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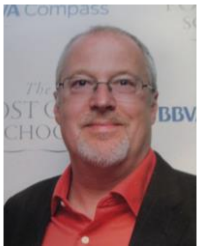
MARINA BAY SANDS

TUTORIAL ON THE APPLICATION AND DESIGN OF INTEGRALLY GEARED COMPRESSORS



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Mr. Bosen is the Executive Chief Engineer at Samsung Techwin with a Master in Mechanical Engineering from Technical University (RWTH) of Aachen, Germany. Mr. Bosen has 37 years in the design, testing and development of centrifugal compressors and radial inflow expanders for a wide range of industrial applications. He has 12 patents on expanders and integrated high-speed motor/generators for turbomachinery. Over the course of his career he has lead the design of numerous product lines and integrally geared compressors from concept to market realization.

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Mr. Pelton holds a B.S. and M.S. in Mechanical Engineering from Brigham Young University. Mr. Pelton worked in the aero/hydro design group at Concepts NREC for eight years where he was responsible for the aerodynamic design and program management of a wide range of turbomachinery design project for systems including radial compressors, pumps and turbines. Mr. Pelton currently is the manager of the development group at the Houston Turbomachinery Design and Development office of Samsung Techwin. In that position, he oversees a team of engineers responsible for the analytic design of new products and custom engineered machines. This includes the aerodynamic, structural, and rotordynamic performance of multi-stage integrally geared compressors.

ABSTRACT

Integrally geared compressors (IGC's) are common in plant/instrument air service as well as air separation applications, and continue to gain acceptance over a wide range of other applications. An IGC can achieve high efficiencies but is subject to complicated mechanical interactions. As a result of the mechanical complexity: design engineers, application engineers, and even end users of IGCs benefit from a diverse and in-depth knowledge of all of the engineering principles applied to arrive at an efficient machine with robust operating characteristics. This paper emphasizes the practical aspects of sizing and selection criteria for an integrally geared compressor for a range of applications and promotes a thorough understanding of practical limits of this type of compressor. Underlying aerodynamic principles are reinforced and limiting design aspects such as: gear tooth loading, lateral rotordynamics, bearing surface speed and loads, low- and high-cycle fatigue of impeller blades are all iterated to find compromises to meet the demands of each application. Understanding the application and applying appropriate design limits is essential to meeting ever more challenging installation requirements.

INTRODUCTION

Objective

The objective of this tutorial is two-fold: 1) to provide the reader with an understanding of the markets served by integrally geared compressors and 2) to provide a reference to the key design considerations required in the development of such compressors. The first step in this process is to understand the evolution of integrally geared compressors from inception to present day. This tutorial illustrates the degree to which a wide range of engineering principles must be integrated and optimized for an integrally geared compressor.

Basic IGC configuration

An integrally geared compressor is comprised of one or more compressor stages attached to the end of one or more high speed pinions. The pinions are mounted in a housing that contains a large low speed bull gear which drives the individual pinions. Figure 1 shows an exploded view of a typical IGC.

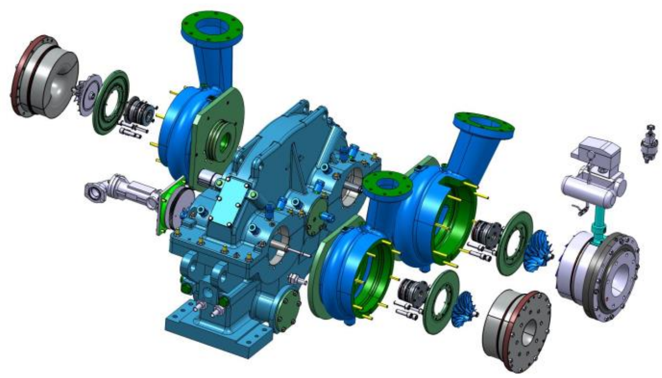

Figure 1. Exploded View of Integrally Geared Compressor

History of Integrally Geared Compressors

Fingerhut, et. al [1991] observes that since the 1950's, integrally geared centrifugal compressors have been in use worldwide, initially in the plant air applications, and increasingly in the process industry. By the use of pinion shafts it becomes possible to achieve high rotational speeds without an external gearbox. This aspect, together with the possible integration of expander stages, leads to a highly economical application of integrally geared centrifugal compressors. Apart from reliability, efficiency and operating range, the most striking features of the integrally geared compressor concept are the versatility and configurability of the machine. The high level of modular construction inherent in the design allows for standardization; allowing designs to be configured using a building block system makes it possible to adapt the machine concept to each field of application and, in particular, to customer-specific requirements.

The integrally geared centrifugal compressor has conquered many ranges of applications in the process industry starting with compressors operated purely on air, followed by nitrogen and then carbon monoxide, carbon dioxide, chlorine, hydrocarbons, acetylenexide, and isobutyric aldehyde, and so forth.

Beaty, et. al. [2000] describes the advantages and risks associated with an integrally geared compressor. In the author's assessment aspects such as: capital cost, installation cost, operating cost, durability, downtime, and maintenance costs are all assessed. The authors list the reduced capital, installation, and operating costs resulting from a compact IGC and the ability of the IGC to have individual pinion rotating speeds to maximize stage efficiency. The change out time for an integrally geared rotor is also relatively short as compared to that of an inline machine. For durability, an in-line compressor is often considered more resistant to process abnormalities and upsets. So for processes where the fluid cleanliness and stability cannot be maintained, in line compressors may be a better choice.

Wehrman, et. al. [2003] reported that the direction of the industrial gas business was such that complex integrally geared compressors were starting to be required. The authors presented a tutorial to give examples on design issues and experiences from two companies that operate and use IGCs. This demand for API617 compliant IGCs originated with the advancements in the design and manufacture of the individual components that improved operational aspects and reliability. This new sophistication included supplying compressors with six to eight stages, potentially with process inlets that compress different gases, and designs that use steam turbine drives. Because of this growing use of compressors in industrial applications, the seventh edition of API 617 [2002] standard for centrifugal compressors, was modified to include a chapter specifically for integrally geared configurations.

Srinivasan [2013] describes the evolution of IGCs entering the process gas industry with several relevant examples. The author describes how multiple rotor speeds from the individual pinions are applied to achieve an aerodynamically efficient design for each stage. The ability to intercool the gas between every stage of compression using inter-coolers benefits the overall thermodynamic efficiencies. Multiple processes can be combined into one unit reducing overall costs and real estate requirements. In this paper the authors present the technical challenges associated with designing, building and testing these units.

APPLICATIONS

General

Figure 2 shows the typical zones where various compressor types are applied. Centrifugal compressors are most frequently found in the mid flow range of the plot with pressures from 1.2 bar to over 1,000 bar. From a marketing perspective, integrally geared compressors are unique in that they compete with every type of compression technology at various points in their range of applicability. For example, at lower flow rate and lower pressure applications screw compressors are also widely used. Similarly reciprocating compressors are applied in the lower flow capacity region and are capable of producing extremely high pressures. Centrifugal blowers cover low pressure applications and are most commonly configured with single stage direct drive or driven through a step-up gear box. For high pressure applications in the mid flow zone, barrel (or inline) compressors are most common as the sealing challenges are minimized. For high flow and low pressure requirements, most frequently axial compressors are employed.

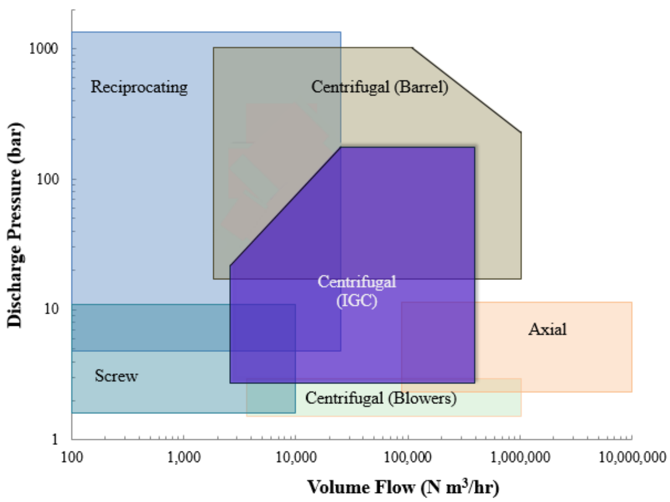

Figure 2. Typical Zones of Applicability for Compressor Types, Overlaid against IGC, for General Industry

Applications served by IGCs start with inlet capacities around 1,000 m³/h. For flows below 1,000 m3/h the required pinion speeds become large and impeller sizes become too small to maintain competitive performance levels with other compression technologies. Low cost screw compressors are typically oil flooded and will serve applications where some entrained oil is acceptable while IGCs can be oil free and cater to industries such as medical, electronics, painting, and food processing where oil free air is required. The required pressure ranges for plant air applications typically extend from 4 to 25 bar which means 2-, 3- or 4-stage IGCs with intercoolers between each stage with the upper limit of pressure ratio per stage of about 2.25.

IGCs are also unique in that different challenges are present across the entirety of their range of application. Many of the common industrial applications in which IGCs are applied are shown in Figure 3, relative to the current operating range of IGCs. Comparing the industries served shown in Figure 3 with the common architectures from Figure 2 it becomes evident how the various compression technologies compete in various industries. From here it becomes relevant to describe each of the separate industries in a bit more detail to show how the complexity of the cycle changes and how this impacts the design requirements that will follow for the IGC.

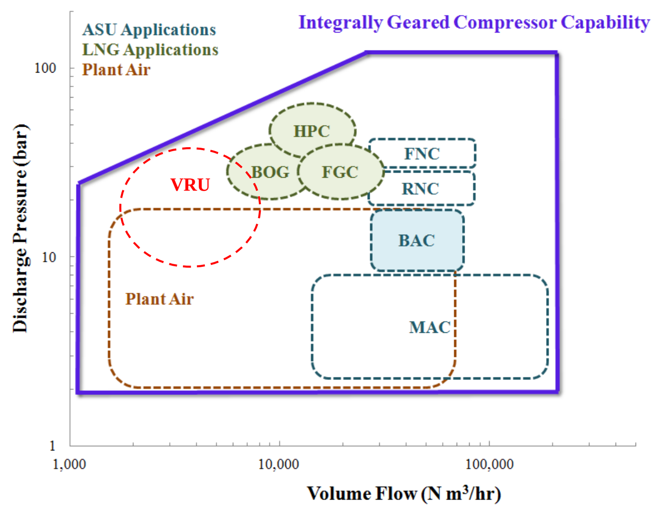

Figure 3. Applications for IGC as Function of Pressure-Capacity

Plant Air Compressors

"Plant Air" compressors are normally used to provide pressurized air that is used to drive pneumatic tools as well as drying or heat-up processes, aeration or flow-lift transportation. Examples of some typical applications are of industrial production facilities like automotive, electronics, plastic-synthesis and others. Plant air compressors typically deliver shop air at pressures from 8-11 bar and are typically fairly simple in design. Plant air compressors are offered as a standard product for many OEMs with a focus on offering a low cost and robust design that minimizes power consumption for the end user, in an easy to control package.

Figure 4 shows the basic arrangement of an IGC for a plant air application. In this application, atmospheric air is taken from an exterior environment and filtered to remove particle contaminants. The IGC is usually driven by an electric motor and then a dryer is used to remove moisture from the discharge air prior to delivery to the facility.

Plant air applications are usually non-API machines as this is a highly cost-competitive market. API672 qualification is not essential for an atmospheric inlet machine, such as for plant air, but some critical applications (mostly oil and gas plant related) are driven more by quality and reliability, and will pay a premium for an API672 compliant machine to reduce risk of down time and unanticipated problems.

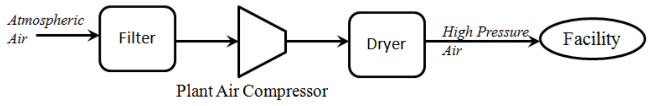

Figure 4. Schematic of a Typical Plant Air Application

Air Separation Units (MAC, BAC, FNC, RNC, N2, O2)

ASU applications have relied on integrally geared machines since the early 1970s. Power ranges on these individual units in ASU plants can be as high as 15,000 kW. [Fingerhut 1991] The air separation industry is one of the most important customers for IGC-OEMs since specific features of IGCs like: multi-speed aero lay-outs, ease of installing intercoolers between each stage, relative simplicity to create very flexible and efficient compressor designs out of standardized and proven components help to improve both capital expenditures (CAPEX) and operating expenditures (OPEX.) CAPEX is the cost of developing and/or providing non-consumable parts for the product or system whereas OPEX is an ongoing cost for running a product, such as the cost of electricity to drive the motor as well as the system's maintenance costs.

The compressors are almost the only power consumers in an air separation plant and also represent one of the major investment items in such plants. For that reason, providing a unit that provides an optimum balance of compression efficiency and cost is of special importance for ASU's. Therefore the best design requires close interaction between the end user and the equipment manufacturer through an optimization process that looks at the overall efficiency and production costs of the compressors. The best results are obtained when ASU plant designers and operators co-operate with the compressor OEM within frame-agreements for special sizes and capacities of a family of ASU-plants.

Figure 5 shows one basic cycle that an ASU plant may use and how several different compressor packages are integrated into the process. This cycle uses six different engineered compressors; a main air compressor (MAC), a booster air compressor (BAC), a main nitrogen compressor, a nitrogen feed and recycle compressor, and an oxygen compressor. The process is fed by atmospheric air drawn in by the Main Air Compressor (MAC) and delivered to the ASU center that separates the high pressure air into oxygen and nitrogen (other gases neglected for simplicity.) The ASU may require a cooling component, where liquid nitrogen can be applied. Oxygen is then compressed in another compression cycle. Nitrogen flow is split in two separate flow streams. One flow stream is further compressed to supply high pressure Nitrogen. The other flow stream goes to a "feed" compressor that precedes a liquefaction process. A recycle compressor then provides the nitrogen back to the liquefaction process to produce liquid nitrogen. The feed and recycle compressors may be separate electric motor driven compressors or the feed and recycle compressors may be combined into a single compressor.

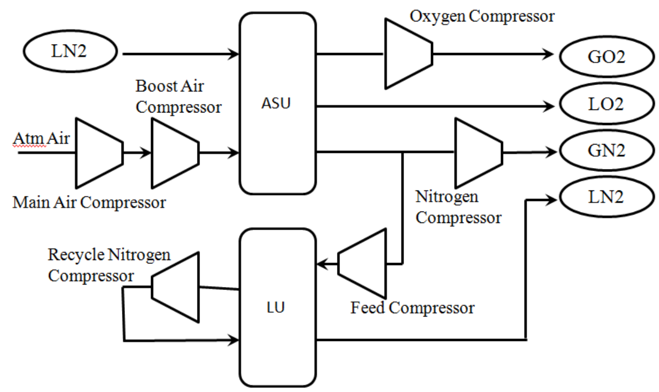

Figure 5. Typical Cycle Schematic for Air Separation Facility

Each of the six compressors detailed above must be engineered to match the specific cycle definition and plant production objectives. These considerations can change the specific architecture of the individual machine. Figure 6 shows a common 3-stage configuration for a MAC, while four stage designs are also common. Figure 7 shows a configuration for a combination feed and recycle nitrogen compressor where the recycle sections are on the lower two pinions and the feed section is located on the upper highest-speed pinion.

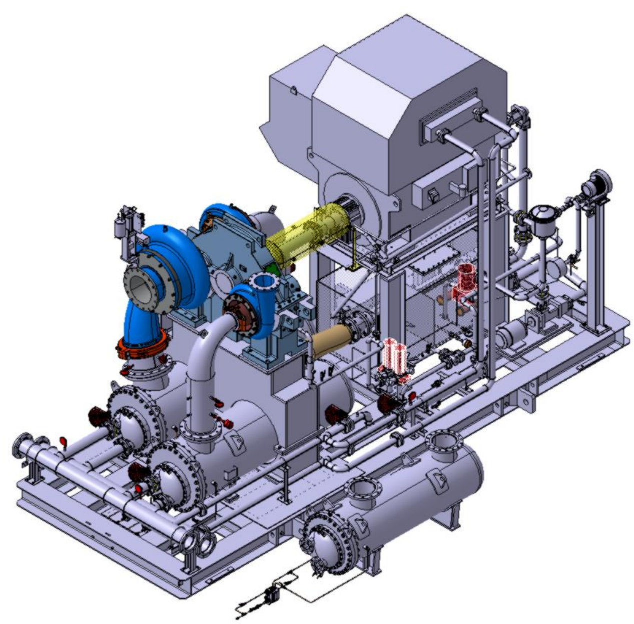

Figure 6. Main Air Compressor (Skid Arrangement)

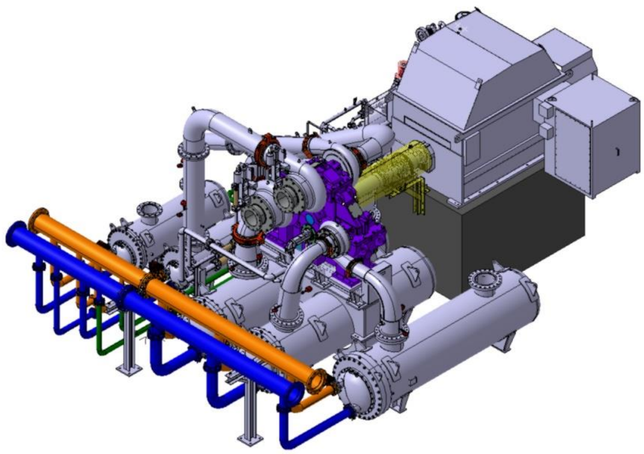

Figure 7. Combination Feed-Recycle Compressor (Non-Skid Arrangement)

LNG Terminal Applications

Each LNG facility is quite complex with unique demands for the turbomachinery. For methane, BOG compressors are used to recover gas that evaporates during the transfer from ship-shore when offloading from a vessel at the facility terminal. The boil off gas compressor is described in greater detail in the following section. Once the gas is re-injected back into the system, secondary pumps raise the pressure of the fluid and transfer it to meet the demands of the final application: power plant, city gas, etc. It is also possible to augment the pressure with a higher-pressure compressor after the BOG compressor directly to meet delivery needs. This is more commonly done with reciprocating compressors but advanced IGC designs can have the ability to meet the needs depending upon flow and pressure requirements.

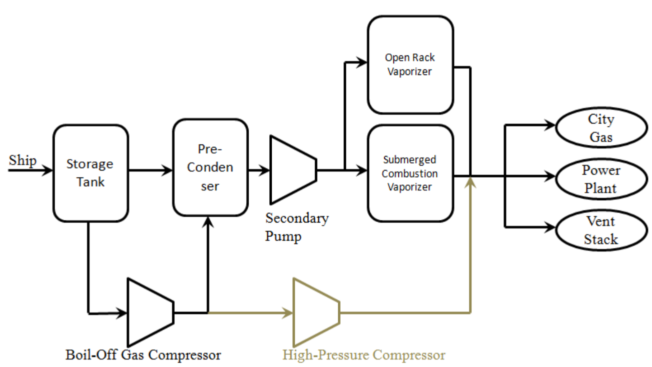

Figure 8. Sample LNG Terminal Configuration.

Boil-Off Gas

LNG transport and processing facilities operate efficiently by maintaining the fluid in a liquid state at low pressures and extremely low temperatures. Chaker [2015] discusses some of complexities and challenges of managing boil off gas (BOG) in natural gas liquefaction plants. As the temperature of the fluid fluctuates, small amounts of LNG change state from liquid to vapor (hence the term "boil-off gas"). Within a storage tank expanding vapor causes the pressure inside the storage tank to increase. Eventually the boil-off gas must be removed to maintain safe pressure levels. The boil off gas can be removed and routed through a gas refrigeration compressor to be pressurized for designated service to either: storage tank, for burning as fuel, or to pipeline for distribution.

Many unique design features must be integrated to produce a package suitable for extreme conditions a boil off gas compressor is subject to, where temperatures may be as low as -140°C. At these temperatures the bearings and gears must be insulated from the cold gas flowpath using special materials to maintain lubrication flow. Since IGCs use overhung volutes, the stages can be separated from the central integral gearbox by means of sandwiched heat barrier walls which prevent excessive heat flows between compressor stages and gearbox/bearing-housings. Special materials must also be selected for the stage housings with sufficient fracture toughness under cryogenic conditions, like high alloyed stainless steel grades or even aluminum. Additionally, since the working fluid is a hydrocarbon, API 617 design guidelines must typically be rigorously followed.

Relatively low-cost reciprocating compressors have historically been used to compress the boil-off gas. However, advances in sealing technology (DGS) are now allowing IGCs to offer competition to traditional BOG compression with OPEX showing a fast ROI. Figure 9 demonstrates the reduced size and also shows how the reduced dynamic loading requires less supporting mass.

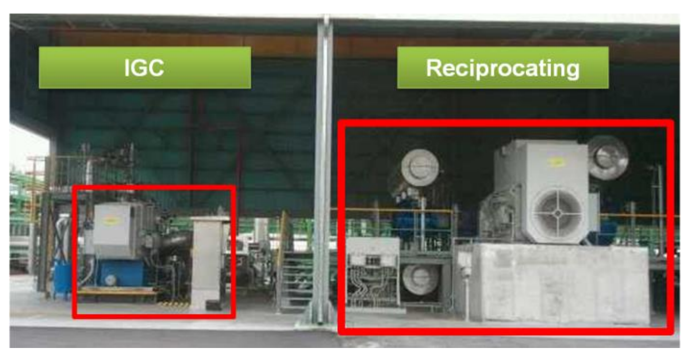

Figure 9. IGC and Reciprocating Compressor in Same BOG Application

In the case of tanker transport, the amount of boil-off gas heavily depends on the laden-condition of the LNG tanker. This requires that the boil-off compressor, the so called Low-Duty-Compressor, handle a wide operating range down to about 30% of its design condition. To achieve this wide operating range the compressor package may be fitted with a variety of technologies to ensure stable operation across the full range of operating conditions. One approach includes multiple recycle and bypass loops that can be utilized as demand changes. This particular approach also requires variable diffuser geometry on the first compressor stage.

CO2 Compression

One of the most challenging applications for IGCs has been in CO2 compression for pipeline transportation of the fluid. In 1999 an eight stage integrally geared CO2 centrifugal compressor was installed for pipeline service in North America. Olson [2004] explains the design considerations required to take CO2 from near atmospheric conditions to the super-critical phase that allows efficient pipeline transportation. The authors explain the many challenges of dealing with supercritical CO2 (such as the fluid behaves in some respects as a liquid, however with physical properties varying more like a gaseous state.) Inlet guide vanes were applied to regulate flow and a careful control system was required to account for inter-stage characteristics to avoid surge. The overall compression ratio across these units exceeded 180:1. Previously, CO2 compression for pipeline service had been accomplished by using numerous, multi-stage reciprocating compressors.

Fuel Gas Compressor

Fuel gas compressors (FGC) are often employed in gas turbine driven power generation applications. In such plants, low pressure gas delivered from the pipeline must be pressurized to match the gas turbine requirements, see Figure 10. A multi-stage IGC is often used. In these cases dry gas seals are required to minimize the amount of flammable process fluid escaping to the atmosphere. These applications frequently have requirements for explosion proof motors.

Gas turbines require a certain pressure of fuel gas to combine the fuel gas and pressurized air before combustion. Historically, when main pipelines were built the gas pressure was adequate. Normally, the gas pressure was reduced from this high pipeline pressure and delivered at the gas turbine's required pressure. In the North American market, fuel gas compressors were not often required.

However, as higher efficiency gas turbines were developed, the required gas pressure kept increasing. The required gas pressure for conventional types of industrial gas turbines is around 250 to 300 pounds per square inch gauge (psig), or 17.5 to 21 barG on average. However, the latest generation of industrial gas turbine requires relatively high pressure gas, such as 500 to 600 psig (35 to 42 barG). In addition, aeroderivative-type gas turbines now require gas pressure as high as 700 to 1,000 psig (49 to 70 barG).

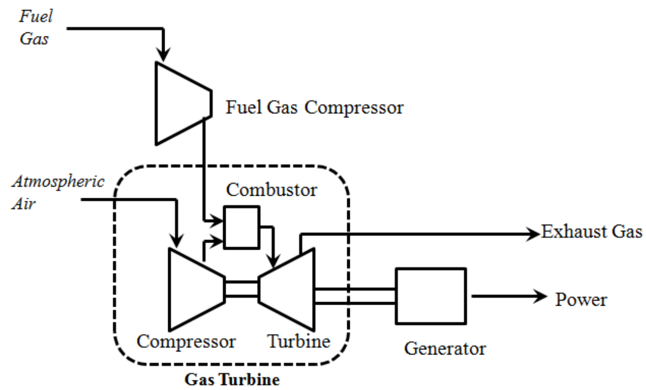

Figure10. Fuel Gas Compressor and Gas Turbine.

Off-Shore Platform Applications

Locating an integrally geared compressor on an offshore platform has some intrinsic benefits, such as the small foot-print size, high efficiency, and low levels of dynamic forces. For an off-shore platform there will be strict safety requirements which will translate to unique requirements for each application and a demand for maximizing API617 compliance with minimal exceptions. Two typical applications gaining prevalence for off-shore platform applications are instrument air compressors and vapor recovery compressors.

An instrument air compressor (IAC) is used to supply air to the various pneumatically operated processes located on the platform, such as control valves. Air is also needed to supply the nitrogen generators onboard. It is far more efficient to have a concentrated main source of air for all the systems onboard than it is to have discrete individual sources. An instrument air compressor is a safer option to replace natural gas-powered pneumatic controls. With an IAC the end user can realize cost savings, reduce methane emissions and ensure the safety of their equipment and the overall installation.

A vapor recovery compressor (VRC) is another example application for use in an oil & gas platform. The vapor recovery compressor offers the combination of relatively high suction capacity and very high head capacity by using multiple centrifugal compressor stages in an overhung design. This makes an IGC suitable for vapor/steam recovery compression because it can handle inlet pressures down to partial vacuum levels. On offshore oil rigs, many of the processes produce residue gases that are close to atmospheric pressure and contain flammable and/or lethally toxic components that must never be allowed to escape to atmosphere for safety and environmental reasons. Past practices called for routing these residues to the platform flaring system but they are no longer considered environmentally friendly solutions, either. On most modern designs, the residue gases are gathered and fed into vapor recovery compressors that raise their pressure to allow injection into some other gas process, e.g. the inlet gas stream to gas lift or gas injection compressors, where they can be used to enhance the production capacity of oil.

In the case of steam recovery, the low molecular weight of steam requires a very high head increase to gain reasonable pressure ratios, especially in case of high inlet temperatures near the boiling point or even in the super-heated condition. Also, the ease of installing pipe-integrated water-spray coolers between stages makes a multistage IGC the ideal solution for a vapor/steam compression process "along-the-saturation-line".

Off-Shore Transport Applications

In a similar manner to onshore LNG applications compressing boil-off gas, IGCs can be adapted for use on LNG carrier vessels, Figure 11 shows a sample configuration of one such unit. The wide range of operating conditions associated with these applications can be realized relatively easily by an overhung IGC compressor stage since its diffuser area is accessible to actuate variable diffuser vanes from outside. In this way a turn-down range of nearly 75% at constant discharge pressure can be achieved.

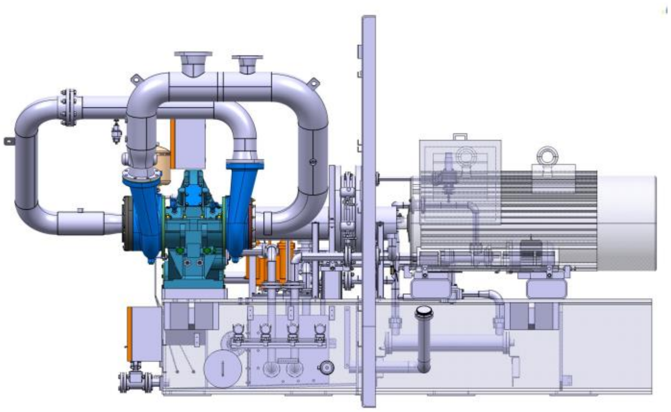

Figure 11. Unique Aspects of Some Transport Applications

Extreme Applications

Compressor inlet conditions of high pressure in conjunction with high temperature challenge a turbo compressor in both structural and dynamic ways. Insulation and special materials of the stage housings, comparable to the above description of cryogenic compression, are required to mitigate the high temperatures.

A high inlet pressure means high density which increases compression power and results in a high "power-density" design. High power density designs are more mechanically challenging than typical applications. Housing materials of the stages need to be suitable for high temperatures as well as needing to be of high yield strength to bear high pressure loads. 17-4 PH casting is such a kind of high strength alloy with good temperature stability up to about 500°C as a typical example of a housing cast material.

In case of an IGC, a high power density resulting from the high pressure inlet also requires advanced gear design at high pitch line velocities up to 185m/s in order to reduce forces on bearings and teeth-flanks. Advanced pinion journal bearing designs are also required which can handle specific bearing-loads up to 3 MPa with surface velocities up to 115m/s. Both the gears and bearings are available on proven designs that can meet these criteria., A suitable gear-design directly depends on the accuracy of gearbox and gear-set manufacturing while direct lubricated (evacuated) offset tilting pad designs of the pinions' journal bearings can handle the high bearing loads and velocities.

Special care must be taken with increased cross-coupling excitations created by increased density around the impellers, the upstream impeller eye seals and shaft seals. The influence needs to be analyzed by suitable rotordynamic software. It stability requirements cannot be met the choice of bearing geometries and oil viscosity may be modified in order to dampen the resulting sub-synchronous vibrations (forced or resonance-type) and also to control operating bearing temperatures. Both of them may easily become an issue with such high pressure compressor designs.

Figure 12 shows a configuration, without insulation or sound enclosure, of a high temperature, high-pressure integrally geared compressor package (without drive motor). The compression section is skid mounted with the motor mounted on a separate concrete block and an air-cooled heat exchanger that is applied to recycle the hot air during surge control and start-up.

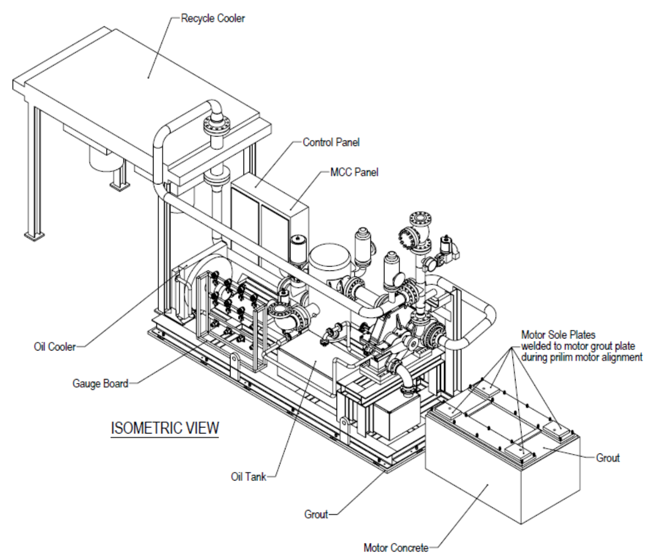

Figure 12. Packaged Layout of a High-Pressure, High-Temperature Air Compressor Showing a Recycle Air Cooler

Common Specification/Standards

Mainly in the oil and gas industry there are so called standard specifications for compressors in general and IGCs in particular which represent end users' field operating experiences. These specifications are applied to the detailed design on both the aerodynamic and dynamic/structural analysis of the main components like: impellers, rotors, bearings, gear-sets, shaft seals, drive-train and coolers.

Those standards are

- API 672 for IGC-type plant air compressors in critical applications (such as oil and gas plants)
- API 617 for gas compressors with its Chapter III for IGC-type gas compressors
- IGC DOC 27/12 for centrifugal oxygen compressors (as an example of an international non-API spec.)
- Det Norske Veritas (DNV) – Certification for off-shore installations.

Most of the non-API end users and customers, like the air separation industry, do have their own simplified specifications according to their special needs and plant experiences, but very often they are derived from API 672, [2004].

Some other end-users, like marine (LNG mid-stream) and off-shore applications often add more detailed specifications onto API 617, [2009]. These additional specifications may include NACE compliance in the case of sour gas environment or additional special regulations of ship-build authorities like DNV, ABS and others in case of ship mounted IGCs that belong to the ship-propulsion, loading/unloading, and/or re-liquefaction of product like LD-, HD-compressors and/or N2-companders on LNG vessels.

Some of the limits placed upon API617 applications are derived from more traditional barrel style applications and are not necessarily realistic for IGC applications, such as:

- 20% of overall allowable vibration sub-synchronous vibration limit – vibration signatures of pinions of IGCs have multiple sub-synchronous components related to bull gear, harmonics, etc. that are separate from the premise of this limit to control lateral instability.
- 25 micron allowable overall vibration magnitude for the bull-gear rotor. The bull gear operating speed is very low and it is able to operate very satisfactorily for the design life of the machine at higher vibration levels than this. More reasonable limits would be 30-35 microns. Achieving such a low vibration level on the bull-gear may require trim-balancing on the test stand, which takes time and adds cost.
- Unbalance verification testing for high-speed pinions is doable, but adds both risk and unnecessary complexity. For unbalance verification testing a known unbalance is placed at specific locations to excite specific modes. In an IGC easy access to locations of the pinion to add and remove known unbalances is difficult and the benefits of conducting the unbalance verification test are arguable.

Table 1 shows some of the various interactions between different applications and unique requirements.

Table 1. Various Applications and Different Requirements.

	Plant Air	ASU	BOG	FGC
Impeller Type	Open	Open + Closed	Open	Closed
Casing Material	Cast Iron	Cast Iron	Cast Iron	Cast Steel
Coolers	Air or Water	Water	Water	Water
Flow Control	IGV	IGV	IGV + Recycle	*Varies by mfg*
Seals	Labyrinth	Carbon Ring	Dry Gas	Dry Gas
Pinion Bearings	Fixed Geometry or TPJB	TPJB	TPJB	TPJB
Driver Type	Induction Motor	Induction or Synchronous	Induction	Induction Motor
Certifications	Non-API, API672, or ISO 8573-1	Non-API or API672	API617	API617

	LNG Transport	VRU	Other Extreme Applications (CO2, etc)
Impeller Type	Open	Open + Closed	Open or Closed
Casing Material	Cast Steel	Cast Steel	Cast Steel
Coolers	Water	Air-Cooled	Water
Flow Control	VDV+Recycle	*Varies by mfg*	*Varies by mfg*
Seals	Carbon Ring	*Varies by mfg*	Floating Ring or Dry Gas
Pinion Bearings	TPJB	TPJB	TPJB
Driver Type	Induction	Induction	Induction
Certifications	API617/DNV/ABS	API617/ NACE	API617

SIZING AND SELECTION

Regardless of manufacturer or application, an engineered integrally geared compressor essentially travels through the same development process. The first portion of that process is established by optimizing the cycle and the aerodynamic performance; next, the mechanical implications are considered and iterated against the aerodynamic components for optimal configuration. Figure 13 shows the various inter-connectivity of aspects that must be iterated to achieve a satisfactory design.

The importance of the aerodynamics cannot be overstated as shown in the "satellite configuration" on all the aspects that are influenced by aerodynamics. Likewise errors or poor assumptions in other aspects can result in problems that prevent operation of the unit altogether. As referenced, some issues that prevent operation are: bearing over temperatures, excessive rotor vibration, impeller fractures, seal leakage, etc.

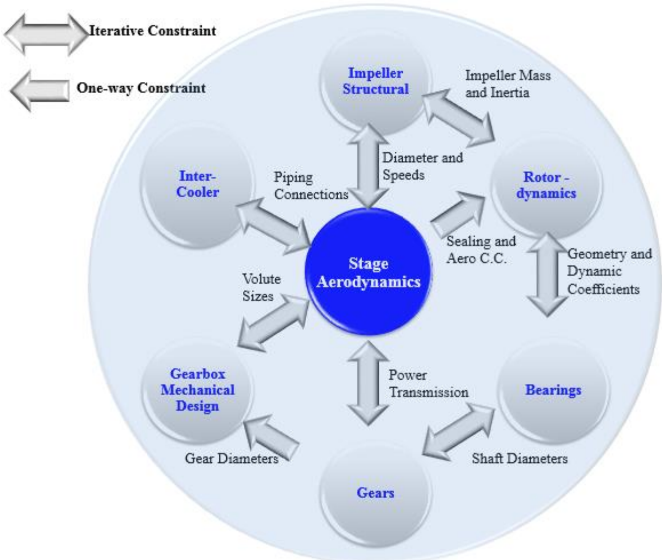

Figure 13. Component Connectivity for Design Process

Sizing and Selection Process

The sizing and selection of a compressor for particular applications depends on matching aerodynamic performance to cycle requirements and integrating mechanical limitations into the aerodynamic design. A rough description of the sizing and selection approach follows:

Step 1: Number of Stages and Inter-Coolers

The number of stages is determined based upon the overall head. Higher head requirements can lead to increased torque requirements per pinion and result in unacceptable bearing and gear loads.

Step 2: Impeller Speed and Diameter

Structural limitations of the rotating impellers are considered based on the impeller tip speed and material constraints.

Step 4: Aero Power and Gear

The aero power of each stage on a pinion sums to the total mechanical loading that the gear teeth must transmit. These gear forces must also have a reactionary component – which is the bearing. Therefore the limiting factor for power transmission of a pinion is frequently the load capacity of the bearings.

Step 5: Bearing Selection

To minimize mechanical loss, the pinion bearing diameter is minimized to the maximum extent. To handle the maximum load it is common to maximize the axial length of IGC pinion bearings. The primary issue is to ensure that the bearing is sufficiently large to handle the reaction load from the gears and sufficiently well designed to ensure a stable rotor design.

For the bull gear bearing the limiting factors are: 1) shaft strength, 2) lateral rotordynamics, and 3) torsional rotordynamic considerations.

Step 6: Sum Stage Power and Mechanical Losses

Now that preliminary configurations for the bearings and gears are completed it becomes possible to better estimate the level of parasitic losses that will be present. The parasitic losses are added to the stage power to determine the required motor power. Of course, some margin is required, as discussed later. Parasitic losses include mechanical losses in the gear meshes and bearings. In application, the mechanical losses are evident in the increase in oil temperature.

Step 7: Secondary Gas Path

A multi-stage IGC will have more seals than a comparable inline compressor. Whereas an inline compressor must seal at the inlet of the first age and the discharge of the last stage, each stage of an IGC will require a seal. The leakage from the gas path must be assessed to accurately estimate the performance of the unit. The leakage rate will depend upon the type of seal applied and the pressures. The greater the pressure the greater the sealing challenge. Once the leakage rate is known this can be compared to original estimates of leakage to determine if additional iteration is required.

A more detailed review of each of the key design considerations follows.

Aerodynamic Design

The compressor stages of an IGC are typically designed to maximize efficiency at the specified operating conditions. Since engineered compressor packages are delivered with performance guarantees, the manufacturer must be able to accurately estimate the performance of the unit when quoting the job. To minimize aerodynamic performance risk, most manufacturers avoid designing entirely new stages for individual applications since this is both costly and increases the uncertainty in the performance. Therefore, most engineered compressor stages are either scaled from an existing stage, or developed following a detailed set of standard design guidelines.

When scaling from an existing design, it is critical that the performance of the stage has be accurately measured. Hence, the designs used in scaled applications are carefully developed, manufactured and performance is independently validated via stage laboratory testing. Testing should cover the full range of potential operating conditions to define the head, flow and efficiency profile across the operating map. Performance characteristics of master components should be referenced in non-dimensional parameters such as head and flow coefficients and efficiency, to facilitate easy application to a wide range of applications. A few master stages may be used to cover a broad range of flow coefficients. Figure 14 shows how three master impellers can be applied to applications ranging in flow coefficients between 0.05 and 0.15. Each individual design is optimized at a given flow coefficient and can be flow cut to operate at reduced flow. A appropriate number of master impellers must be chosen to avoid gaps in coverage of the design space or large flow cuts which adversely affect performance.

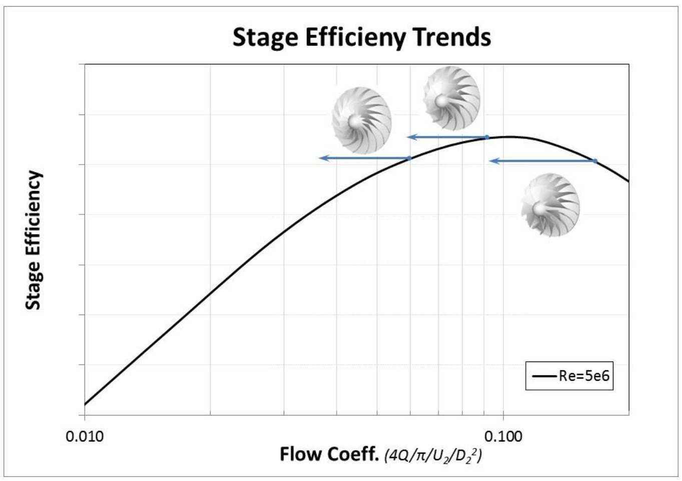

Figure 14. State-of-Art Stage Efficiency

Alternatively, if master stages are not used, then new stages are developed following a validated set of design guidelines. Design practices should be sufficiently well developed and demonstrated to consistently produce acceptable performance which matches expectations. Stages developed in this manner will share similar characteristics such as blade angle distributions, passage areas, incidence distributions and diffuser configurations, etc. It is also critical that a well validated experience range be established where the design guidelines are established to perform acceptably.

Impeller Aerodynamic Flows

The individual flowpath components to be well matched with each other to maximize stage performance. Impellers, diffusers and volutes should be sized to operate with minimal loss near the design point although component matching can be adjusted to meet specific operating requirements. For example, it is common to adjust a diffuser setting angle to tune a compressor characteristic to match overall performance guarantees in the field.

Stall and Surge Limits

Modern IGCs face ever increasing demands for greater capacity and improved efficiency while maintaining their compact size. This requires the compressor stages to be designed at increasingly high flow coefficients and to operate at higher machine Mach numbers. As the Mach number and flow coefficient increase, it becomes more challenging to maintain a high efficiency and acceptable range. Figure 15 shows how the operating range reduces as the machine Mach number is increased. Stage efficiency will follow a similar decreasing trend with increase in Mach number.

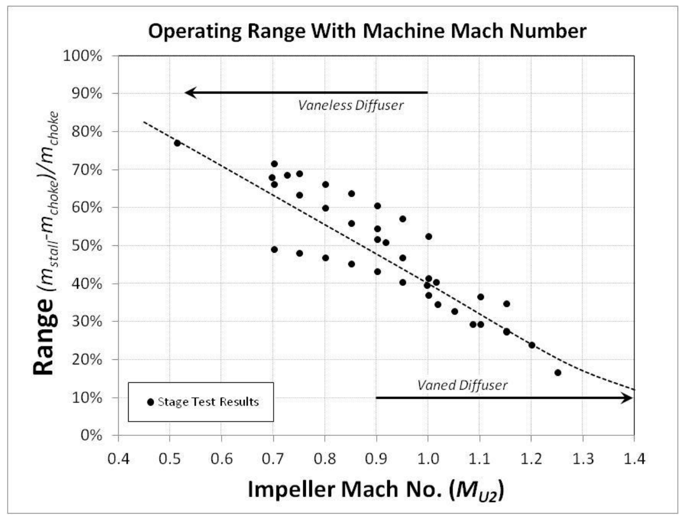

Figure 15. Operating Range for Various Impellers Versus Mach Number

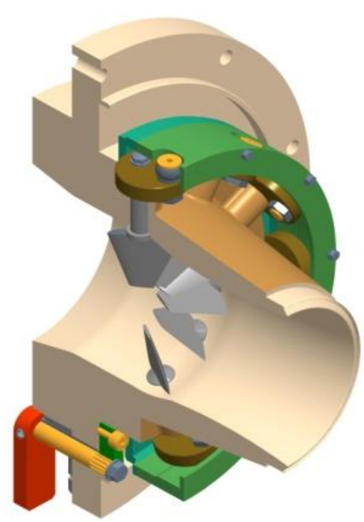

Figure 16. IGV Mechanism to Control Flow Entering

Impeller

Range Extension and Variable Geometry

Compressor packages are typically operated at a constant discharge pressure with variable flow rate. To achieve flow control a variety of approaches may be used including stage bypass, inlet throttle valves, variable diffusers or variable inlet guide vanes (VIGVs). VIGVs add swirl ($C_{\theta 1}$) to the inlet flow which directly reduces the work according to the Euler turbomachinery equation shown in Equation 1. As the IGV's close the flow of the compressor is reduced while maintaining the required constant discharge pressure.

$$Wx = U_2 C_{\theta 2} - U_1 C_{\theta 1} \qquad (1)$$

Variable diffuser vanes (VDVs) are also used to extend low flow operating range. They function by matching the diffuser to the impeller exit flow state at all operating conditions to minimize pressure loss to postpone stall in the diffuser inlet and impeller exit. VDVs can potentially offer greater throttle range than IGV's, see Figure 17, but are more complex and costly to integrate into the design (Figure 18 shows the vane section of a VDV located downstream of the impeller exit.

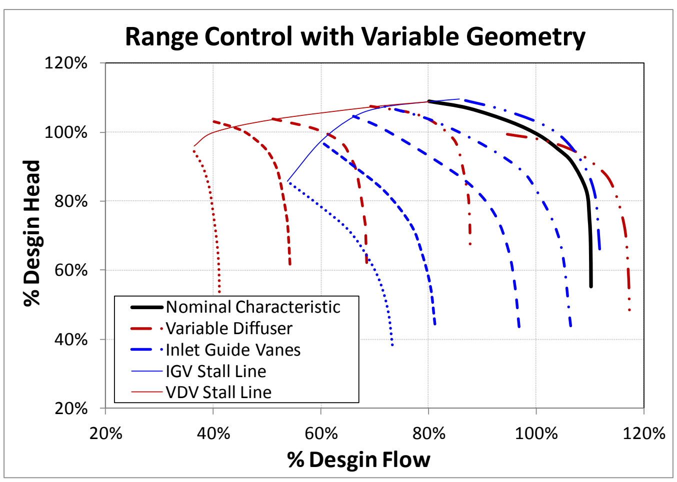

Figure 17. IGV Versus VDV Performance

A number of authors have analytically and experimentally examined the stage performance with a variety of different IGV and VDV designs. For example, Handel [2014] shows how different IGV profiles can be used to adjust the control characteristic of the stage.

Figure 18. Vane for a VDV Assembly

Secondary Flow Paths

Each compression stage in an IGC requires effective sealing technology to operate efficiently. Both open and covered impellers utilize a shaft seal to prevent the high pressure gas from leaking out at the shaft. Covered impellers also require an eye seal on the inlet to minimize leakage back to the stage inlet. Depending on the type of seal used, leakage may vary from near zero to several percent of the flow for each stage. Failing to include these losses will result in incorrect estimation of the power consumption and/or delivered flow of the unit.

In an open impeller there is no eye seal, but the clearance between the rotating blades and the stationary shroud must also be considered when assessing the performance of a machine. In application, the tip clearance may need to be increased beyond optimal levels to account for potential axial movement of the pinion, thermal growth and manufacturing and assembly tolerances. The effects of these changes on the overall stage performance must be carefully estimated and included in the overall performance predictions.

Inter-Stage Cooling

The architecture of integrally geared machines makes it simple to inter-cool between stages. Inter-cooling is key to maximizing compression efficiency since it allows the overall compression process to follow a path closer to an ideal, isothermal process. Since each stage is mounted in a separate housing and the discharge is collected in a volute for transfer to the next stage, it is simple and inexpensive to incorporate inter-cooling between stages. However, care must be taken in the design of the inter-stage components to ensure that the performance improvement (power reduction) obtained by inter-cooling is not nullified by excessive pressure drop through the coolers and piping.

Mechanical Design

In order to maximize mechanical robustness on a “one off” style machine an approach is adopted where individual components fall within past experience ranges, to the maximum extent possible, even though the design in its entirety exceeds past experience. All components are subject to detailed analytical assessments against relevant design standards (API 617, ASME BPVC, etc.). Figure 19 shows the core compressor unit and highlights the key components requiring detailed analysis: gears, bearings, seals, pressure boundaries, impellers, shafts, and bolting.

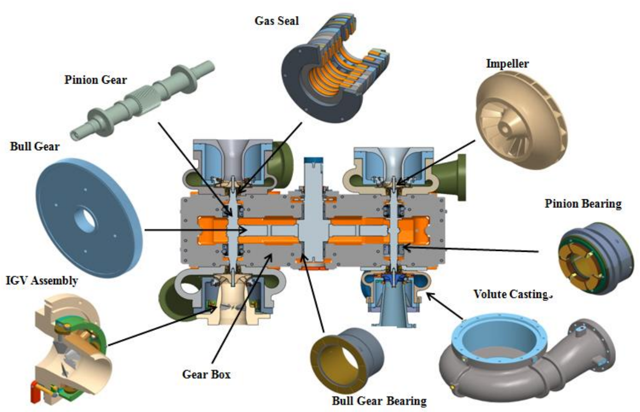

Figure 19. Primary Components of Core Section of Integrally Geared Compressor

The primary design objective for an IGC is safety and reliability. To achieve an optimal design requires balancing the aerodynamic considerations previously mentioned with mechanical concerns. The mechanical limits on an IGC are associated with aspects listed in Table 2.

Table 2 Mechanical Factors and Relevant Design Standards

Component	Limiting Design Factor(s)	Relevant Design Standards
Impeller Structural	High-Cycle Fatigue, Low-Cycle Fatigue	Separation Margin > 10% LCF > 1 x 10^5
Pinion Shaft	Lateral Rotordynamics	API 672 API 617
Pinion Bearings	Load Capacity*, Surface Speed*,` Dynamic Coefficient	API 672 API 617
Coupling	Lateral Rotordynamics	API 671
Gears	Torsional Rotordynamics	AGMA
Bull Gear Shaft	Power Transmission Lateral Rotordynamics Torsional Rotordynamics	API 672 API 617 API617
Thrust Management	Thrust Bearing Thrust Collar	API617
Gas Seals	Leakage Requirements Dynamic Coefficient	Application Specific API617
Volutes	Localized Stress	ASME BPVC
Inlet Guide Vane	Structural and harmonic	OEM Specific

** oil film temperature is underlying limit that sets load and speed.*

After the aforementioned mechanical and aerodynamic design for a particular unit is established, the overall performance can be reviewed. In a general sense some size-dependent characteristics can be seen for various sizes of machines. Typical characteristics in terms of mechanical loss are shown in Figure 20. Above 10,000 kW mechanical losses are typically between 2.2 and 2.6% depending upon design and manufacturing of the key components, and operating conditions. Below 10,000 kW a wide range of mechanical losses are seen in industrial machines. This variability is attributed to: thrust collar v. thrust bearing, number of stages, manufacturer design rule variations, etc. Below 1,000kW the mechanical losses begin to increase rapidly as a percentage of total power and for this reason screw compressors are more common in this zone.

$$P_{tot} = \sum P_{journal} + \sum P_{thrust} + \sum P_{mesh} + \sum P_{windage} \quad (2)$$

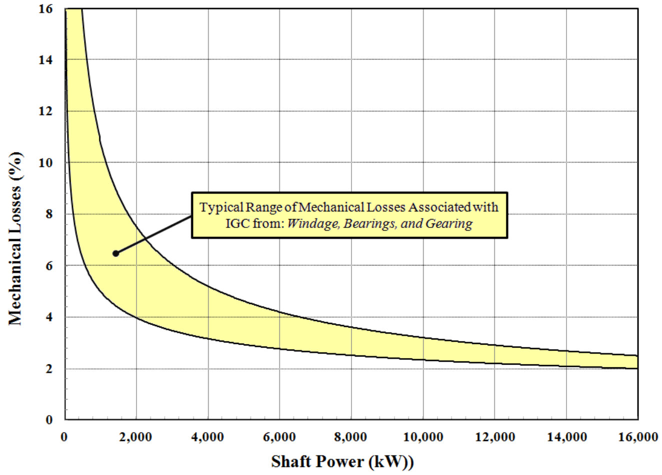

Figure 20. Typical Mechanical Losses as Function of Machine Power

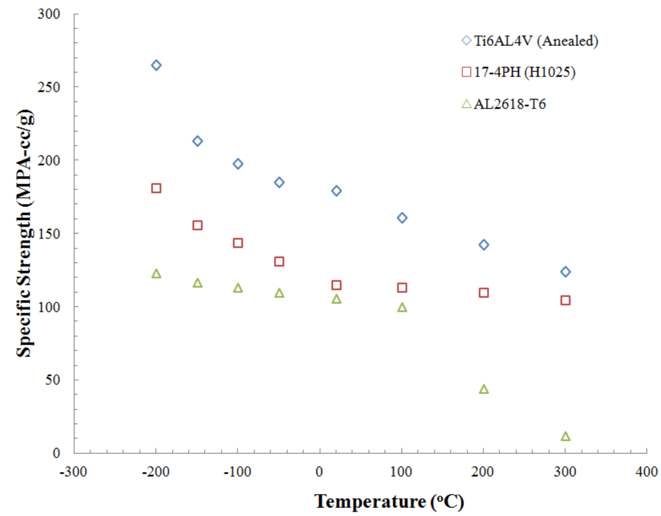

Figure 21. Specific Strength Limits of Common Impeller Materials

Impeller Structural Design

The material selection for an impeller plays a significant role in establishing the design limits. Common impeller material selections for atmospheric impellers in plant air and ASU applications are martensitic stainless steels 15-5PH and 17-4PH, with heat treatments that provide reasonable levels of ductility to improve defect tolerance. Figure 21 shows the trends of common materials vs. temperature. Ti6Al4V provides superior strength to weight ratio for temperatures below 300 C. The lower density of Ti6Al4V as compared to 17-4PH can also offer significant advantages in the rotordynamic design due to the reduced mass and inertia. However, the cost differential of titanium to stainless steels is significant enough to be avoided in many applications unless absolutely necessary. Light density gases and/or applications requiring high tip speeds make use of titanium essential. For low-temperature or cryogenic applications aluminum alloys, such as AL2618, can be used with success.

Gears

The gears of an integrally geared compressor are most frequently high-speed single helical gear configuration. Occasionally double helical gear configurations can be found. But the challenge with the double helical arrangement for IGC is that the net aerodynamic thrust produced on the pinion is not zero. Therefore one face of a double helical gear will always be loaded more than the other. Also the cost of a double helical is more than a single helical gear. A single helical gear can be arranged such that the gear mesh thrust will partially offset the aerodynamic thrust and help reduce the size of the thrust management system – thereby reducing bearing losses.

Several AGMA standards are applied to gear design itself: AGMA STD 6011-J14 is used to set the limits of a single high speed helical gear. API 613-2003 is used for the basic gear design and factor of safety in bending and pitting, with AGMA STD 925-A03 used for scuffing probability. AGMA 6011 specifies gear life of minimum 40,000 hours while AGMA 2101-D04 specifies life of 20 years or 175,200 hours Both with a factor of safety of 1.4. API617 also requires a design life minimum of 20 years and some customers require 30 years or more.

Figure 22 shows assembly of a small three pinion (six stage) compressor gear box. Two horizontal splits are applied in the assembly. The lower horizontal split is used to set the bull gear and two of the pinions. A second upper split is applied to locate the third pinion in contact with the open portion of the gears. Some manufacturers use vertical splits that reduce the complexity of sealing the gearing splits but the disadvantage of a vertical split is more complicated field service. Inspection ports are frequently applied, as shown above the workers hands, to allow visual inspection of the gears in assembly, pre and post mechanical test, and field service.

Figure 22. Three Pinion Gear Box, with Mid-section Being Lowered In Place.

The power transmitted from the bull gear to drive rotation of the pinions results in a driving torque, T, (N-m) shown in Equation 3, where P (kW) is the transmission power and N is the rotational speed (rev/min.)

$$T = \frac{P}{N} \tag{3}$$

For transmission through a helical gear the torque is resolved into a transmission force (F_T) at the gear pitch diameter (p_d) as given in following equation.

$$F_T = \frac{2 \cdot T}{p_d} \tag{4}$$

For balancing the thrust forces the axial force on the pinion from the helical gear must be known. Likewise the primary force that the journal bearings must withstand is from the gear forces. These axial and radial force vectors resolve from the transmission force. Equations 5 and 6 define the force vectors, where F_a is the axial force and F_r is the radial force, respectively.

$$F_a = F_T \cdot \tan(\beta) \tag{5}$$

$$F_r = F_T \cdot \tan(\varphi_t) \tag{6}$$

Figure 23 shows a helical gear system with a single pinion shaft where the pinion aerodynamic thrust forces and helical gear forces are balanced by a thrust collar that transmits the resulting thrust from the pinion to the bull gear. The bull gear in turn reacts the net aerodynamic thrust with the bull gear thrust bearing.

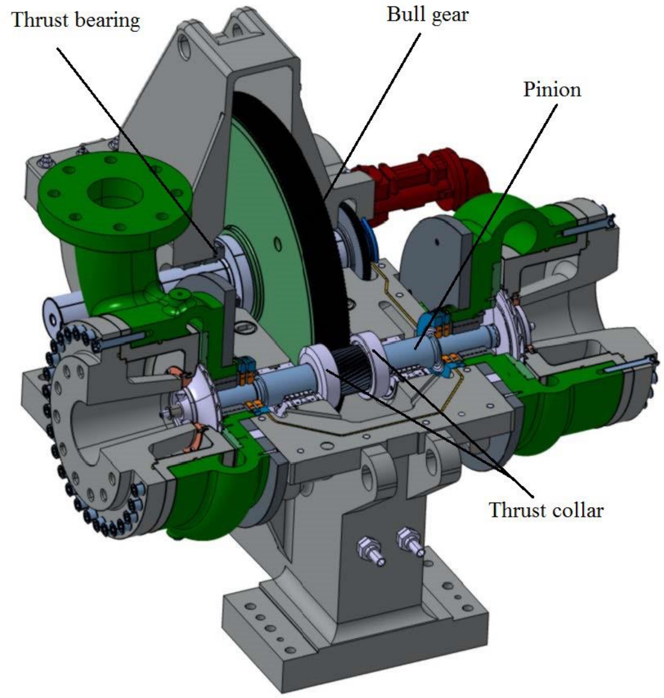

Figure 23. Assembly Showing Pinion, Bull Gear, Pinion,Thrust Collar, and Thrust Bearing on Bull Gear r.

Equation 8 shows the most common relationship for determining gear mesh stiffness from a helical geared system. This mesh stiffness is applied when conducting a coupled gear-pinion lateral vibration analysis [API 617, 2009]. In the relationship the net face width of the gear has a linearly proportional influence on the gear stiffness. The other primary factor is the helix angle. Beyond this no other factors are considered pertinent.

$$\overline{K}' = C \cdot FW \cdot \cos^2(\beta) 10^6 \tag{8}$$

The mesh stiffness may be converted to the x-y coordinate planes. The primary stiffness is determined from either the API method in Equation 5 or the newly proposed method of Equation 4.

$$\begin{gathered} k_{m_{xx}} = \overline{K}'\cos^2(\gamma) \\ k_{m_{xy}} = \overline{K}'\cos(\gamma)\sin(\gamma) \\ k_{m_{yx}} = k_{m_{xy}} \\ k_{m_{yy}} = \overline{K}'\sin^2(\gamma) \end{gathered} \tag{7}$$

Bearings: Low-Speed Bull Gear and High-Speed Pinions

The most common bearing type found on integrally geared compressors for industrial applications are fluid film bearings. The fluid film bearing offers the advantage of high load capacity, inherent damping, and proven reliability with a sacrificial lead-tin babbit that can protect the more expensive shaft assemblies in the event of damage. However, the babbit material's low melting temperature and strength are also one of the limitations on journal allowable surface speed and bearing loads.

Nicholas and Kirk [1990] provide a general design approach of fixed geometry and tilting pad bearings that shows the influence of static and dynamic coefficients on rotordynamics. The reference is particularly well suited to assessing fundamentals associated with the fixed geometry bull gear bearings. The authors describe the influence of the various bearing types on the stability and forced response of rotor systems. Two other excellent references to the design aspects of tilting pad journal bearings are given by Nicholas [1994] and He [2016], respectively, which more than adequately describe the limiting factors such as temperatures and gives the implication of design variables.

Minimizing the journal diameter to the greatest extent possible helps limit the amount of oil shear loss. The high-speed pinion journal bearings are the largest single contributing factor to the machines' mechanical loss when a thrust collar is employed to control thrust. Therefore journal diameters will be minimized up to the maximum unit load and/or lateral rotordynamic extent possible to generate minimal mechanical losses.

For the low-speed bull gear the radial bearings are most commonly zero-preload fixed geometry type with two, three, or four axial grooves. Since the bull gear shafts frequently operate below critical speeds or in the vicinity of critically damped rotor responses, the stability benefits of tilting pad configurations are not usually required. Typical bull gear radial bearings operate with surface speeds between 15 m/sec and 30 m/sec with projected unit loads of 20 bar or less.

For the high-speed pinion bearings tilting pad journal bearings are most common. The preference is usually for five pad load-between-pad, although four-pad or five pad load-on-pad configurations are not atypical. Surface speeds are frequently 60 m/sec. to 100 m/sec. The over-riding design limit is minimum film thickness (25 um) and maximum babbit temperature (100 ^{o}C). In order to achieve the maximum load capacity length-to-diameter ratios will frequently be close to 1.0. Extending the length-to-diameter ratio further can lead to edge load/wear and limited tolerance to misalignment. The OEM will most commonly gauge risk for a particular application relative to past experience plots similar to that shown in Figure 24.

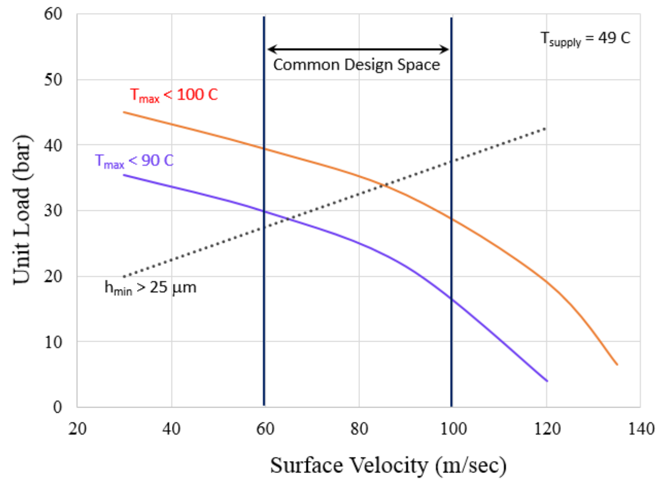

Figure 24. Typical Experience Limits for Pinion Bearing Design

For tilting pad journal bearings there are several different types of commonly applied pivot types: rocker back, spherical pivot, and flexure pivot. Each pivot type has advantages and disadvantages that warrant a separate discussion altogether. Key aspects of each pivot type relative to IGC can arguably be stated as follows: Rocker back provides the greatest pivot stiffness, spherical pivot provides greatest misalignment tolerance, and flexure pad allows for minimum radial space. With care most types of pivot can be applied successfully across the broad spectrum of IGC machines.

Pinion Thrust Management Options

The net axial force from the impellers may be entirely or partially offset at the design condition by the axial force from the helix of the pinion gear. To carry the residual and any off design loading the net thrust on the pinion must be resolved through either a thrust bearing or a thrust collar. Figure 25 shows an arrangement where one face of the thrust bearing is located on the opposing face of each of the pinion journal bearings. Figure 26 shows an arrangement where the thrust from the pinion is mitigated by use of a thrust collar acting against the rotating bull gear side face.

A pinion thrust bearing directly resolves the net thrust from the pressure differential on the impellers and the gears. Pinion thrust bearings may have various arrangements: 1) one face integrated on to the outboard side of each journal bearing as in Figure 25, or 2) one face integrated on to the inboard face of each journal bearing, or 3) on the opposing face of just one of the journal bearings, or 4) as a separate thrust management system for the pinion from the journal. However, any arrangement of a thrust bearing on a high speed pinion acts to consume substantial mechanical power and requires substantial oil flow. The advantage of a thrust bearing, as compared to a thrust collar, is reduced axial motion. Therefore, unless absolutely necessary, the preference is to apply thrust collars to control pinion position.

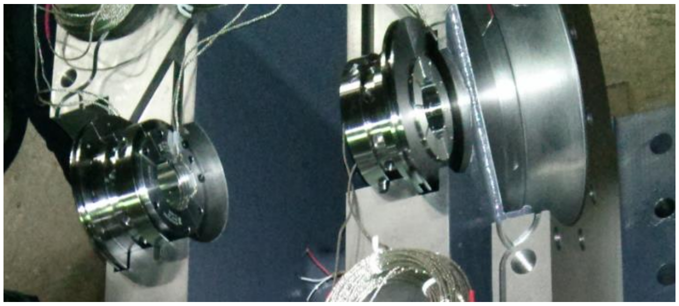

Figure 25. Pinion with Outboard Thrust Bearing Arrangement

Figure 26. Pinion with Thrust Collar Arrangement

Lubricated thrust collars is another method that may be applied in integrally geared compressors to transmit axial loads from the compressor stages on the sides of a pinion gear to the main shaft in the bull gear. The area of load transmission is relatively small as it is formed by overlapping sections of the outer diameters from the bull gear and a thrust collar on the pinion gear. San Andres, et. al. [2014] shows the approach to thermal-mechanical and dynamic assessment of thrust collars to understand their performance characteristics. A thrust collar relies on the same hydrodynamic principle as a thrust bearing. The difference being that relative motion between the bull gear face and the thrust collar face must be considered as both surfaces are in rotation around different axes, illustrated in Figure 27.

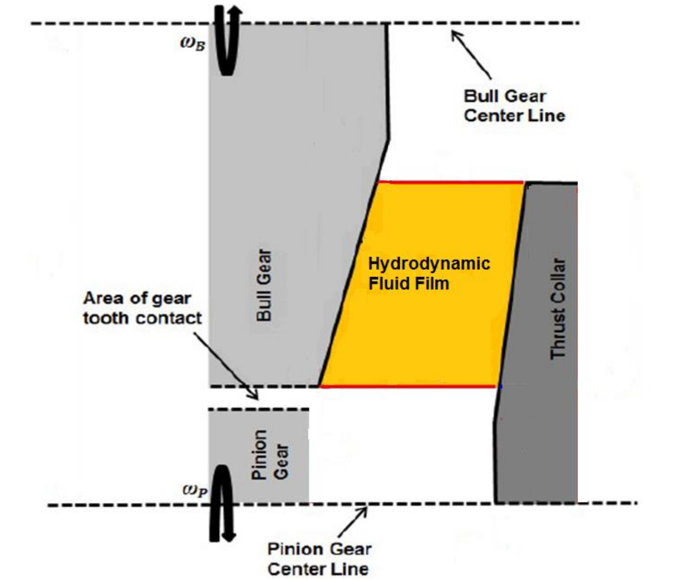

Figure 27. Hydrodynamic Function of Oil Lubricated Thrust Collar

Oil Flow Requirement

The most common oil types are ISO VG32 and ISO VG46. ISO VG32 is a lower viscosity oil that offers less surface shear and reduced friction losses. ISO VG46 is a higher viscosity oil that offers improved damping properties for the bearings and improved gear adherence for improved gear life and reduced wear/vibration. The total oil requirement is based upon summing the individual component requirements and adding approximately 10% additional margin.

$$Q_{tot} = 1.1\left(\sum Q_{bearings} + Q_{mesh}\right) \tag{9}$$

Lateral and Torsional Rotordynamics of Geared Systems

An accurate rotordynamic analysis is key to developing a mechanically robust compressor design. Lateral assessment of pinions and bull gear rotors is typically conducted in isolation of each other. Most pinions operate between the second damped peak response speed and the third damped peak response speed. The ratio of the pinion speed to the damped first critical speed can range from 8:1 to 1.1:1. For the higher pinion speed ratios the units may be highly susceptible to instabilities and substantial care/effort must be taken to minimize instability risk. Designing the pinion shafts requires iterating the unbalance response and stability characteristics of the pinion with mechanical design considerations such as: gear machining, gear box layout, gas path sealing etc. For the unbalance response other practical considerations come into play such as the required balancing approach. Schnieder [1987] gives an excellent overview of different classes of balancing depending upon shaft flexibility. Ultimately the designer tries to avoid the need for high-speed balancing as this complicates field repair but for high performance machines operating at high speeds this can become a requirement. Most IGCs only require low speed balancing due to the limited shaft length (only two stages per pinion.)

Gruntfest, P.M., et. al. [2001] highlights an important point in that many users mistakenly believe that tilting pad journal bearings eliminate all cross-couplings. While this is true for bearings themselves, substantial cross-coupling can be present from seals and impellers.

For the bull gear the primary factor influencing lateral rotordynamics is the coupling cg and half weight. From the rotordynamics perspective the intent is usually to minimize the coupling weight/inertia and overhung center of gravity to the greatest extent possible.

Advanced analytics can be conducted to couple the bull gear and pinion lateral vibration. There are two coupling mechanisms between the bull gear and the pinions: mesh stiffness and thrust collar moment stiffness.

Whereas the lateral vibration requirements strongly govern the pinion shaft design, static and dynamic torsional requirements frequently control the design of the bull gear shaft. In contrast to lateral vibration problems, torsional failures are extremely serious as the first symptom of a problem is frequently a catastrophic incident: broken shaft, gear tooth, or coupling. Corbo, et. al. [1997] offer practical design advice for mitigating risk and API617 provides sound analytical design requirements that depend upon the type of drive and resulting torsional excitation. The difficulty of detecting incipient failures in the field makes the performance of a thorough torsional vibration analysis an essential component of the turbomachinery design process.

Gas Path Sealing Options: Labyrinth, Floating Ring, and DGS

Different industries have different requirements associated with sealing the process gas from atmosphere. For integrally geared compressors the seals are placed outboard of the bearings and inboard of the compression stage. Table 3 lists the merits of various seals. For plant air compressors the most common sealing type is the labyrinth. Labyrinth seals are the least expensive option for sealing the gas path. However these seals can have a destabilizing influence on the rotordynamics. For the air separation market floating ring carbon seals are gaining in prevalence due to reduced the leakage rates. For the process industry and when compressing hydrocarbons, dry gas seals are frequently the only option considered. As the compressor stage is overhung, adding tandem or double DGS sealing acts to extend the overhang length and can have detrimental effects on rotordynamics. So the option of single and tandem or double DGSs for redundancy must be carefully examined.

Table 3 Benefits of Various Gas Seal Options for IGC Applications

Component	Advantage	Disadvantage
Labyrinth Seal	Lowest Cost	Elevated leakage rates (especially for later stages), destabilizing at high pressures
Floating Ring Carbon Seal	Reasonable Cost and lower leakage rates than labyrinth	Wear, rings can lock due to high pressure and be destabilizing
Dry Gas Seal (Single)	Low leakage rates, not known to be destabilizing	High Cost
Dry Gas Seal (Tandem)	Same as Single benefits – plus redundancy in the event of failure of one of the seals	Higher cost, increased complexity, and increases of shaft length leading to rotordyamic challenges
Dry Gas Seal (Double)	Same as Single benefits – plus avoids all leakage of process gas to vent.	Higher cost, highest complexity, slight increases of shaft length leading to rotodynamic challenges

Volutes

The complex flowpath shape of the volute derives from aerodynamic considerations. The intent is to maintain consistent gas velocity throughout the circumference of the volute. This leads to the most challenging structural design aspect of a volute being the "tongue" area. This area acts as a strong stress riser located at the point where the volute transitions to the external connection and is shown in Figure 28. Requirements that are used to set stress limits are based upon API617 and/or ASME BPVC, both of which produce comparable design stress limits. Optimizing the shape of the tongue area can limit the net area stress, but for high pressure applications this may not be sufficient design practice. Increasing wall thickness and introduction of external webs will help support the external structure and minimize global deflections and thereby minimize tongue stress. Wall thickness levels must be considered in connection with casting requirements that may limit the thickness. Alternatively high strength casting materials may be applied.

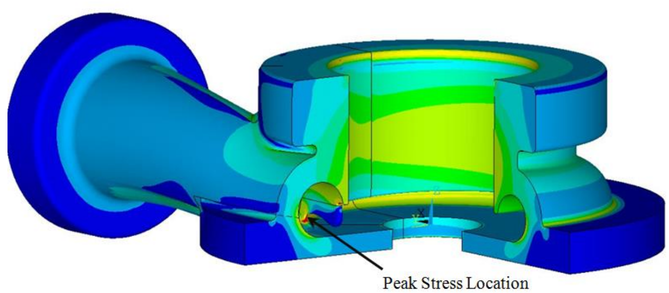

Figure 28. Peak Stress Location in Volutes

Structural Analysis of Assembly

For applications involving significant thermal gradients a thermal assessment of the unit using finite element analysis should be conducted. The thermal analysis provides an indication as to stress levels developed between connecting pipes as well as anticipated distortion of key interface components. Limiting factors are based upon flange loading limits and deflection limits for mating connections. An example of a high temperature inlet for a two-stage boost air compressor with no inter-cooling is shown in Figure 29. From the thermal analysis the piping configuration was optimized in terms of pipe lengths to achieve acceptable stress levels at the connections and other joints between components. It is also possible to then determine whether flexible inter-connections are required.

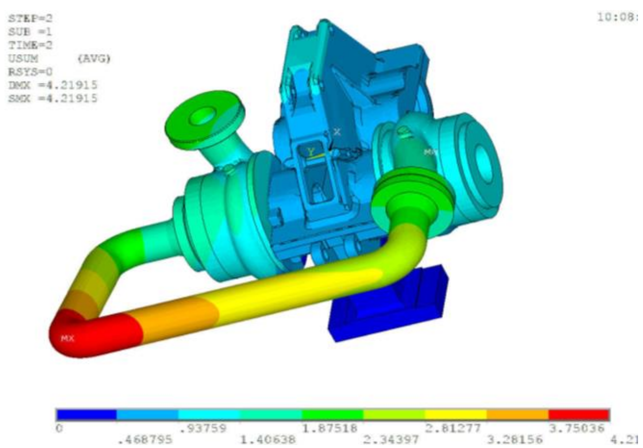

Figure 29. Structural Analysis of Compressor Core Reaction to Thermal Influences

Synchronous Motor, Induction Motor, and Mechanical Drives

The type of driver used for the IGC has an implication on the design as well. The implication is related to the transient torsional condition that the unit must experience during each start. If for example a synchronous motor is applied, then the level of dynamic torques during the acceleration up to speed is much higher than results when using an induction motor or turbine drivers, ergo a larger bull shaft gear diameter and potentially special coupling designs may be required.

A *synchronous electric motor* is an alternating current motor in which, at steady state the rotation of the shaft is synchronized with the frequency of the supply current; the rotation period is exactly equal to an integral number of alternating current cycles. In locking to the supply current large torsional pulse must be examined through start-up and short-circuit simulations of the entire rotating system. For the lateral rotordynamics operation is limited to single speed operation. Synchronous drives are most commonly found in ASU applications.

An *induction motor* will always rotate at a speed less than synchronous speed due to the rotating magnetic field produced in the stator that will generate flux in the motor and cause rotation, but due to the lagging of flux current in the rotor with flux current in the stator, the rotor will never reach its rotating magnetic field speed i.e. the synchronous speed. There are basically two types of induction motor that depend upon the input supply - single phase induction motor and three phase induction motor. Single phase induction motors are not self-starting motors which we will discuss later and a three phase induction motor is a self-starting motor. The torsional rotordynamic analysis of an induction motor driven unit is relatively straightforward in simply avoiding undamped natural resonances.

Mechanical drives may also provide the power to operate the IGC. Mechanical drives may be steam turbines, gas turbines, or expanders (which may also be of IG type.) Mechanical drives will frequently have a range of operating speeds that creates additional challenges for lateral critical speed avoidance, but the speed variability provides additional range of operation potential. Mechanical drives are more common for the petrochemical industry.

Flexible Couplings

An IGC does not in itself require any different type of coupling between driver and rotating equipment than typical industrial turbomachinery. Most frequently an IGC is coupled to the electrical motor or mechanical drive through a flexible coupling. The flexible coupling allows for reasonable tolerance in alignment and spacing between the driven and driving equipment during installation. The flexible coupling in itself may have resonances that require checking and also requires balancing. API 671 provides typical specification requirements for a flexible coupling.

Maintenance Considerations

A critical component of the design process of turbomachinery is ensuring that the machines can be maintained easily. Downtime to perform maintenance means lost production and any feature that allows easier and faster disassembly and re-assembly will enhance the operator’s ability to maximize the availability of the machine.

Several features are present on IGCs to enhance their maintainability:

- Removable gearcase cover – permits easy access to the bearings, vibration and temperature probes for inspection and replacement.

- Horizontally split bearings and, in some cases, seals – permits their replacement without requiring the removal of the rotors.
- Appropriate design of auxiliary systems – ensures components (e.g. pipes, filters, valves, etc.) are not in the way of removal of the core items (as above) so that excessive other disassembly is not required to carry out the work.

Naturally, it is important for operators to also ensure that they have purchased appropriate quantities of spare parts to also minimize the downtime. Especially in the case of engineered-to-order machines, the OEM should not be expected to stock spare parts for every different unique design. However, by judicious development of standardized building blocks for each frame size, some stocking of commonly used rough stock material could help minimize the manufacture of replacement parts.

CONCLUSIONS

The reader should have gained an appreciation as to the range of markets served by integrally geared compressors and understand the key parameters that define an efficient and reliable design. Several of the key aspects thereof are restated:

1. An integrally geared compressor competes with various machinery types depending upon the application served.
2. Each application imposes unique design constraints onto an IGC. Still, several factors tend to be common across each application: 1) maximizing unit load on bearing helps reduce mechanical losses, 2) larger capacity applications are more mechanically efficient, 3) fundamental aerodynamics limit cycle performance, inter-cooling stages improves efficiency.
3. An IGC offers the advantage of high efficiency due to the inter-cooling that can be provided between each stage. The IGC is also a compact solution for many applications. However, sealing, rotordynamics, structural integrity, are more challenging than some competing machines and need specific care in the design stage.
4. Proper design of an IGC for any given application requires balancing numerous competing elements.

NOMENCLATURE

Variables

$C_{\theta 1}$	= Impeller Inlet Tangential Velocity	(deg)
$C_{\theta 2}$	= Impeller Exit Tangential Velocity	(m/s)
D_2	=Impeller Diameter	(m)
F_T	= Transmitted gear force	(N)
F_r	= Radial gear force	(N)
F_a	= Axial gear force	(N)
h	=Enthalpy	(J/kg)
k_{mxx}	= Mesh Stiffness in x-x Plane	(N/m)
k_{myy}	= Mesh Stiffness in y-y Plane	(N/m)
k_{mxy}	= Mesh Stiffness in x-y Plane	(N/m)
k_{myy}	= Mesh Stiffness in y-x Plane	(N/m)
N	= Rotational Speed	(rpm)
p_d	= Pinion Pitch Diameter	(mm)
P	= Power	(kW)
P_{tot}	=Total parasitic power loss	(kW)
$P_{journal}$	= Parasitic power loss from journals	(kW)
P_{thrust}	= Parasitic power loss from thrust films	(kW)
$P_{windage}$	= Parasitic power loss from windage	(kW)
P_{mesh}	= Parasitic power loss from gear mesh	(kW)
T	= Torque	(N-m)
Q	=Volume Flow	(m^3/s)
Q_{tot}	= Total required lube flow	(l/min)
Q_{brg}	= Bearing required lube flow	(l/min)
Q_{mesh}	= Gear mesh required lube flow	(l/min)
U_1	= Impeller Inlet Tip Speed (m/s)	(m/sec)
U_2	=Impeller Exit Tip Speed	(m/sec)
Wx	= Work	(kW/sec)
β	= Helix Angle	(deg)
ϕ	= Transverse Pressure Angle	(deg)
ϕ	=Flow Coefficient ($4Q/\pi U_2 D_2^2$)	(-_)
ψ	=Head Coefficient ($\Delta h/U_2^2$)	(-)

Subscripts

1	= Compressor Inlet
2	= Compressor Discharge
r	= Radial
a	= Axial
θ	= Tangential

Acronyms

API	= American Petroleum Institute
ASME	= American Society of Mechanical Engineers
ASU	= Air Separation Unit
BAC	= Boost Air Compressor
BOG	= Boil-Off Gas
BPVC	= Boiler Pressure Vessel Code
CAPEX	= Capital Expenditure
DNV	= Det Norske Veritas
FGC	= Fuel Gas Compressor
FNC	= Feed Nitrogen Compressor
GO2	= Gaseous Oxygen
HCF	= High cycle Fatigue
HPC	= High Pressure Compressor (Methane)
IGC	= Integrally Geared Compressor
IGV	= Inlet Guide Vane
LCF	= Low Cycle Fatigue
LN2	= Liquid Nitrogen
LNG	= Liquid Natural Gas

LO2	= Liquid Oxygen
LU	= Liquefaction Unit
MAC	= Main Air Compressor
OPEX	= Operating Expenditure
RNC	= Recycle Nitrogen Compressor
VDV	= Variable Diffuser Vane
VRU	= Vapor Recovery Unit

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