vs.
 Dimensionless Acoustic Specific Speed**

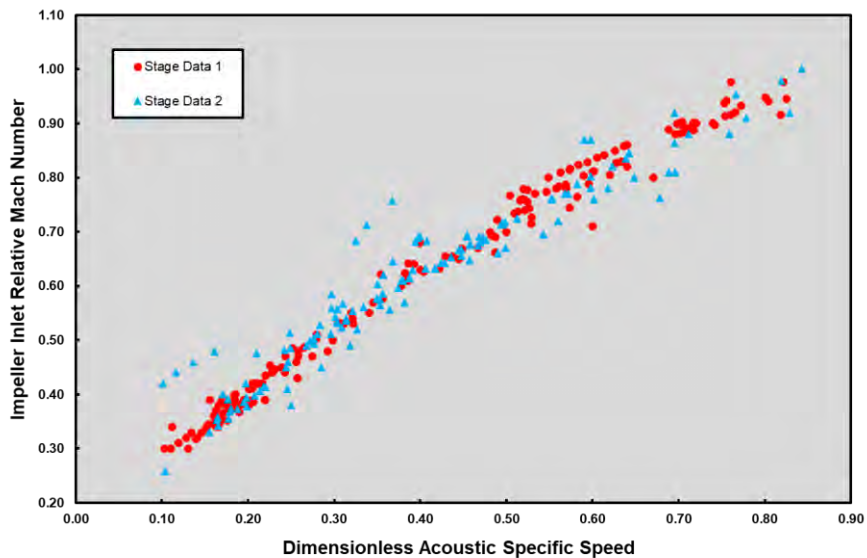


Figure 37. Inlet Relative Mach Number versus Acoustic Specific Speed.

A trend of increasing inlet relative Mach number with acoustic specific speed is evident with some amount of scatter present. Some or all of this scatter may be attributed to variations in calculation methods to obtain the inlet relative Mach number, differences in the location where the inlet relative Mach number is defined, and geometric differences between impeller designs. Meher-Homji, et al. (2007) defined a comparable relationship between flow coefficient and the inlet relative-to-machine Mach number ratio (stiffness ratio). The benefit of utilizing the acoustic specific speed over the stiffness ratio is that both independent variables of flow coefficient and machine Mach number are included in the prediction of the inlet relative Mach number and the magnitudes of their respective influence is readily separated. These two independent variables are typically provided or readily calculated given the amount of information normally provided on a compressor data sheet.

As stated previously, the inlet relative Mach number represents another important performance parameter that should be included and evaluated for high flow coefficient, high molecular weight applications along with the other similarity parameters identified above. Evaluation of different selections and design is made possible using these parameters. A topic that deserves further investigation, though, is the limiting value or possible range of values of inlet relative Mach number that result in predictable and continuously stable compressor performance. Wislicenus (1965) theorized an analogous relationship between acoustic specific speed and pump suction specific speed where he stated that pump suction specific speed provided limits where negative impacts to pump performance by changes in suction conditions (cavitation) could be predicted. Comparable limits likely exist for the acoustic specific speed. An inlet relative Mach number of unity relates to the possibility of increased losses and performance impacts in impellers due to the potential presence of shock structures and/or regions of flow separation, however, there is no existing guideline that establishes a limit or range of inlet relative Mach numbers where impacts on compressor performance become significant.

A further illustration of the potential utility of the acoustic specific speed is demonstrated when selected constant values of this parameter are superimposed on an experience plot produced by an LNG licensor. This is provided in Figure 38.



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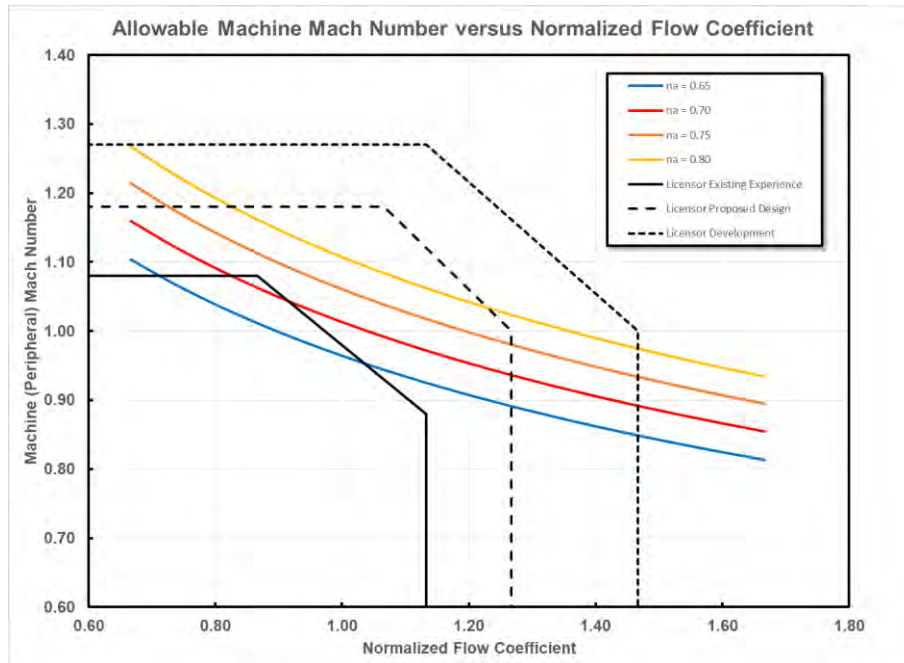


Figure 38. Experience Plot of Flow Coefficient versus Machine Mach Number.

Due to the lack of familiarity and use of the acoustic specific speed, a value of 0.7 is proposed as a maximum limit where potential applications beyond this limit should initiate additional actions that are intended to ensure acceptable performance is achievable. An acoustic specific speed value of 0.7 corresponds to an inlet relative Mach number in the range of 0.9 across a typical range of flow coefficients for current refrigeration compressor applications. Above this limit, an equipment supplier should be expected to demonstrate its ability to satisfactorily design and predict performance through a number of means including proof of previous experience, additional analysis such as computational fluid dynamic (CFD) stage modeling, individual impeller/stage full scale or model testing, and enhanced factory acceptance performance testing. Should the acoustic specific speed become a more adopted and commonly used similarity parameter, it is expected that further refinement of limiting values, performance impacts and necessary due diligence verifications will be documented and better understood.

ADDITIONAL PERFORMANCE AND MECHANICAL DESIGN RELATED ISSUES

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. Rather than providing firm limits and dictates to the compressor vendor, the better approach is to examine the fit of the machinery to the process and examine the novelty in terms of vendor compressor experience. In all cases design compromises are needed and interactive sessions with the operating company, compressor vendor, EPC contractor and process licensor are the best way to rapidly resolve issues. Current compressor technology is covered further by Pellagoti and Baldassarre, (2014), and by Ghizawi et al., (2012), and Valente, et al., (2018). A valuable discussion of the tradeoffs involved in compressor design is also provided by Sorokes (2003).

The increasing trend of refrigerant mass flow rate and accompanying compressor volumetric flow rates are being driven by higher



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capacity LNG process train designs and the economy of scale they provide. Impeller diameters have become proportionally larger due to the desire to accommodate the increased volumetric flow rates and maintain restraint to the degree possible on the increase in flow coefficient and machine Mach number (tip speed). This has resulted in impacts to mechanical design, manufacturability, and material handling for the equipment suppliers that often require compromise. As noted previously, the high machine Mach number has the following effects:

- Reducing the operating range of the compressor (surge-to-choke margin)
- Flattening of the curve shape

In almost all LNG processes, the propane compressor represents the major challenge especially if a single casing design is desired (single entry). Generally, the following criteria should be considered in the design and selection of a LNG refrigeration compressor, particularly the primary refrigerant compressor (e.g. propane compressor):

- Compressor operability at site ambient temperature low and high temperature limits
- Design point margin to predicted surge flow
- Design point margin to predicted stonewall (or end of curve flow)
- Operating point and equivalent ambient temperature where anti-surge control line is reached and partial recycle is initiated
- Low ambient temperature where operation at stonewall is reached (represents maximum achievable flow rate)
- Compressor speed range afforded by connected driver
- Equivalent high ambient temperature where available driver power equals compressor absorbed power in partial recycle (higher ambient temperatures result in bog-down and ultimate shutdown of compressor train)

End of Curve Operation

The end-of-curve capacity index (EOC), is a function of compressor aerodynamic design, and indicates centrifugal compressor capability to operate at higher than design flow. The end-of-curve, which is defined on the compressor design speed curve, can be calculated using the equation shown below.

$$EOC = \left(\frac{Q_{EOC}}{Q_{DP}} \right) \cdot 100 \quad (21)$$

where: Q_{EOC} = volumetric flow at end of curve
 Q_{DP} = volumetric flow at design point

Recently, operation in deep choke has gained some attention as it is possible to have impeller damage due to extended operation in this region. This can be very detrimental on high pressure compressors or high molecular weight machines. Anti-choke protection is available from surge control systems manufactures that will not allow operation to the right of the defined performance curve. It is not recommended that compressors be operated outside their defined map envelope. While the surge line is considered as a firm limit line, where it is commonly known that operating in surge is dangerous, operating outside the performance map at higher flows in or near choke is often considered “acceptable” by operators. Apart from the associated loss in efficiency, there are potential mechanical risks in operating in overload related to impeller vibration and fatigue failures. Sorokes et al (2006) and Kushner et al (2000, 2004) have addressed this issue.

Pressure (Head) Rise to Surge

A positive continuous pressure (head) rise to the compressor surge point at constant speed is crucial to stable compressor operation and reliable, responsive process control. The achievable pressure rise to surge is directly influenced by the required compressor operating range. The pressure rise to surge (PRTS) is a measure of the rise in the pressure (at constant speed) at the surge line flow relative to the design point flow. The PRTS is a function of compressor aerodynamic design and is influenced by the flatness of the performance curve at lower than design flow. It is derived from the compressor design speed curve and its value in percent can be



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calculated using the equation shown below.

$$PRTS = \left(\frac{P_{d,SRG}}{P_{d,DP}} - 1 \right) \cdot 100 \quad (22)$$

where: $P_{d,SRG}$ = compressor discharge pressure at surge flow
 $P_{d,DP}$ = compressor discharge pressure at design flow

Other common terms related to PRTS include head rise to surge and pressure ratio rise to surge. In some cases, specifying a hard number such as 5% may result in a reduced design point efficiency and a reduced “overload” capacity as the vendor may shift the curve and sacrifice efficiency to meet the PRTS criterion. Selecting a lower PRTS may result in a higher design point efficiency and an increased operating margin.

The refrigeration process requires operation over a relatively narrow range of discharge pressures due to the condensing temperature. Either the driver speed or suction pressure must reduce as flow is reduced towards surge. Reduction of suction pressure below atmospheric pressure is usually not desirable. Due to the fact that the surge to stonewall/choke flow range for refrigeration service is less than that seen on other types of applications, the imposition of an excessive PRTS and/or surge margin criteria may result in only minimal overload capacity. A recent application has shown a compressor working well with PRTS below 2% on the guarantee point and lower on alternate operating points.

Operating Stability Margin

Operating stability margin (OS), Taher and Meher-Homji (2012), which is a function of compressor aerodynamic design, indicates how far away the design point (assumed peak efficiency point) is from the surge point on the compressor design speed curve. The equation below can be used to calculate the operating stability in percent.

$$OS = \left(1 - \frac{Q_{SRG}}{Q_{DP}} \right) \cdot 100 \quad (23)$$

where: Q_{SRG} = volumetric flow rate at surge
 Q_{DP} = volumetric flow rate at design point

Operating stability margin can be used as an index to determine the turndown capability of a centrifugal compressor. For multistage centrifugal compressors that operate at high machine Mach numbers, a high operating stability may be achieved at the cost of lower efficiency at design point. On the other hand, a high efficiency at the design point may lead to a narrower operating stability and may result in recycle at lower operating capacities to avoid surge.

Some suggested compressor selection guidelines are provided in Table 4. These are recommendations and not firm limits and should be discussed with the compressor supplier on a case by case basis depending on project specifics and constraints.



Table 4. Suggested Compressor Selection Criteria Guidelines.

Parameter	Selection Criteria
Impeller Inlet Relative Mach Number	0.90 (measured at shroud and based upon a 2D calculation)
Maximum Design Point Flow Coefficient	0.14 (parallel casings) / 0.17 (single casing)
End of Curve Capacity Index (EOC is defined as end of supplier performance curve or 85% of design or rated head, whichever is lower)	105% from Design Point or 103% from Rated Operating Points
Minimum Stability Margin (SM)	20% if PRTS < 10% or 10% if PRTS > 10%
Pressure Rise to Surge (PRTS)	5% for Design Point 3% for Rated Points

Important Note: These guidelines are provided as an initial target. All criteria should be open for discussion with the compressor supplier and based on operating experience, test data, and project specific constraints. It is likely to be very difficult to satisfy all criteria for challenging refrigeration compressor applications.

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 or 230.3 inches). In general, these large rotors are quite rigid and have good separation margins. API 617, 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. It is important to note that the use of 3D impellers with large flow coefficients coupled with a large number of sidestreams results in a longer rotor which could exceed the OEMs comfort level in terms of L/D ratio. For LNG compressors, a detailed rotordynamic analysis is often done in the FEED stage to ensure that there are no lateral or stability concerns.

Casing, Piping and Nozzle Considerations

Larger flow rates and impeller diameters increase the physical size of casings to contain the larger impellers and stationary flow components. The increased flow rates also require larger diameter, potentially non-standard nozzles to be incorporated into the casing designs and associated process piping.

Compressor nozzle velocities should be carefully reviewed. For new compressors, nozzle velocities should be limited to 34-37 m/sec (110-120 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

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.



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These larger casing sizes also demand the ability of equipment suppliers to be able to manufacture, machine and handle larger both unfinished components (castings, forgings, etc.) and finished parts and assemblies. The ability to transport and install these larger pieces of equipment also impact the design and construction of the facilities where they are involved.

COMPRESSOR DRIVER SELECTION

While this tutorial focuses on compressor selection, the drivers connected to the refrigeration compressor trains may significantly impact compressor selection. A limited number of different driver types have been employed for LNG main refrigeration compressor drives, including steam turbines, gas turbines (heavy duty and aero-derivative), and electric motors (synchronous and induction). Steam turbines and electric motors have also been employed in starter/helper service to assist gas turbine start-ups and supplement power output under high ambient temperature conditions. The following driver characteristics will likely have influence on the selection and design of the compressors.

- **Output Speed:** The output speed of the driver is very important as it will likely dictate the impeller diameters and frame size of the compressor and may define if a speed change is warranted or possible. Although output speed may be varied with the potential application of a gear, this should be considered carefully due to the limited experience with the use of gears (particularly speed increasing gears) at the transmitted power levels necessary for many large scale LNG refrigeration applications. In general, the higher the molecular weight of the gas handled by the compressor, the lower the optimum rotational speed of the compressor.
- **Speed Range:** The ability to vary driver speed impacts the operability and flexibility of the process. Most single shaft gas turbines have a speed range of only 95-101% while most two shaft gas turbines, including aero-derivatives, have speed ranges of approximately 70-105%. Steam turbines also may be designed with various output speeds with considerable speed ranges. Electric motors will generally only be available in limited, discrete output speeds, but significant speed ranges may be made available with the inclusion of a variable frequency drive (VFD).
- **Start-up Torque Capability:** Most single shaft gas turbines cannot start up a LNG compressor string from settle out pressure or for that matter even with reduced pressure without some supplemental starting means. Consequently, every time a startup is done, depressurization must be accomplished. In comparison, steam turbines, electric motors with VFDs, and two-shaft gas turbines can typically start up a compression string from settle-out pressure or a compressor loop with only partial depressurization. The ability of a driver to supply start-up torque to an entire compression train must be considered and incorporated into the system design.
- **Overall Thermal Efficiency:** Thermal efficiency of the main compression drivers has a major impact on the LNG plant auto consumption. This must be evaluated carefully for each driver type. The thermal efficiency of gas turbines is relatively straightforward and directly related to the thermal efficiency of the gas turbine, whether in single or combined cycle configuration. Steam turbine drive systems must include the thermal efficiency of the turbines and the steam generation system. Electric motor drives should include the efficiency of the motor, any associated VFD, and the source of electric power generation itself, which may be purchased from an existing grid or self-generated.
- **Maintenance Strategy:** Maintenance strategies and intervals for the different driver types vary significantly. Electric motors are generally considered to be highly reliable with little or no planned maintenance practices, however, any power generation or VFD maintenance requirements should be included in any maintenance plan and budget. Steam turbines will require some periodic maintenance and inspection, but the associated steam generation system will likely dictate the maintenance program for this driver type. Gas turbines usually dictate the most specific and detailed maintenance strategy of all the different driver types. Well defined gas turbine engine maintenance intervals and maintenance philosophies (e.g. site overhaul, engine rotation, lease engine program) can have significant influence on LNG plant turnarounds and impact project economics. Any driver maintenance strategy must be harmonized with required maintenance activities with other equipment included in an LNG plant.
- **Fuel Gas Pressure:** Required fuel gas pressure level is a function of the gas turbine compression pressure ratio, whether they are used as prime movers or electrical power generation. Fuel gas pressure may impact the liquefaction system design in terms of boil-off gas (BOG) compressor pressure ratio, end flash gas (EFG) compressor pressure ratio or in some processes, when extraction of feed gas from the liquefaction train is required to provide fuel gas.



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Steam Turbines

Steam turbines were utilized as prime movers in some of the earlier LNG plant designs but are not common in more recent, larger capacity LNG plant designs. One exception is a recent floating LNG (FLNG) plant design that has incorporated steam turbines as main refrigerant compressor drivers as a result of safety considerations. A major drawback of the application of steam turbines is associated with the design and operation of the steam generation system and related equipment (water supply and quality).

Gas Turbines

Gas turbines represent the most predominant type of LNG refrigeration compressor drivers currently employed in LNG plants. The evolution of LNG plant designs and individual train capacities has been significantly influenced by the development and application of different gas turbine design xxx and output power limits. Common gas turbine designs employed as LNG compressor drivers may be classified as:

- Single-shaft, heavy duty (industrial): Typical engines are the Frame 5001, Frame 6001B, Frame 7001EA or Frame 9001EA. Single-shaft gas turbines have been primarily utilized due to their relatively high available power ratings as compared to other designs. Initially developed for electric power generation, they are characterized as having a very narrow speed range and limited start-up torque capability. They are normally supplied with some type of starter/helper drive and require partial or total blowdown of the connected refrigerant compressors for starting. Installations include both simple cycle and combined cycle, where the simple cycle efficiency is increased closer to the higher aero-derivative simple cycle efficiencies.
- Two-shaft, heavy duty (industrial): Typical engines are Frame 5002, the SGT 750, and more recent introduction of the MHI H-110. Two-shaft, heavy duty engines have been incorporated into a select number of LNG plant designs. Their designs offer a wider speed range for improved compressor capacity control and eliminate the need for complete compressor blowdown for starting. Until recently, though, they have only been available in limited power capacities which has diminished their application in the evolving larger LNG train capacities. The MHI H-110 represents a significant evolution of two-shaft power output and has recently been announced as the selected compressor driver for an LNG design under development. Their simple cycle thermal efficiencies are comparable to the heavy duty, single-shaft engines with lower compression ratios and firing temperatures.
- Aero-derivative: Typical engines include the LM2500 family, the LM6000 family, the RB-211 and the Trent 60. Aero-derivative gas turbine engines have become increasingly popular in LNG plant designs due to their higher simple cycle efficiencies, variable output speed capabilities, and more flexible maintenance requirements. Although more limited in power output capabilities than their heavy duty, single-shaft counterparts, they have found increased acceptance when installed in parallel compressor train arrangements. These higher compression ratio, higher firing temperature engines represent the application of higher technology components and require higher fuel gas supply pressures which must be accommodated in the plant designs.
- Large hybrid: An example is the LMS-100PB. This type of engine incorporates both heavy duty and aero-derivative gas turbine engine components. Its thermal efficiency is comparable to smaller aero-derivative engines and includes a separate power turbine which allows variable speed operation and start-up from compressor settle-out conditions. A current LNG project in development has announced that it will utilize this engine for the main refrigeration compressor drives.

Some more detailed information on selected LNG gas turbine drivers (both existing and potential) is provided in Table 5 below.



Table 5. Selected Gas Turbine Drivers for LNG Liquefaction.

Gas Turbine Model	Engine Type	ISO Rated Power (kW)	Thermal Efficiency (%)	Engine Compression Ratio	Air Flow Rate (kg/sec)	Number of Shafts	Free Power Turbine (Y or N)	Output Shaft Speed (rpm)
PGT 25+G4	A	34,328	41.3	24.4	197	2	Y	6100 / 3600
PGT 25+G5 (G4 Growth)	A	38,000	41.3	24.4	-	2	Y	6100
LM 6000 PF	A	44,261	42.8	29.0	275	2	N	3600
LM 6000 PF+	A	54,000	42.7	32.0	319	2	N	3930
LM 9000	A	65,000	43.0	33.0	395	2	Y	3429
SGT A65 (Trent 60)	A	54,200	43.6	34.3	340	3	N	3430
Frame 7001 EA.03	HD	93,000	33.0	12.6	296	1	N	3600
Frame 9001E	HD	130,142	34.6	12.7	418	1	N	3000
LMS 100 PB+	HY	110,000	44.4	42.7	494	3	Y	3428
MHPS H-110	HD	121,000	38.9	20.0	695	2	Y	3000

Engine Type – A: Aero-derivative, HD: Heavy Duty, HY: Hybrid

A plot of thermal efficiency vs. specific work for a large population of aero-derivative and industrial engines is shown in Figure 39. In this figure, the yellow triangles indicate aero-derivative engines. Aero-derivative gas turbines achieve significantly higher thermal efficiencies than industrial gas turbines that are commonly used in LNG service currently. Commonly used engines in the LNG liquefaction process are included in the ovals. 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 because of the ability to completely change out a gas turbine gas generator (or even a complete turbine) within 48 hours versus 14 or more days that would be required for a major overhaul of an industrial gas turbine.

It is often said that there is a convergence of gas turbine technology between industrial and aero-derivative engines. This may be true if advanced industrial turbines are being considered, but in examining the common industrial gas turbines used in LNG service (Frame 6001B, Frame 7001EA, and Frame 9001E), these E class machines tend to fall into the “older” industrial gas turbine zone shown above, and consequently operate at significantly lower pressure ratios and turbine inlet temperatures and consequently at lower thermal efficiencies.

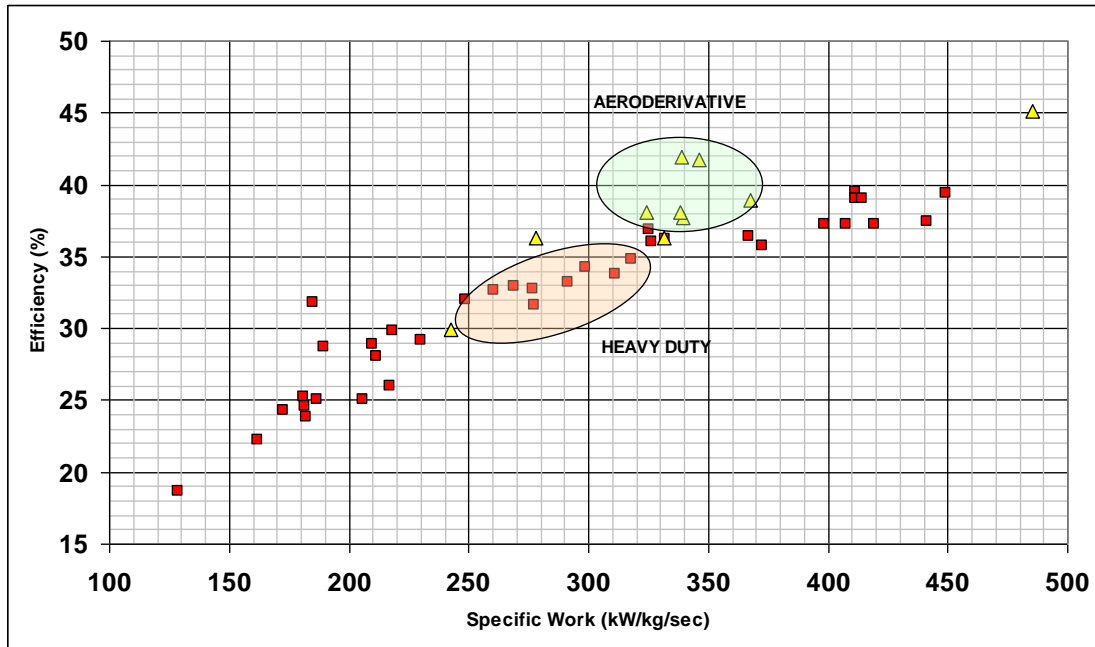


Figure 39. Map of ISO Thermal Efficiency vs. Gas Turbine Specific Work.

The area of gas turbine performance degradation is important when sizing the compressor as a realistic value of fouling and degradation should be used as opposed to traditionally high values which can result in off design operation of the compressor in reality. The area of performance deterioration is covered by Kurz et al., (2003) and Kurz et al., (2013). Krishnamurthy et al., (2015) provides a methodology for selection of drivers for LNG applications.

Fuel considerations and fuel flexibility is an important requirement for LNG drivers as they have to take changes in the nitrogen content of the fuel gas and changes over time to accommodate rapid changes in fuel due heating value (Wobbe index) to process changes or transients and during startup operations. Additional information on gas turbine combustion stability may be found in Meher-Homji et al., (2010).

Details on the application of aero-derivative engines in LNG service is provided in Meher-Homji et al., (2007, 2009, 2011), Montgomery, (2001) and Patwardhan et al., (2016). Operational experience with aero-derivative gas turbines is provided by Weatherwax and Shah (2019). Future trends in LNG gas turbines and compressors are addressed in Pelagotti, 2017. Finally, more information on the matching of gas turbines and compressors may be found in Taher and Meher-Homji, (2012).

Electric Motors and Drives

Electric motors have become increasingly considered as main refrigerant compressor drivers for LNG service. This has been primarily influenced by advancements associated with variable frequency drive (VFD) technology and its ability to provide variable speed electric motors and more consistent output torque and power across a significant range of motor speeds.

Two basic types of motor designs are available, synchronous and induction. Synchronous motor drives predominate large refrigeration compressor drives installed to date and are also used as starter/helper drives for large, single shaft gas turbines. They are characterized by their operating speed being coincident with the power signal frequency, hence the name synchronous motor. In comparison, induction motors operate at some speed below the power signal frequency that is related to the transmitted torque of the motor that is identified as



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the slip frequency. As the transmitted torque increases, the slip speed and the difference between power signal and motor speed frequency increases. Induction motors have yet to be applied at the same power levels as synchronous motors in LNG service, although some have been used as starter/helper drives.

Variable Frequency Drives

In principle, variable frequency drives accept an alternating current power signal, rectify it into a direct current power source, and then convert it back to an alternating current signal of a different frequency through an inverter. There are two main control architectures of variable frequency drives, current source inverters and voltage source inverters. Each type has its benefits and limitations. Both are capable of modifying the power output signals to subsynchronous or supersynchronous frequencies relative to the input power signal. Due diligence needs to be applied to the mechanical design of the motor to ensure that the applied torque due to lower frequency power signals does not exceed mechanical design limits for expected output power.

Current source inverter (LCI) drives represent the most prevalent and highest power capacity drives. They may be applied to only synchronous motor designs. The largest drives currently in operation are LCI drives with output power ratings of 75 MW, however, larger sizes are possible. A major issue with LCI drives is connected with their production of power signal harmonics that may excite mechanical torsional natural frequencies of driven equipment trains and may also damage electrical components and systems connected to the drive. Installation of electrical filters is normally required to eliminate or attenuate these harmful harmonics and complex torsional analyses are necessary to identify and address any harmonic induced torsional excitations.

Voltage source inverter (PWM) drives may be applied with both synchronous and induction motor designs and are more limited in power output capacity with the current technology. An existing limitation is approximately 25 MW, however, recent developments are claiming transmitted power limits approaching those of LCI drives but have little or no LNG operating experience at these levels. PWM drives are not plagued with power system harmonics that are significant as are present with LCI drives.

PROCESS CONFIGURATION AND IMPACT ON COMPRESSOR TECHNOLOGY

The propane precooling system duty requirements in a large baseload LNG facility represents some conditions that make the compressor aerodynamic and mechanical selection challenging.

Process Duty Conditions That Create Aerodynamic Selection Challenges

In an earlier section of this tutorial, the basic configuration of vapor refrigeration cycles was introduced. As described earlier, the propane chillers/kettles in a propane refrigeration system use latent heat transfer to precool the incoming natural gas and liquefaction refrigerant. Since the chillers/kettles operate at propane bubble point conditions, the 1st stage of the propane system will typically use the lowest propane pressure possible without creating vacuum conditions in the process equipment to allow the lowest practical precooling temperature. With an allowance for pressure drop between the LLP chiller/kettle and the compressor suction, the propane compressor 1st stage suction is the lowest pressure point in the propane system. A typical design constraint for the LLP stage suction on propane systems is a minimum 1st stage suction pressure of 15.95 psia (1.1 bara). Generally, a process simulation will want to optimize the propane system to have the lowest 1st stage suction pressure possible. To accommodate the required propane mass flow for refrigeration, the low suction pressure creates high volumetric flow rates in the first stage impeller. Unlike a straight through compressor arrangement, side streams coming into the propane compressor for the higher pressure levels within the propane system will also have high volumetric flow rates. It is not uncommon for the 2nd, 3rd, or 4th stages of the propane compressor to have more challenging aerodynamics than the 1st stage. In propane compressors, the machine Mach numbers will typically create a practical limit for maximum impeller diameter due to the heavier molecular weight fluid with higher acoustic velocity. Because of this, there is a limit to the head coefficient available in a propane impeller selection. It is not uncommon for propane duty points to be close to the upper allowable limits for machine Mach number. The combination of these design factors creates impellers with high flow coefficients and high machine Mach numbers.



Mechanical Design Aspects That Limit Aerodynamic Selections and Create Mechanical / Rotordynamic Challenges

Propane compressors incorporated into current LNG plant designs are either three or four process section machines, with the most common arrangements having all process sections in a single casing. This requires four to five nozzles on the casing, which represents a significant amount of axial space along the compressor shaft. Given the typical limitations around axial length of the compressor in the frame sizes required for large, baseload propane compressors, the combination of high flow coefficient impellers and the addition of two to three nozzles in the overall length of the compressor limits the number of impellers that can be used in a single casing arrangement. Typical process stages in propane service will have only one impeller per stage. In some cases, a stage may have two impellers (this has been typical for the last propane stage).

A typical propane compressor arrangement for a 4-section propane refrigeration system is shown below in Figure 40.

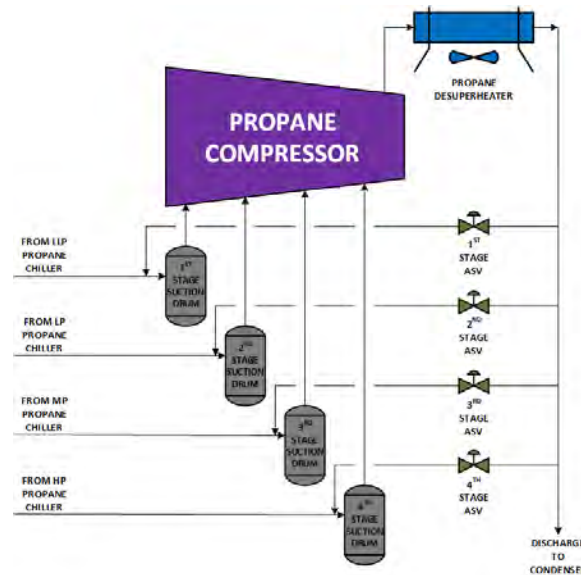


Figure 40. Typical Four-Section Propane Refrigeration Compressor Arrangement.

Compressor Aerodynamic Constraints

For large scale LNG facilities, one of the most common constraints that has limited LNG train capacity has been the size of the refrigeration compressors. The propane compressor has been a service that has required evaluation of aerodynamic capability on several projects. Two key compressor design parameters that were covered in detail in previous sections of this tutorial are machine Mach number and inlet flow coefficient. These parameters can be used to evaluate a set of propane compressor duty conditions relative to available compressor aerodynamic designs. Typical compressor supplier scatter plots showing supplier aerodynamic experience in terms of impeller designs at various machine Mach numbers versus inlet flow coefficients are shown below in Figure 41. An accepted criterion in the past has been to use scatter plots like these to establish aerodynamic limits in the design phase for an LNG process to ensure that the compressor duty conditions fall within reasonable and acceptable design limits for the compressor suppliers. To ensure some margin, a line can be drawn on the scatter plot that represents approximately 95% of the supplier’s experience, and design duty conditions can be selected that are within these accepted design areas.



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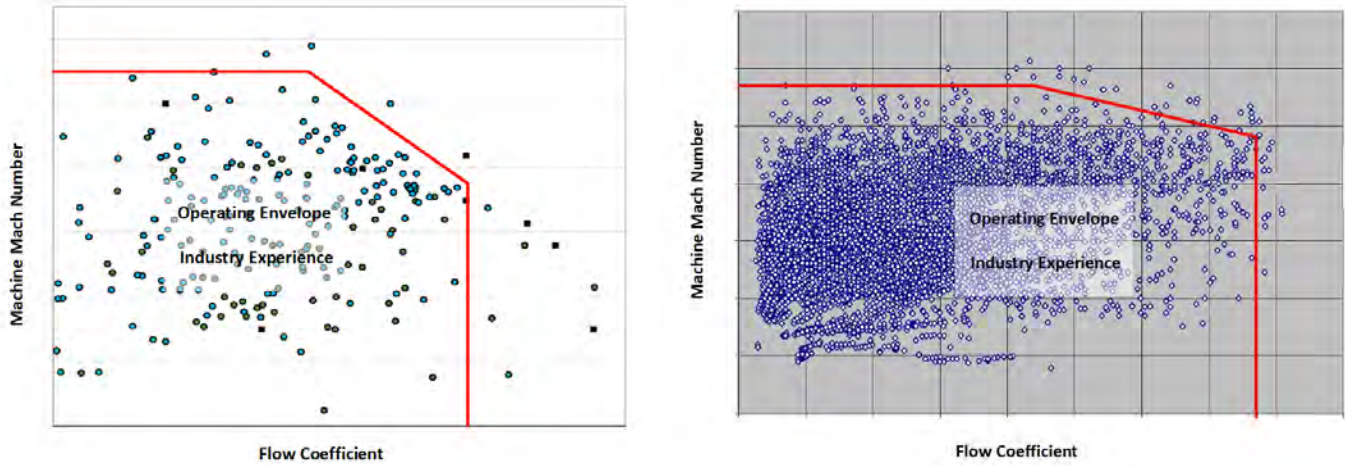


Figure 41. Compressor Supplier Scatter Plots for Impeller Designs.

As LNG production capacity per train is increased, additional refrigeration flow is required. As the compressor volumetric flow increases, the aerodynamics required for the refrigerant compressors to meet the required flow may exceed existing referenced designs or even the capabilities of commercially available refrigerant compressor designs.

Figure 42 below summarizes the general capabilities of the available LNG propane compressor designs relative to LNG train size. For current operating LNG facilities, the operating conditions for the propane compressors have fallen within the well-referenced operating envelope (yellow shaded region) for train sizes up to nominally 5 MTPA with single compression string arrangements. This region roughly corresponds to the aerodynamic design areas shown in the compressor supplier scatter plots above.

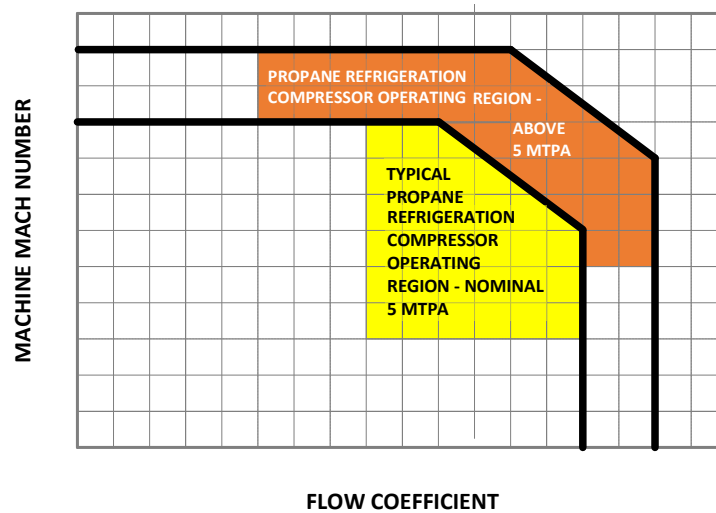


Figure 42. LNG Train Sizing Relative to Compressor Inlet Flow Coefficient and Machine Mach Number.



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As LNG production increases beyond 5 MTPA, the compressor design operating points tend to move beyond the well-reference operating envelope into the red shaded region in the plot above. In these areas, there may be no impeller designs that are referenced in operation, and there may be little or no test data to support the predicted performance of a selected impeller. Looking at the compressor supplier scatter plots again, a second set of lines can be shown that shows this second region as shown in Figure 43.

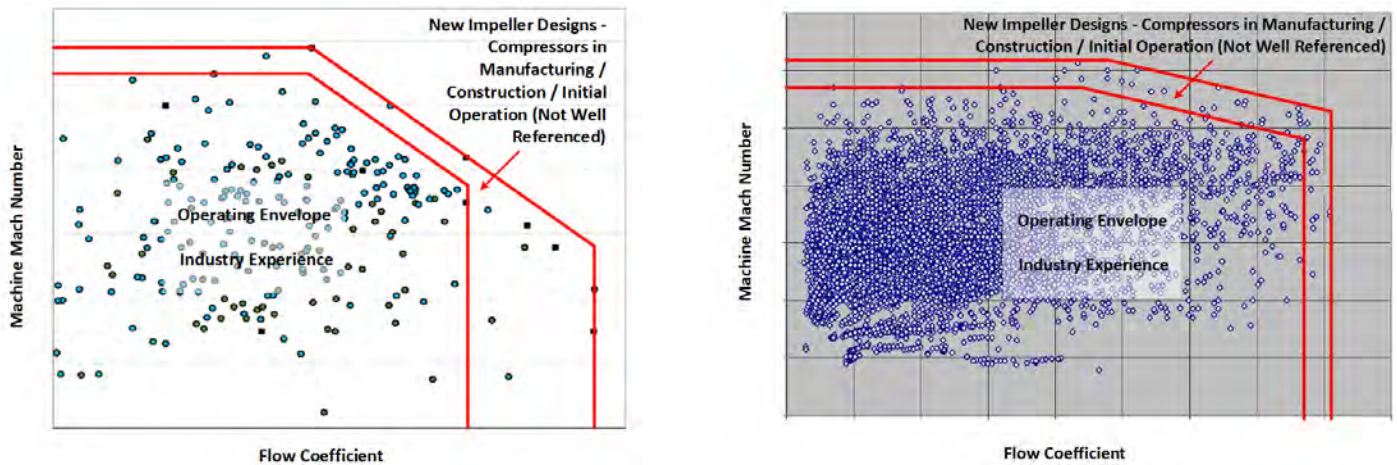


Figure 43. New Impeller Designs Compressor Inlet Flow Coefficient and Machine Mach Number.

Compressor performance in these regions on the compressor scatter plots represent an elevated risk to an LNG facility project, as there is a more significant possibility for the propane compressor to not meet the stated design performance, which can translate to a direct performance penalty for the LNG facility with the potential for higher power consumption per unit of LNG produced and potentially lower overall LNG production. When the size of a propane compression string causes the operating duty to be in these higher risk regions of the compressor scatter plot, there are several options that can be explored. One option can be to develop an alternate compressor arrangement that allows the use of multiple propane compressors in a configuration that lowers the flow coefficients of the impellers and places the compressor performance in a more acceptable, lower risk area within the compressor supplier's experience. If this approach is deemed to be less acceptable and the proposed compressor design is within the design and manufacturing capability of the selected compressor supplier, the compressor process licensor, engineering company, and end user can develop acceptance criteria to allow selection of duty conditions outside of operating or tested references.

Alternate Compressor Arrangements

Series Compression

To accommodate the use of more than one impeller for a single process stage, the propane compressor duty can be split between multiple casings in series. One series option for the propane compressor is to split the last section of a 4-section propane system into a separate casing as shown in Figure 44.

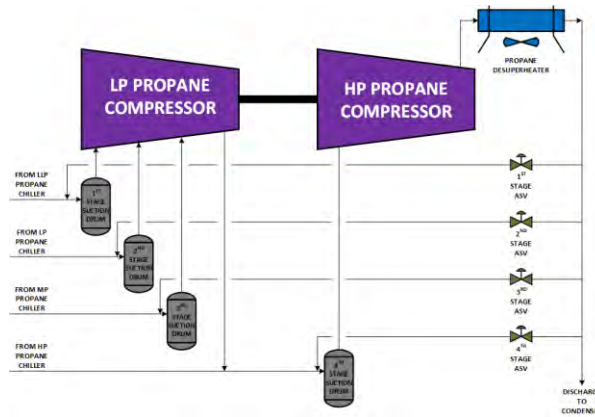


Figure 44. Series Casing Propane Compressor.

Alternate Series Compression 1-4/2-3 Arrangement

Another option for allowing multiple propane compressor casings to be used in series is to use an arrangement referred to as a 1-4/2-3 arrangement as shown in Figure 45. In this configuration, propane stages 1 and 4 are in the first casing and stages 2 and 3 in the second casing. There are some special design consideration for this arrangement. The inlet pressures to the four stages may be different than the single casing compressor design. Also, note that the discharges from the third and fourth stages are at the same pressure since they are connected to a common condenser. The process licensor and compressor supplier can jointly develop this design to maximize the compressor stage efficiency using this arrangement. Unlike the typical single casing configuration, each stage would likely have multiple impellers. The primary benefit of this series arrangement is the minimized complexity of the suction piping.

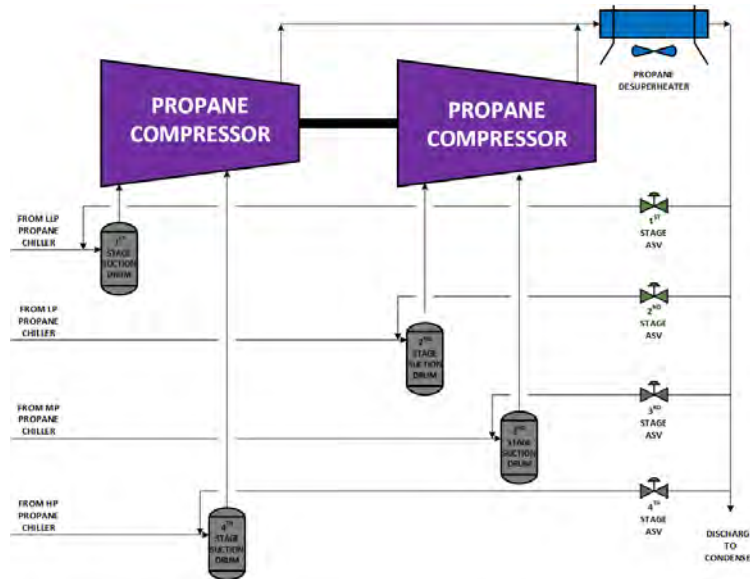


Figure 45. Split Casing 1-4/2-3 Arrangement.



Alternate Double Flow Series Arrangement

Double flow arrangements can also be used as shown in Figure 46, however, as stated previously the 2nd and 3rd sections in some processes may represent higher volumetric flows so this arrangement may not adequately address compressor volumetric flow considerations.

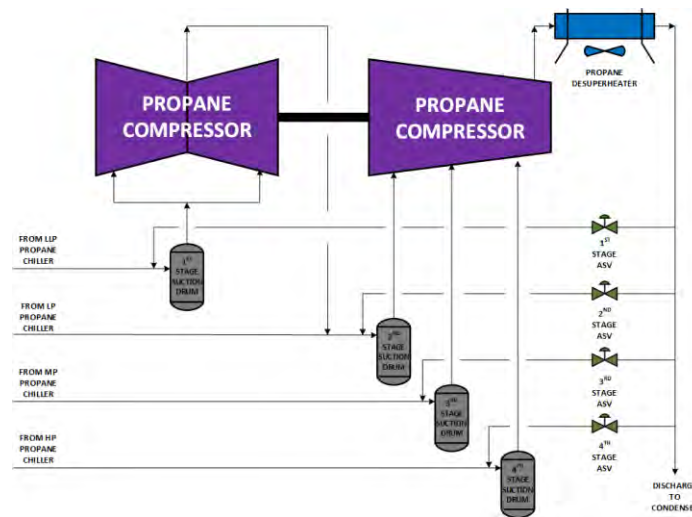


Figure 46. Double Flow Split Casing Arrangement.

Parallel Compression

One method that has gained acceptance has been to utilize parallel compression to allow larger liquefaction train sizes without reaching or exceeding the referenced limits of the compressor suppliers in terms of aerodynamic performance. Several LNG plants are utilizing parallel refrigerant compressor strings to increase liquefaction capacity, improve compressor aerodynamics, or better utilize preferred drivers. A parallel propane compressor arrangement is shown in Figure 47 below.



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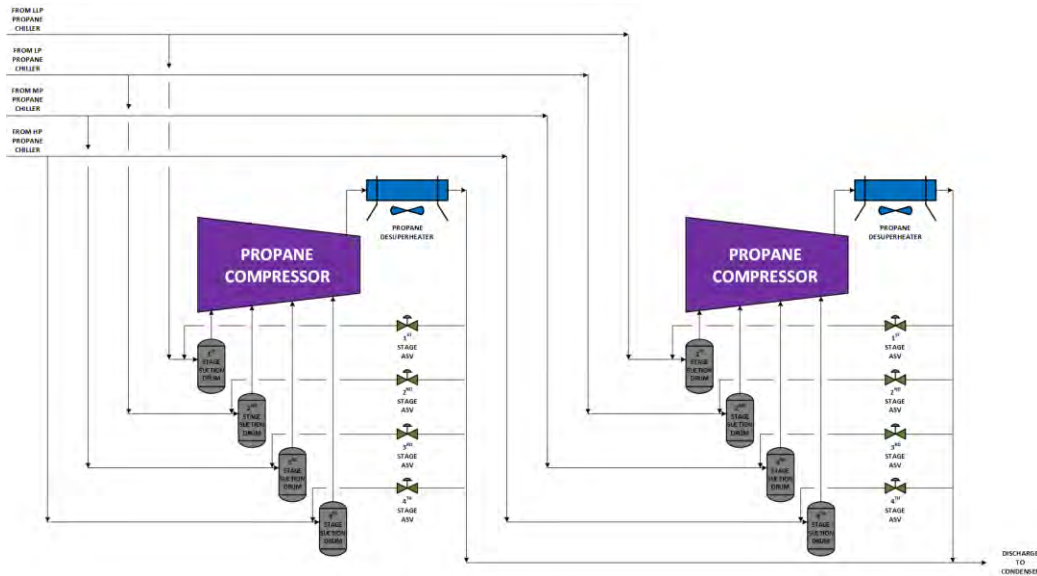


Figure 47. Parallel Compressor Arrangement.

Benefits of parallel compression are detailed below:

- Avoidance of aerodynamic constraints: A parallel compressor arrangement will ideally divide the required flow rate for the liquefaction train in half. This allows a corresponding reduction in impeller flow coefficients of approximately 50 percent as shown in Figure 48. For a typical propane compressor duty, this allows the impeller selections to be shifted to a point in the operating experience of the compressor supplier that has a more referenced operating history and represents a lower overall aerodynamic design risk.



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PROPANE IMPELLER SCATTER PLOT

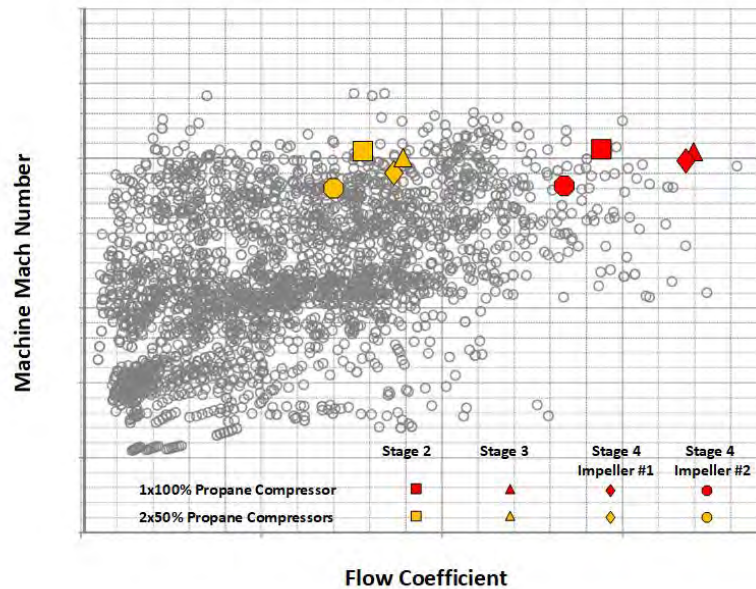


Figure 50. Impact of 2x50% Configuration on Flow Coefficient.

- Wider range of available drivers: In past baseload LNG train designs, the accepted practice has been to use a single steam or gas turbine driver for each of the main refrigeration services. When gas turbines first began seeing use in LNG facilities, the available gas turbine models were heavy duty frame turbines. As the gas turbine market has evolved, there are several options available for smaller gas turbine drivers in the form of either light frame industrial or aero-derivative turbines. These turbines can offer advantages to the operator of a large LNG facility. To be able to utilize these smaller gas turbines in a large LNG facility, several gas turbines must be used in parallel arrangements to facilitate having the required overall liquefaction power while using a smaller gas turbine design.
- Ability to more efficiently turn down train: For a single string compressor arrangement, the turndown flow available on a centrifugal compressor will be limited to approximately 80 percent before the compressor will need to go into recycle. The static equipment in the train (cryogenic heat exchangers, air coolers, etc.) can run at much lower turndown rates than the compressor. Having the ability to run with one of two parallel strings allows the train to run at a nominal 50 to 60 percent turndown rate without needing to have the compressors operate in partial recycle, minimizing power consumption at turndown.
- Ability to perform compressor/driver maintenance on a single string without a full train shutdown: If needed, one of the parallel compression strings can be shut down for planned maintenance while running the train in turndown/reduced capacity. This prevents the need to shut down the entire unit, which avoids several issues:
 - Thermal cycling of the cryogenic heat exchangers
 - Loss of product associated with unit shutdown and subsequent restart of the unit (defrost/cooldown)
 - Loss of refrigerant gas for restarting of compressor strings
- Ability to continue operation through a refrigeration compressor string trip

Drawbacks of parallel compression:

- Additional Mechanical Equipment: When a parallel compression system is selected, additional mechanical equipment is



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required in the train design. In addition to the extra compressor casings associated with parallel designs, there must also be duplicated suction drums and compressor aftercoolers/desuperheaters. More complex piping must also be included to facilitate the splitting and subsequent combination of the flow between multiple compressor casings. Additional mechanical equipment, such as inlet throttle valves and quench valves may be required (these are discussed in more detail below).

- **Control System Complexity:** Additional controls and capabilities are necessary to ensure the refrigerant compressors can tolerate operating conditions unique to parallel equipment. The control system for the compressor strings will need to be designed to provide additional functionality to properly balance the compression load between two or more strings. If a quench system is added, control of this system will also need to be incorporated into the overall compressor control logic. Significant attention must also be given to designing the control system to prevent the trip of one compressor string causing sympathetic trips of subsequent compressor strings and to allow the restart of an offline compressor string while the plant is in operation in parallel compression systems. These control system additions are discussed in more detail below.

Design Considerations for Parallel Compression

When designing a parallel compression system, consideration needs to be given to designing the system to allow for both a trip of one string without a subsequent sympathetic trip of the string that remains on line, and for the start of one string while the second string is on line at normal operating conditions.

1. Quench for Hot Start of Offline String

When an LNG train is in startup, the process only requires one of the parallel compressors to be in operation initially. The second, and potentially, third compressor(s) will not be started until later in the train startup process. When the plant is ready to bring additional parallel compressors online, generally, the operating conditions at that time for the compressor string in operation will be different than the conditions for the strings that are to be started and placed online. These different operating conditions will generally include:

- The offline string(s) will be operating at higher gas temperatures than the online string
- The offline string(s) will have lower discharge pressure than the online string
- For mixed refrigerant services, the offline string will often have a different gas composition

In order to effectively bring parallel compression strings into operation, systems must be in place to allow the offline string(s) to be brought to operating conditions that are close to the online string before attempting to combine the flow of the parallel compressor(s) into the process. One system used for adjusting the operating conditions of an offline compressor string is a quench system.

The quench system takes a small flow of refrigerant from a source in the process where liquid refrigerant of the same composition is present and injects this flow into the inlet of the offline compressor (typically into the compressor recycle piping downstream of the anti-surge valve) through a control valve referred to as a quench valve. This liquid refrigerant source can be from the bottom of an accumulator (single component refrigeration systems, such as propane, or warm mixed refrigerant systems for dual mixed refrigerant plants) or a refrigerant separator drum (mixed refrigerant systems). A typical quench system valve arrangement for a propane compressor is shown below in Figure 51.

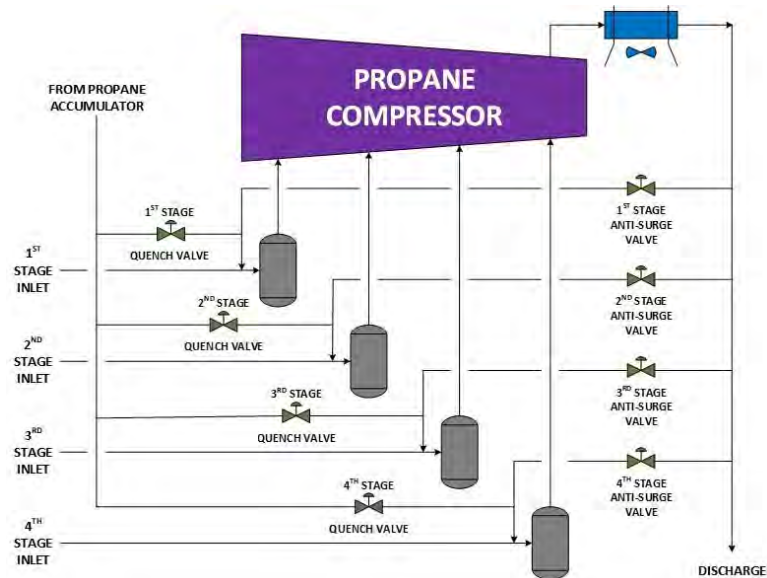


Figure 51. Typical Liquid Quench System Schematic.

The parallel compressor is first started on full recycle to bring a parallel compressor string online. At this point, the gas temperature in the compressor string in recycle will be warmer than the gas temperature in the online string. The quench valve is opened to allow the flow of cold liquid refrigerant into the compressor piping. The liquid flow is distributed in the compressor recycle piping through a nozzle that injects the liquid quench into the stream of hot recycle gas as a fine spray. The evaporative cooling effect of liquid quench lowers the temperature of the gas stream in the offline compressor. If the gas composition of the offline parallel string is different than the gas composition of the refrigerant in the online string (an example being having defrost gas in the offline string of a mixed refrigerant cycle), over time, the quench flow will slowly change the composition of the offline parallel string and allow it to approach the composition of the online string. The quench valve throttling position will typically be temperature controlled by a temperature element in the compressor suction piping downstream of the suction drum. The temperature set point for startup of the parallel string will be slightly above the temperature of the corresponding stage of the online string.

Once the operating conditions of the offline string are close enough to the online string, the offline string can be brought online. The procedure described above is used for both a start of an offline parallel compressor string during unit startup and for restart of an offline string during plant operation due to either a compressor string trip or planned shutdown.

2. Crossover Piping for Balancing Compressor Strings and Minimization of Refrigerant Loss

For certain driver arrangements, when the offline compressor string needs to be started, the pressure in the offline string may need to be reduced to meet the maximum startup torque requirement of the driver. Traditionally in these systems, in order to lower the pressure of the gas in the compressor system, refrigerant gas would be vented to the plant flare system or to a recovery system. When parallel strings are installed in a facility, crossover piping can be installed between the two strings. Crossover piping connects the discharge side of one compressor string to the suction side of the other compressor string. An example of crossover piping arrangement for a three-stage mixed refrigerant compression string is shown below in Figure 52.



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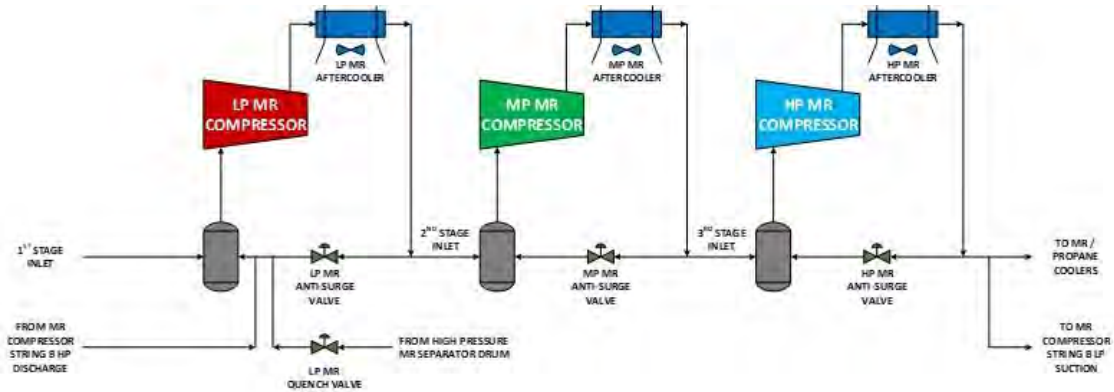


Figure 52. Example of Crossover Piping Arrangement.

The crossover piping isolation valves are opened to allow gas from the discharge side of the offline compressor string to flow into the suction of the online string to use the crossover piping to restart an offline parallel string. The flow of gas from the offline string to the online string vents pressure from the offline string to the online string. The crossover piping arrangement can also be used as an alternate method of cooling an offline parallel string in preparation for tie-in to the online string.

3. Inlet Throttle Valves to Eliminate Sympathetic Trips Due to Gas Turbine Bog Down

One scenario that must be considered in the design of a facility is the response of the system when one parallel compressor string trips during normal operation with another parallel compressor string using gas turbine drivers. A benefit of parallel compression is the ability to continue to operate the LNG facility (at reduced rate) after the trip of one compression string. In order to be able to realize this benefit, the parallel compression system must be designed to prevent a sympathetic trip of the string that remains online after the trip of the first string due to bog-down of the driver.

When one compressor string trips offline, the LNG liquefaction process will typically be operating at full production rate, with the full design flow of refrigerant circulating through the system. At the moment of a compressor string trip, the surviving compressor string will temporarily see a large surge in inlet flow and the liquefaction control system must respond to the sudden loss of a refrigeration string. This sudden increase in flow can be sufficient to cause an overload condition on the gas turbine driver that causes a loss of speed. This condition is referred to as turbine bog-down. Depending on the gas turbine selected for the refrigeration compressors, this bog-down may be severe enough to drop the gas turbine speed below the minimum continuous operating speed for the turbine, causing the surviving compressor string to trip offline soon after the first compressor string trip.

To prevent the sympathetic trip of the surviving compressor string due to gas turbine bog-down, inlet throttle valves can be installed in the suction of the compressor strings. When a compressor string trips offline, the inlet throttle valve on the surviving compressor string can be throttled (partially closed) to control the load on the compressor and prevent the surge of inlet flow that can cause the gas turbine to bog down.

Control Considerations for Parallel Compression

Along with the requirement for additional mechanical equipment, the compressor control system will need to have enhanced functionality to satisfy the needs of a parallel compression arrangement.

1. Load Balancing

Parallel compression systems will be designed with identical compressor strings and efforts will be made to make the



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components of the parallel compressor system external to the compressor (piping, suction drums, aftercoolers, anti-surge valves) as close to identical as possible. However, in operation, systems that are designed to be identical will have small differences and deviations, which will force two compressors in a parallel arrangement to operate at slightly different points on their operating curves. Although this may not be of significant consequence when the facility is running at design conditions, when the compressors are operating close to their surge control lines, it is possible for one compressor to be operating with no recycle, while the other compressor is operating on its surge control line with recycle.

Parallel compression systems use load balancing (also referred to as load sharing), where the control systems of the two compressors are linked together through a master controller, allowing the two compressors to operate at the same distance from surge when not operating on the surge control line. The control system should also balance the recycle flows between the two compressors when operating on the surge control line.

2. Quench Control

The quench system used to start an offline parallel compressor string was described in the section above. In addition to load balancing control, additional controls for the quench system must also be included in the parallel compressor system.

3. Control Valve Tuning and Positioning to Prevent Sympathetic Trips

An additional consideration when designing a parallel compression system is the inclusion of specific control valve positioning and tuning capabilities along with gas turbine speed control to help prevent sympathetic trips of the surviving online compressor string after a parallel compressor trip. The inclusion of an inlet throttling valve as described in the section above to be programmed to move the inlet throttle valves to predetermined positions at the time of a compressor trip to prevent gas turbine bog-down may also be necessary. There are other control actions that can be included in the compressor control system, such as prepositioning of anti-surge recycle valves and automatic speed adjustment of the gas turbine to aid in preventing a sympathetic trip.

The use of dynamic simulation is essential for determining the need for additional mechanical equipment, such as inlet throttle valves, when designing parallel compression systems. Dynamic simulations performed between the process licensor, compressor supplier, and design EPC can also provide a validation of the initial design of the control logic in terms of its ability to handle compressor string trips and restarts and provide initial tuning parameters and valve settings for critical valves such as the anti-surge valves.

Mitigation Criteria for Single String Compressor Arrangements

Should the operator of an LNG facility choose to not utilize an alternate compressor arrangement and accepts a single string propane compressor configuration, one potential risk mitigation step is to utilize an impeller model test with the compressor supplier for any compressor stages that are outside of referenced operating regions or that approach a known design limit for the compressor supplier.

A model test can be performed early in the design and manufacturing of a compressor, allowing the supplier to evaluate the aerodynamic performance of the selected impeller under controlled conditions, prior to the impeller going into manufacturing. If a design revision must be made, the model testing allows the manufacturer to determine mitigating steps prior to having a compressor in a full-scale performance test, where the inability to meet contractual performance obligations can represent either a significant cost and schedule penalty or potentially force an end user to accept compressor performance that is below the guaranteed design conditions, limiting the production capacity of the new LNG facility.

Effect of Ambient Temperature Variation on Compressor Selection and Operation

The pre-cooling refrigerant in a typical baseload LNG facility will be designed to fully condense to liquid phase in the refrigeration loop. This is the design basis for a propane pre-cooled cycle design, such as the AP-C3MR or Conoco Phillips Optimized Cascade



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process. In order to be able to remove the heat of compression and subsequently fully condense the refrigerant, a cooling medium must be used in the plant design. This cooling medium is most often ambient air that is used in the form of air coolers; some plant designs use water cooling in either a once through cooling loop (example being sea water cooling) or an open loop cooling water system with a cooling tower providing evaporative cooling.

A plant will be designed for an average ambient temperature that represents the weighted mean temperature between the anticipated minimum and maximum ambient temperatures. This average ambient temperature will determine the condensing temperature and corresponding pressure of the pre-cooling system. This pressure will set the final discharge design pressure requirements for the compressor. When the compressor design is selected, this design point for the compressor can be selected at a position in the overall compressor operating curve that corresponds to the highest efficiency point at the design condition, which for most compressor curves, centers the operating at approximately the center of the operating curve.

It is normal for the plant to operate at ambient conditions that are different from the design ambient where the operation of the propane compressor must accommodate the change in ambient temperature. This requires the compressor to operate at different points on its operating curves for high and low ambient temperature conditions.

High Ambient Temperature

When an LNG plant is operating at high ambient temperatures, the cooling medium that is available, whether ambient air at dry bulb conditions, ambient sea water, or ambient air at wet bulb conditions used in an evaporative cooling system, will be at a higher temperature. This will cause the coolers within the plant to operate at a higher temperature. Since the propane precooling system requires a full condensing of the propane to liquid phase, condensation at higher temperatures requires higher operating pressures in the propane system. To provide higher operating pressures, the propane compressor must generate higher head at high ambient conditions than at average ambient conditions. The higher head is distributed across all stages of the propane compressor. This requires the compressor to operate at a point on its curve that is closer to surge on each of its compression stages.

Low Ambient Temperature

When an LNG plant is operating at low ambient temperatures, the available cooling medium will be at a lower temperature relative to the design temperature. This will cause the coolers within the plant to operate at a lower temperature. Full condensation of the propane at lower temperatures requires a lower operating pressure in the propane system. To provide lower operating pressures, the propane compressor is required to generate less head at low ambient conditions than at average ambient conditions. The lower head is distributed across all stages of the propane compressor. This requires the compressor to operate at a point on its curve that is closer to stonewall on each of its compression stages.

LNG Train Capacity and Compressor Operating Point

It is likely that the propane compressor will reach either its surge control line (high temperature limit) or its maximum flow at end of curve conditions across the range of possible ambient temperatures which represents a limit in overall LNG train production. When the compressor selection is made, in addition to selecting a reasonable design point within the available curve, attention must be paid to where the compressor will operate at the extents of high and low ambient conditions. This requires a reasonable margin to be selected for both stability margin (surge margin for high ambient operation) and stonewall margin (end of curve margin for low ambient operation). Depending on the compressor selection, developing a design that allows for both high ambient and low ambient operation may represent a challenge, especially for high flow coefficient impeller designs, where the overall range of the impeller/stage may be limited.

A compromise may need to be accepted between high ambient performance and low ambient performance. For a site where high ambient performance is valued above low ambient performance, the operating point may be “shifted” to the right in the operating curve to allow a higher stability margin. This allows for a higher operating range between average ambient and high ambient conditions before the



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compressor reaches its surge control curve and can no longer offer higher head. This may result in two design compromises:

- The compressor may no longer be operating at the highest efficiency point at average ambient conditions, as the selected design point may be to the right of the peak in the efficiency curve.
- Low ambient temperature operation may be limited as the operating point is shifted toward the end of the operating curve (near choke)

Speed Variation

To accommodate the wide range of operating points required of the propane compressor, particularly for LNG facilities that require operation in a large ambient temperature range, driver speed can be used to facilitate additional range in the compressor. The extent of the benefit of using speed control to increase propane compressor range will be limited by several factors, such as:

1. The use of single shaft gas turbines with small adjustable speed ranges (95% - 101%) versus multi-shaft gas turbines with wider speed ranges (70%-105%)
2. The use of electric motor drives or turbine inlet air cooling to reduce the effect of power loss at high ambient temperatures versus gas turbines with no inlet air cooling
3. Multiple services on a single driver (propane on the same driver as mixed refrigerant/ethylene) versus dedicated drivers for individual refrigeration services

Design Considerations for Compressor String Startup

The refrigeration compressor driver selection can have a significant effect on the LNG train startup, particularly for a restart after a trip of the LNG train. Each of the major driver types will have different limitations.

Restart of a Compressor String after a Trip

After a compressor string trip occurs during operation and the compressor has decelerated and has fully stopped, the compressor system will reach a settle-out pressure. The final settle out pressure of a system is based on several design conditions outside the compressor itself, such as piping and vessel volumes on the suction and discharge side of the compressor, location of check valves, and the inclusion of isolation valves within the system.

Single-Shaft Gas Turbines- Starting Considerations

In a single-shaft gas turbine design, the gas generator and power turbine are both on a common shaft. This common shaft is then coupled to the shafts of each of the compressors in the string and the starter/helper motor. To be able to start the compressor string, sufficient torque must be supplied to overcome the inertial load of the entire compressor string (gas turbine rotor, compressor rotors, and helper motor rotor), the load of the axial compressor in the gas turbine, and any gas loads on the compressor rotors. A typical large compressor string in a baseload LNG facility can represent a significant starting torque requirement. For a single-shaft gas turbine, large helper motors are typically required to provide this startup torque. One drawback of a single-shaft gas turbine is that, even with a large helper motor (sometimes as much as 25 MW), there is not sufficient starting torque to allow gas loads that are created at pressures much above ambient pressure. This prevents being able to restart a compressor string from settle-out pressure. In order to be able to restart a compressor string, the gas pressures within the compressors must be reduced to a point low enough where the gas turbine helper motor has sufficient torque to start the string. This typically requires isolation of the compressors from the refrigeration system and venting of the refrigerant from the compressors until a sufficiently low enough pressure is reached to allow a restart. Some LNG facilities have utilized refrigerant recovery systems to capture some of the refrigerant that must be vented from the compressor strings, but for most facilities, venting refrigerant from the compressor systems involves flaring of the refrigerant. This can represent a significant loss for an LNG facility after a unit trip and subsequent restart in terms of both refrigerant loss and additional restart time.



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Multi-Shaft Gas Turbines Starting Considerations

A multi-shaft gas turbine design is split into two or more individual shafts. In the simplest configurations, the gas turbine axial compressor will be located on one shaft, driving a high-pressure power turbine. This shaft is referred to as the gas generator. A second shaft is introduced into the gas turbine, with a low-pressure power turbine. This turbine is referred to as a “free” power turbine, since it can rotate independently of the gas generator. This power turbine is coupled to the shafts of each of the compressors in the string. When the gas turbine is started, the initial torque requirement from the starter motor of the turbine is only based on the inertial load of the gas generator shaft and the load of the gas generator axial compressor. This allows the turbine to start and be brought up to operating speed with a much lower starting torque requirement. This also allows the gas turbine to provide sufficient torque to overcome the gas loads on the process compressors in the string as the gas generator net generated power increases. This can allow the compressor string to be started at pressures that are either closer to settle-out pressure or potentially at full settle-out pressure.

Electric Motor Drives- Starting Considerations

Electric motor drives will normally be provided with a variable frequency drive that facilitates starting the compressor strings at higher pressures close to or at settle-out pressure.

Dry Gas Seal and Support System Design for Refrigeration Compression

Refrigeration systems in LNG facilities are closed loop systems. In the selection of dry gas seals and seal support systems, care must be taken to provide a safe and reliable sealing system that does not provide an inadvertent means of contamination of the refrigerant.

For hydrocarbon refrigerant streams, such as propane, mixed refrigerant, ethylene, or methane, the typical accepted practice has been to use tandem dry gas seals with inter-stage labyrinth seals. This seal design provides a safe and reliable means of sealing the refrigerant gases in the compressor. Typically, with refrigeration loops being generally clean services the seal supply gas is taken from the discharge of the compressor downstream of the compressor aftercooler/de-superheater and upstream of propane/mixed refrigerant condensers.

The use of a double dry gas seal in place of a tandem dry gas seal in a hydrocarbon refrigeration system presents an issue when selecting the barrier gas. Although an inert barrier gas (such as nitrogen) may seem to be the most reasonable choice, use of an inert gas provides a means of contamination of the refrigeration loop. Maintaining refrigerant purity in pure component hydrocarbon systems is critical for the proper operation of the refrigeration loop. With a double dry gas seal, small quantities of barrier gas will leak into the refrigerant gas over time. If a nitrogen barrier gas is used, this small leakage of gas over time will cause a buildup of non-condensable gas in the refrigerant system, which will be seen in the refrigeration loop in the form of higher suction and discharge pressures, higher compressor power, and less overall refrigeration capacity. Once the nitrogen accumulation is enough to affect the refrigeration performance, the non-condensable gas must be removed from the refrigeration loop by venting refrigerant to the flare (typically by venting from the overhead of the accumulator). For a mixed hydrocarbon refrigerant system, even though nitrogen is a typical component of mixed refrigerant, the increase in nitrogen content over a period of time will change the overall refrigerant composition sufficiently to affect liquefaction performance, and the excess nitrogen would need to be vented from the refrigerant loop. The use of process gas for the barrier if a double seal is used is more acceptable in terms of overall performance of the process, as the leakage from the seal into the process would not dilute or contaminate the refrigerant loop. This option would need to be evaluated to determine the feasibility of using pressurized hydrocarbons as the barrier gas. Both single and tandem dry gas seals have been used in nitrogen refrigeration services.

During train startup, it is often necessary to have a source of external seal gas available. There are several sources of external seal gas that can be considered:

- Propane Compressor:
 - Propane from the propane accumulator
 - Nitrogen
- Ethylene Compressor:
 - Ethylene from the ethylene accumulator



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- Nitrogen
- Mixed Refrigerant Compressor / Methane Compressor:
 - Treated (dry, mercury free, sweet) feed gas/defrost gas
 - Nitrogen
- Nitrogen Compressor:
 - Nitrogen (External Source)

In any instance where the startup seal supply gas differs from the refrigerant gas (such as the case of using nitrogen), the duration of time that the startup gas is in service should be minimized to prevent unnecessary refrigeration loop contamination.

CASE STUDY – LNG COMPRESSOR SELECTION

Case Study – Process Optimization to Accommodate Aero Design for Large Propane Compressor

A Pre-FEED feasibility study was conducted to begin development of a process design for a baseload LNG facility with train capacities in excess of 5.0 MTPA. The decision was made by the operating company early in the process to use single strings for the refrigeration compression, requiring large propane and mixed refrigerant compressor selections to meet the requested train capacity. The compressor configuration is shown below in Figure 53.



Figure 53. Split Mixed Refrigerant Refrigeration Train Schematic.

The process licensor developed a process simulation to allow development of preliminary compressor duty specifications. After development of the initial propane duty specifications, the licensor conducted further review of the duty conditions for development of a preliminary compressor selection. The initial compressor duty conditions were placed on an impeller scatter plot from the selected compressor supplier to show where the duty conditions fell relative to the supplier's experience. Two options considered for the initial compressor selection are shown below:

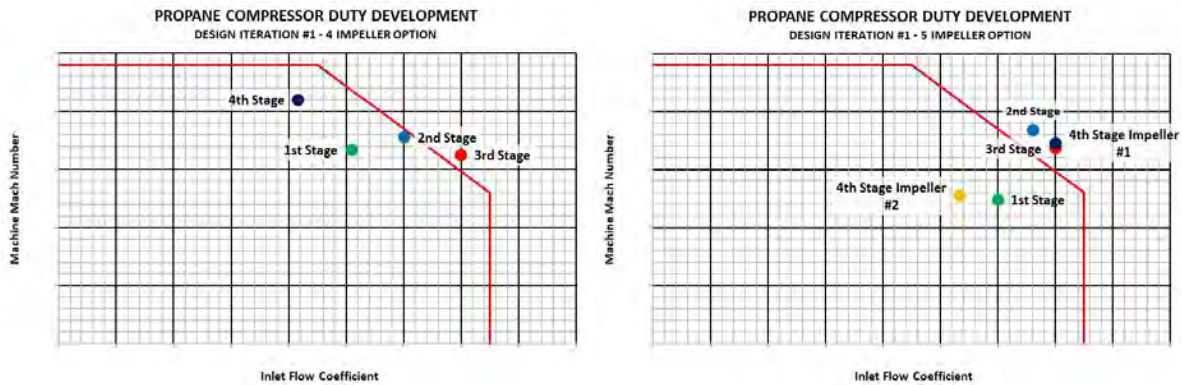


Figure 54. Process Licensor Propane Compressor Initial Staging Evaluation.



A review of the proposed duty conditions against the selected compressor supplier’s referenced operating experience indicated that for both configurations, at least two impellers were at or above the aerodynamic limits established for the supplier. Based on this observation, the compressor duty conditions would not be acceptable to provide to the supplier.

The process licensor, in consultation with the compressor supplier, chose to develop the four-impeller option. The process licensor performed additional simulations to modify the propane duty specifications to allow all four impellers to fall within the aerodynamic design limitations of the compressor supplier. The final iterations, the last one of which was offered to the compressor supplier, are shown below in Figure 55.

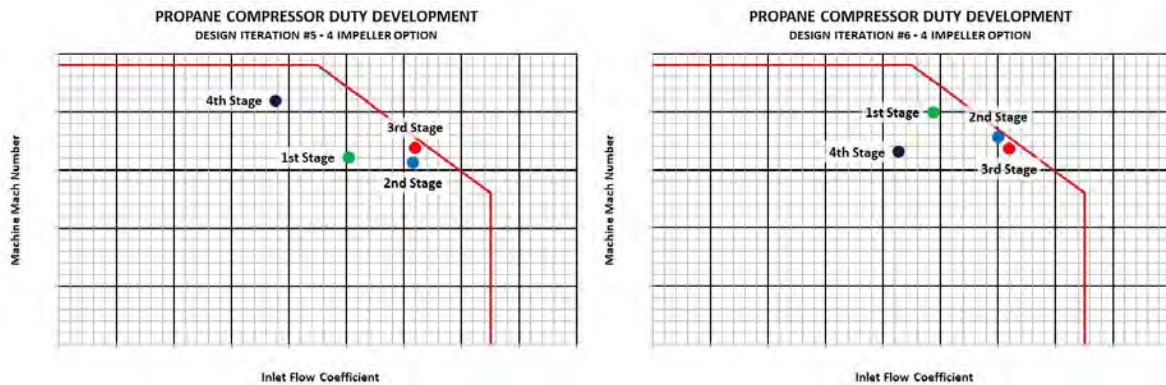


Figure 55. Process Licensor Propane Compressor Final Staging Evaluations.

The compressor supplier used these duty conditions to develop a preliminary selection shown below in Figure 56. With one final iteration between the process licensor and the supplier, a selection was developed for the propane compressor, shown below in Figure 56.

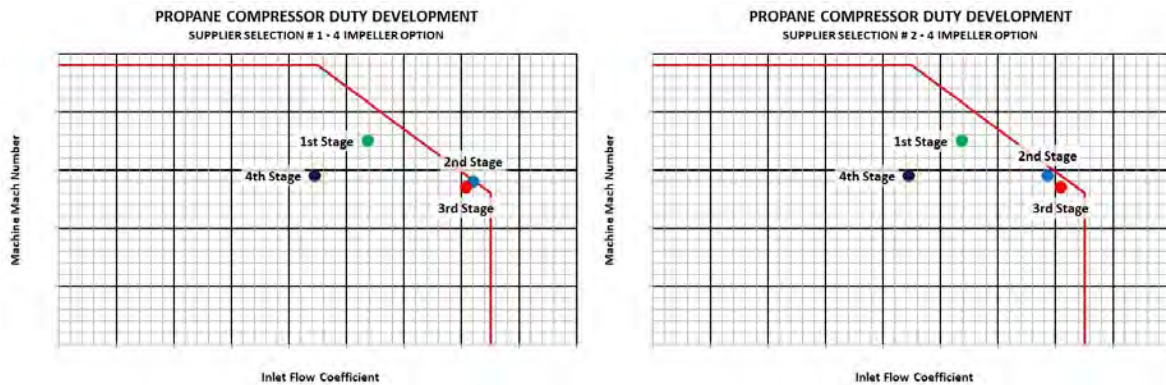


Figure 56. Compressor Supplier Propane Compressor Final Staging Evaluations.

This case study demonstrates that collaboration between the process licensor and the compressor supplier in the early stages of a process development study can allow the process to be optimized for best efficiency while staying within the aerodynamic design constraints of the compressor supplier.



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CONCLUSIONS

There is currently a significant growth in the LNG liquefaction market. One of the most critical components of an LNG liquefaction facility are the refrigeration compressors which have a significant influence on overall plant performance and production efficiency. Refrigeration compressors are challenging to design due to large volume flows, high Mach numbers, low inlet temperatures, and complex side stream flows. Compressor drivers for these facilities include both heavy duty industrial and aero-derivative gas turbines that range in size from 30 MW to 120 MW, and several recently proposed projects are considering the use of large electric motor drivers up to 75 MW. As technology is being pushed to larger train sizes and highly loaded compressors, it is important for end users to understand the technical challenges involved in compressor selection. This tutorial has covered, in a process neutral manner, areas of refrigeration cycles and compressor design. A central theme of this paper is that practical design compromises must be made to obtain a robust compressor solution.

NOMENCLATURE

A_{char}	= Characteristic cross-sectional area
a_s	= Compressor section inlet acoustic velocity
D_2	= Impeller diameter
ds	= Specific diameter, dimensionless form
EOC	= End of Curve index
h_d, h_s	= Compressor section suction and discharge enthalpy, respectively
h_{di}, h_{si}	= Compressor section suction and discharge enthalpy at section i , respectively
h_{ss}, h_{ssi}	= Compressor sidestream flow, compressor sidestream flow at section i
H_p	= Polytropic head (work)
\dot{m}	= Mass flow rate
\dot{m}_i	= Mass flow rate in compressor section i
$\dot{m}_{ss}, \dot{m}_{ssi}$	= Compressor sidestream flow, compressor sidestream flow at section i
M_{ir}	= Compressor inlet relative Mach number
M_n	= Compressor machine Mach number
MW	= Gas molecular weight
n	= Polytropic exponent
N	= Rotational speed
na	= Acoustic specific speed, dimensionless form
ns	= Specific speed, dimensionless form
$nsds$	= Product of dimensionless specific speed and specific diameter
$nsect$	= Number of sections in a compressor casing
OS	= Operational Stability index
$PRTS$	= Pressure Rise to Surge index
P_s, P_d	= Compressor section suction and discharge pressure, respectively
P_{si}, P_{di}	= Compressor section suction and discharge pressure at section i , respectively
P_{ss}, P_{ssi}	= Compressor sidestream pressure and sidestream pressure at section i , respectively
Q_{sa}	= Compressor section inlet volumetric flow rate
T_s	= Compressor section suction temperature
T_{si}, T_{di}	= Compressor suction and discharge temperature at section i , respectively
T_{ss}, T_{ssi}	= Compressor section sidestream temperature and at section i
U_{tip}	= Impeller tip speed
v_s, v_d	= Compressor section suction and discharge specific volume, respectively
Win	= Work input to compressor



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- Z_s = Compressor section suction compressibility factor
- γ = Gas specific heat ratio
- η_p = Polytropic efficiency
- $\eta_{plant\ thermal}$ = Plant overall thermal efficiency
- μ_p = Polytropic head coefficient, dimensionless
- τ = Compressor work input coefficient
- ϕ = Dimensionless flow coefficient
- C_x = Equation unit conversion constants (x is sequential number) as defined below

Equation Unit Conversion Constants

Equation Unit Conversion Constants	Imperial Units	SI Units
Equation Parameter Units	P_s, P_d in psia v_s, v_d in ft ³ /lbm T_s in R H_p, h_s, h_d in ft*lb _f /lbm \dot{m} in lbm/min PWR in hp U_{tip}, a_s in ft/sec N in rev/min D_2 in inches Q_{sa} in ft ³ /min	P_s, P_d in bara v_s, v_d in m ³ /kg T_s in K H_p, h_s, h_d in J/kg \dot{m} in kg/hr PWR in kW U_{tip}, a_s in m/sec N in rev/min D_2 in mm Q_{sa} in m ³ /hr
C ₁	1545.349	8314.472
C ₂	4.3633 x 10 ⁻³	5.2360 x 10 ⁻⁵
C ₃	700.333	6.7548 x 10 ⁶
C ₄	32.174	1.000
C ₅	1.68994 x 10 ⁶	3.64756 x 10 ⁸
C ₆	1.00074 x 10 ⁻³	1.74533 x 10 ⁻³
C ₇	1.5373	0.0600

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