THE USE OF INTEGRALLY GEARED COMPRESSORS
BASED ON TWO INDUSTRIAL GAS COMPANIES' EXPERIENCE

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ABSTRACT

The industrial gas industry has been using integrally geared compressors for over 35 years. This industry has the most experience in the design aspects, the commissioning, and the operation of this style compressor. With the inclusion of integrally geared compressors into API 617, and with use of these machines being more accepted as process compressors, these experiences should be shared.

The compressor manufacturers' all have designs and technology they are willing to show and sell to various industries. The intent of this tutorial is to provide some typical design aspects seen, some guidance, and then some issues concerning the integrally geared compressors from the perspective of a user community that has high familiarity with this style of compressor.

INTRODUCTION

The purpose of this tutorial is to give the experiences of using integrally geared compressors (IGCs) from perspective of two industrial gas companies. Between the two companies over the last 40 years, it is estimated that over 3000 integrally geared compressors have been purchased, installed, and operated. The large majority of these compressors are in air or nitrogen service. The compressors range from small product machines to high horsepower, high flow compressors for large integrated complexes.

The direction of the industrial gases business is such that very sophisticated integrally geared compressors are being required. In

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conjunction with this business need, the various suppliers of this type compressor have improved their technology so that the designs meet the companies' required operational and reliability needs. This sophistication includes supplying compressors with six to eight stages, with multiple services, that compress different gases, and that have steam turbine drivers.

Because of the growing use of these compressors, the API 617 (2002) standard for centrifugal compressors has been modified in the Seventh Edition to include a chapter for integrally geared compressors. Beatty, et al. (2000), gave a good report on new integrally geared compressor designs from a manufacturer's viewpoint at this symposium in 2000. The tutorial being presented here has been written for a different purpose; that is to give examples, design issues, and experiences from two companies that operate and use this style of compressor as a normal part of their business.

HISTORY

Most people are aware of the historic use of integrally geared compressors in the industrial gases industry. All air separation plants utilize what is known as a main air compressor (MAC). This compressor is an atmospheric suction compressor that normally has a 5 to 7 bara (70 to 100 psia) discharge pressure. Flows will be from 500 cfm (850 m3/hr) to 250,000 cfm (425,000 m3/hr). These compressors have been used since the early 1960s, although the available sizes of these machines have steadily increased throughout the years.

For liquid nitrogen and liquid oxygen production, nitrogen recycle machines are required. These compressors will take nitrogen product from the separation process and will discharge at about 400 to 500 psia (30 to 40 bara). These machines have been designed as integrally geared machines since the early 1970s. Power ranges on these machines can be as high as 20,000 hp (15,000 kW).

In the last 15 years, booster air compressors (BAC) have been used in the industrial gas industry. These machines are similar to the recycle machines, but they take suction from the MAC discharge after a drying process and discharge at a pressure that is a function of the oxygen pressure delivered to a pipeline. The discharge pressures will be as high as 800 to 1200 psig (70 to 85 barg).

Other services that often use integrally geared compressors are nitrogen product and oxygen product. The latter application requires special considerations regarding the materials of construction, seal system design, and instrumentation. Pressures and flows vary widely on these services.

A summary of the services and sizes of integrally geared compressors used by industrial gas companies is shown in Table 1.

Table 1. General Sizes of Various Compressors Used in the Industrial Gas Industry.

Service	Inlet Pressure psia/(bar-a)	Outlet Pressure psia/(bar-a)	Flowrate Icfm/(m^3/hr)	Power Horsepower/ (Kilowatts)
Main Air Compressor (MAC)	14.5 (1.0)	70-100 (5.0-7.0)	500-250,000 (850-425,000)	1,000-58,000 (750-43,000)
Recycle Compressor	90 (6.0)	400-500 (30-40)	2,000-17,500 (3,400-30,000)	2,500-23,000 (1,800-17,000)
Booster Air Compressor (BAC)	90 (6.0)	150-1,000 (10.0-70)	1,000-33,000 (1,700-56,000)	500-33,500 (400- 25,000)
Product Compressor	Various			

The different reasons for using an integrally geared compressor in lieu of other designs have been documented in various other articles through the years. As a summary, the major reasons for their use in the industrial gases industry include:

● Efficiency due to interstage cooling, axial gas inlets, the use of two or more speeds for the impellers.

● Lower costs.

● Well suited for motor drives.

● Multiservice easily accommodated.

● High head impellers are an option.

● Easily accommodates a wide variety of seal types depending on service requirements.

● Compact design has a small installation footprint and simple foundation requirements.

● A significant degree of packaging is possible even on large units, which reduces installation costs.

● Elimination of the need for high speed couplings.

● Better access for maintenance and turnaround activities.

There are now many examples where this style of compressor is being used more often in the traditional petrochemical services. Some examples of these units used in the new services will be shown here. Other articles in this symposium and in other publications also give examples.

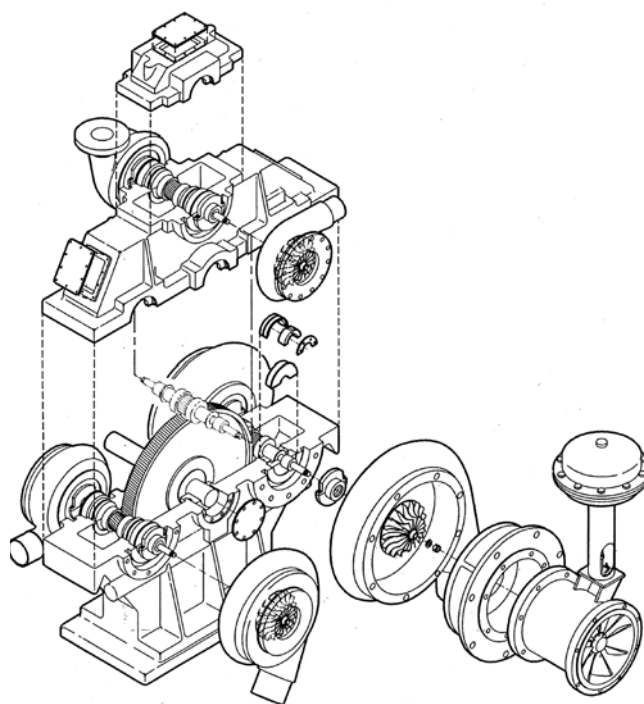A cutaway of a typical integrally geared compressor is shown in Figure 1.

Figure 1. General Cutaway of Integrally Geared Compressor.

GENERAL DIRECTION OF INTEGRALLY GEARED COMPRESSORS

Before going into some general design issues, there are two specific directions that the air separation industry sees in the designs of integrally geared compressors.

Multiservice/Multistage Integrally Geared Compressors

Although the history of integrally geared compressors shows a preponderance of two pinion, three or four stage designs with only two pinions, the current typical design from the industrial gas companies uses up to six stages and three pinions on a gearbox as an accepted design. These multistage machines will have three pinions located at the 9, 12, and 3 o'clock positions. Application analysis is often done on the combining of services (MAC plus BAC or MAC plus product N$_2$) and is based on the costs, operating issues, and risk. In the last several years, *a fourth pinion is being used more often*. This pinion is normally located near the 6 o'clock

position, and is used for either additional stages or as a drive pinion. More will be said about this drive pinion later.

Steam Turbine Drives

Another area where integrally geared compressors are often being adapted is to marry them to steam turbine drivers. This direction is from the industrial gases customers in the petrochemical businesses that need oxygen for their process and who have excess steam available. This direction has been in the larger flow size range, although examples can be found in smaller sizes. Integrally geared compressors can be mated to steam turbines in three basic ways. All have their own positive and negative aspects.

The first method is to have the steam turbine drive a speed decreasing gear set, which drives the integrally geared compressor at the normal bullgear input shaft. The positive benefit of this design is that each major component, including the integrally geared compressor itself, is designed by normal practices of the suppliers. For this configuration, the design risk is low. The negative aspects of this design include the cost for the gear unit, an efficiency hit due to the gear unit, and the possible long-term reliability or maintenance issues with the extra gear and bearings.

The second method of making a steam turbine into an IGC is by a pinion drive (Figure 2). This is a fairly new development over the last four to five years, but has gained acceptance in our industry. In this case the turbine speed and one pinion speed can be matched so the cost and the losses of the external gearbox disappear (although there are still some mechanical losses on the pinion mesh and bearings). Although costs and efficiency are improved, there are still some issues with this design. One of the issues is in the rotordynamics of the drive pinion/coupling. Because there is now a coupling attached to the pinion, there is a large overhang weight on one side. Since the pinion geometry (bearing locations) is often set by the gearbox and scroll spacing, the rotordynamics end up playing a larger part in the design. In the more difficult applications, this is solved in one of two ways: by using a gear type coupling to lower the overhung weight, or installing an intermediate bearing shaft along the coupling length.
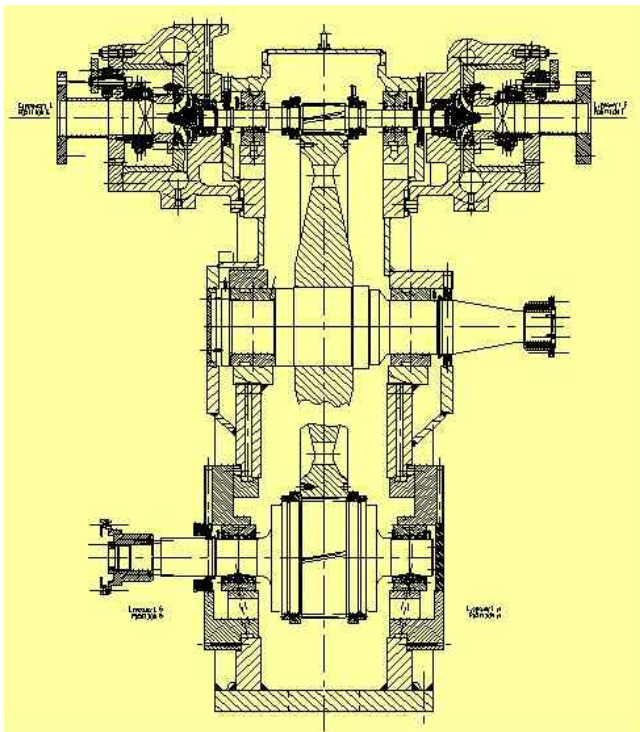
Another issue arising out of this drive method is that the entire drive torque now must be transmitted across one mesh. The normal design of an integrally geared compressor, which is driven at the bullgear shaft (either by motor or steam turbine with speed increaser), will split the torque across the number of pinions. With the use of a pinion drive, the gearing must be designed to take this full torque. The toothing on the bullgear also sees bending forces in both directions with a pinion design (one direction only when driven by the bullgear shafting).

The final method of making a steam turbine into an IGC is a modification of the method just explained. The steam turbine is still coupled directly to one pinion, usually on the horizontal split line. On this same pinion an impeller is installed on the other end. The advantage here is that the gear mesh load is decreased by the torque used to compress the gas by the attached impeller. This allows for a decrease in gear width. The rotordynamics for this pinion becomes slightly more complicated.

One final point on integrally geared compressors that are steam turbine driven is the speed range. The units *are inherently designed for single speed applications*. This is unlike the inline machines that will have a much wider speed range.

SPECIFIC EXAMPLES OF COMPRESSORS

The following examples show integrally geared compressors in configurations that are being used in the industrial gas industry. The purpose of showing these examples is to give a frame of reference of presently considered modern designs that are used. All examples have been operating satisfactorily for years with the exception of Example 5, which will startup at the end of 2003.

Example 1—Combined Recycle and Feed Compressor

Example 1 is a motor driven combined nitrogen feed/recycle compressor in liquifier service. The feed compressor section uses two stages to compress 33,000 lb/hr (11,000 Nm^3/h) of nitrogen from 14.0 psia (0.965 bara) to 78 psia (5.4 bara); the recycle uses three stages to compress 210,000 lb/hr (72,300 Nm^3/h) of nitrogen from 76 psia (5.3 bara) to 358 psia (25.0 bara). The total power consumption is 5600 kW. This is shown in Figure 3.

Figure 2. Cutaway of Pinion Driven Integrally Geared Compressor.

Figure 3. Combined Feed/Recycle Compressor.

Example 2—Oxygen Compressor

This is a motor driven oxygen compressor that uses two stages to compress 70,000 lb/hr (21,700 Nm^3/h) from 115 psia (8 bara) to 385 psia (26.5 bara) with a power consumption of 1550 kW. For

safety considerations this compressor uses multiported buffered labyrinth shaft seals that are monitored for performance. The materials of construction have also been selected to reduce the damage in the event of an internal compressor failure. Figure 4 shows this packaged unit.

Figure 4. Oxygen Compressor, Integrally Geared Design.

Example 3—Steam Turbine Driven Compressor

This is a traditional three pinion integrally geared compressor in main air plus booster air service. The rated power of the system is 6700 kW. The steam turbine drive is a special purpose condensing steam turbine without extraction. The turbine drive transmits its torque to the compressor through a separate speed-reducing gearbox. This configuration allows standard components (turbine, gear, compressor) to be used with minimal technical risk. Figure 5 shows the installation.
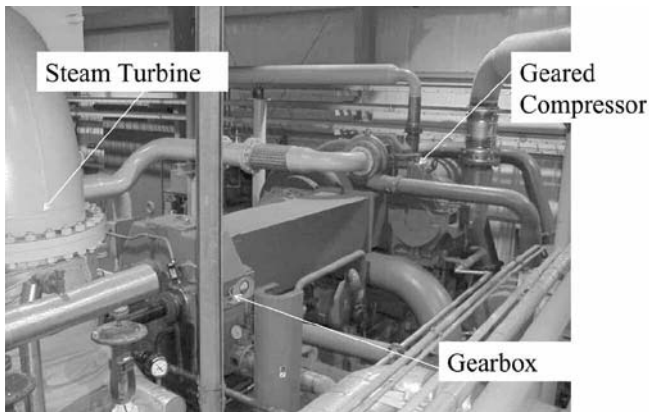

Figure 5. Steam Turbine Driven Integrally Geared Compressor with Gearbox.

Example 4—Steam Turbine Driven Booster Air Compressor

Example 4, as shown in Figure 6, is a double-ended steam turbine, with a main air compressor on one side and a booster air compressor on the other side. The focus of this example is on the booster air compressor. Again, this machine has three pinions for compressor stages. There is a fourth pinion on this drive, which transmits the entire torque from the steam turbine to the compressor bullgear. The use of this pinion as a drive gets rid of the need for a separate speed reduction gear, as was used in Example 3. An interesting note on this example is that the bullgear is also connected to a small 1000 kW induction generator running a four-pole speed.
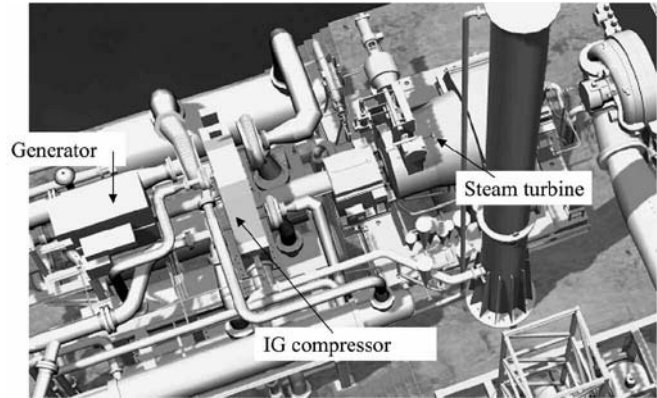

Figure 6. Integrally Geared Compressor with Pinion Drive Direct from Steam Turbine.

Example 5—Large Main Air Compressor

This example is a large motor driven air compressor that uses four stages to compress 120,000 acfm of atmospheric air from 14.3 psia (0.98 bara) to 112 psia (7.7 bara). The power consumption is 20,500 bhp (15,300 kW). This compressor has variable inlet guide vanes (IGV) on all four stages to meet the rangeablility requirements of the application. The synchronous drive motor is started across the line. This machine is shown in Figure 7.
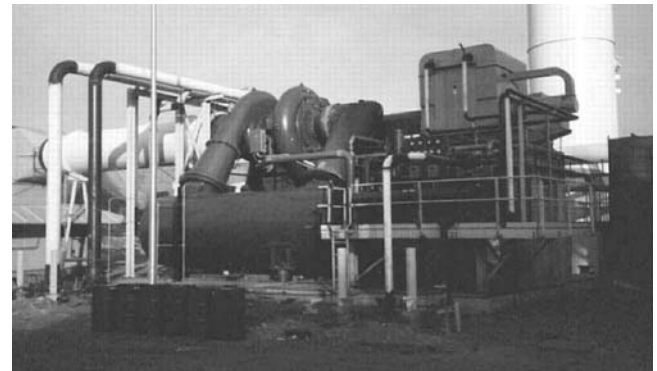

Figure 7. Large, 20,000 HP Integrally Geared Compressor.

Example 6—Compressor in Carbon Monoxide Service

Carbon monoxide (CO) is a toxic and flammable gas that is used in many petrochemical synthesis gas processes. Figure 8 shows one of these compressors rated at 2700 kW that is motor driven. For cost and power reasons, an IGC was selected. This compressor has six stages utilizing three pinions and three sets of inlet guide vanes. The shaft seals (six of them) are single dry gas seal with a labyrinth backup. The seal design was bidirectional and was designed to minimize free carbon formation. The compressor was also designed with other special considerations for CO service, including welded seal gas lines and seal welded utilizing stainless steel tubes and tubesheets in the coolers.

DESIGN ISSUES AROUND
INTEGRALLY GEARED COMPRESSORS

Some design issues must be addressed when looking at IGCs versus the inline or between-bearing compressors. This section will highlight the design issues the industrial gas companies have seen and normally review on new compressor purchases.

Gearing Input Speed

One of the most predominant design aspects of the integrally geared compressor is the gearing itself. The bullgear is large in
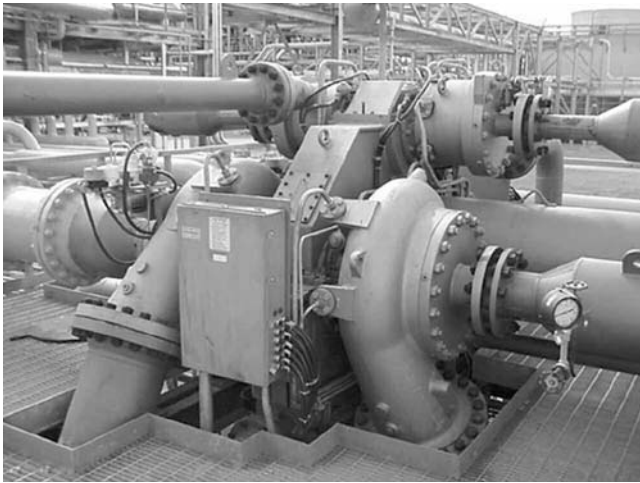
Figure 8. Carbon Monoxide Integrally Geared Compressor.

comparison to the normal speed increases gearboxes. Speed ratios are normally in the 8 to 16 range. The need for the large bullgear is to allow for the stage scrolls to be physically installed on the gearboxes themselves. Although we have shown some examples of steam turbine driven integrally geared compressors, the vast majority of them are still motor driven. Any proposal from suppliers should always indicate input drive speed required. Typically, the higher the drive speed, the lower the cost of the compressor. This is because the bullgear/gearbox is smaller in size, less metal. The input speed is normally set by the frame size of the compressor, but there may be some flexibility depending on the size.

It is always the case that the lower the flowrate, the higher the input speed that will be possible. *One warning is when the supplier is providing a compressor that is near the breakpoint of input speed (say four-pole to six-pole).* The supplier will often quote the four-pole input speed, and then design the gearing such that there are very high pitchline velocities. The lower input speed selection is often less risky technically, but higher in cost. More will be said later concerning pitchline velocities. One other aspect of input speed that should be mentioned is that there is only a slight difference in motor costing between two-pole and four-pole designs. There is a relatively large cost differential to go from a four-pole to six-pole design.

Gearing—Design

Getting back to the gear design itself, the various worldwide suppliers of integrally geared compressors have different methods of providing the gearing. The possibilities range from sourcing the detailed design and manufacture of the gear from a separate gear company as a partner, to designing the gearing and then sourcing the manufacturing from an outside supplier, to complete designing and manufacturing of the gearing within their own company. All methods have been successful. The *purchaser should be aware of how each possible supplier designs and fabricates the gearing*, for an understanding of who owns the details of this critical technology.

Gear designs are most often based on the supplier's inhouse gear design methodology (or from their subsupplier). Most use similar methods to AGMA 6011 or DIN 3990, although, from the author's knowledge, the calculations are slightly modified. *Suppliers generally have acceptable designs for the industrial gases industry, with gear design issues being rare.* Of the few problems that have been seen through the years, they are often found to be other than a design issue, such as gear alignment, quality control, and lubrication contamination. The new API 617 (2002) standard has been issued with specific gear design requirements. The authors have not seen the results of a machine purchased to this standard yet,

although the feedback from the suppliers indicates it will result in a more conservative design. It is prudent to ask for a rating based on AGMA 2001-C95. Beckman and Patel (2000) discussed and compared many of the gear design standards in their paper.

The one area of gearing that has shown itself to be a real issue is the pitchline velocities. These should be reviewed and addressed for every offering. Gear manufacturers have pushed pitchline velocities higher and higher. There are some examples of gear sets having as high as 39,000 ft/m (200 m/sec); however, this is for lower speed ratios. In the industrial gases industry, 33,500 ft/m (170 m/sec) has been seen. A vast majority of machines have been lower than 30,500 ft/m (155 m/sec). There is one example in the industry of a 30,000 hp compressor operating a 32,270 ft/m (164 m/sec) pitchline velocity for eight years. From the authors' viewpoint, the area where *questions should be raised with the supplier is when the pitchline velocity is in the 30,500 to 32,500 ft/m (155 to 165 m/sec) range*. There is no doubt that the technology is driving to higher pitchline velocities. The question with these higher velocities is always on the long-term reliability.

If one is confronted with a high pitchline velocity machine, one should ask additional questions. One such would be the gear helix angle and the meshing velocity. Meshing velocity is the velocity of the gear as the contact goes from the engagement side to the disengagement side. It is based on the pitchline velocity and the helix angle. Typical warning levels are greater than 167,000 ft/m (850 m/sec) for meshing velocity and helix angle less than 14 degrees.

One subject that must be mentioned that is related to the pitchline velocity is the manner of gear lubrication. Depending on the supplier, different lubrication methods are found. These include oil sprays into the engaging mesh, or on the disengagement side, and a combination of the two. Gear lubrication, pitchline velocity, and meshing angle are all interrelated. The higher the pitchline velocity, the higher the centrifugal forces acting to throw the oil from the gear; the lower the gear angle, the higher the pumping velocity of the oil along the mesh.

Pinion gear materials are typical high strength alloy steels. *Pinions gears are normally hardened by either case carburizing (seen more often in European designs) or nitriding (more often seen in the US).* Bullgear designs are similar, although they may not always be case hardened, but may be only through hardened. In the industrial gases business, the different methods are generally accepted. There are instances where bullgear wear has been seen on noncase hardened gears, but only after a 10 to 15 year life with some overloading over original design by upgrading or supercharging the compressor above the original design.

One further note on the gear sets. Presently *the industrial gas companies will insist that the gear set design be such that replacing an individual pinion is possible without replacing the entire gear set* (all pinions and bullgear). Technology has improved to where manufacturing to tolerances between main rotors and spare rotors are low enough that changeouts can occur without any large wearing issues. Bullgears are not always purchased as part of the capital spares philosophy for a given plant. Of course this depends on the criticality of the plant. Presently, depending on the size of the compressor, various manufacturers will stock pinions for many of their frames so that they will often have a spare pinion available on short order if needed.

Motors and Motor Starting

Motor drives are the most common way of driving an integrally geared compressor. Both synchronous and induction (asynchronous) motors are used. Induction motors are most often used at less than 5000 kW due to their cost. Above 10,000 kW, synchronous motor drivers are often used due to power evaluation and reference issues. Some motor manufacturers are now introducing induction motor designs up to 15,000 kW. In the 5000 to 10,000 kW range there is a mixture of motor types. Induction motors are lower in cost, but lower in efficiency. Induction motors also have less maintenance due to the lack of an exciter. *In the industrial gas industry, the user normally*

supplies the motor. There are two reasons. The first is cost, as suppliers will often markup the motor pricing. Secondly, the electrical system coordination (reduce voltage starting, etc.) is normally in the user's scope.

Integrally geared compressors, because of the large bullgear mass, have large moments of inertia. This is a critical issue for motor designers and for electrical systems engineers. Acceleration times for most IGCs with motor drivers are in the 8 sec to 25 sec range. Very large IGCs may take as long as 60 sec for starting. This starting time, when combined with the use of synchronous motors, leads to the next issue, which is the torsional design. With a synchronous motor, there is a high current/torque fluctuation on startup. This is called the pulsating torque, and is a function of the slip speed frequency during startup. The torsional stresses that are applied to the drive shaft are important considerations. All suppliers will analyze the torsional stresses and make appropriate design modifications to the coupling. The industrial gas companies have, in a few cases, found that a highly damped coupling (elastomer type) is needed. *More often, a coupling with a higher torque rating would be required because of this pulsating torque*.

When a synchronous motor is used, solid pole motor designs are now the most typical type designed and sold. This solid pole motor design causes high pulsating torques and relative low pull-in torque. Because of these factors, *the number of starts should be stipulated to the compressor supplier*. Low cycle fatigue, as well as torsional resonances, are issues for the coupling design if not addressed early in the design phase. Close coordination between the motor supplier and the compressor supplier is essential if the pulsating torque and the startup torque are to be managed correctly. Since synchronous motors may need a reduced voltage starting scheme due to the inrush currents and the low pull-in torque, the time it takes to come up to speed is longer. When reduced voltage starting is used, low cycle fatigue/torsional analysis will show lower pulsating torque, but a longer time interval around the torsional critical.

A note for users: *make sure all suppliers know what style motor will be used during the specification stage*. It will make a difference to the supplier's bid. Also, if an induction motor is used, a definition of the full load speed (synchronous speed minus slip) is needed. This is important for the compressor supplier and its gearing/pinion speed designs.

Integrally geared compressors, as well as all other motor driven compressors, have performance and mechanical aspects that must be addressed if there is frequency variation in the electrical supply (for some locations this is found to be ±5 percent). Again, one must stress that there is a coordination need from the mechanical engineer on the compressor design, the motor designer, and the electrical system engineer.

Impeller Design

IGCs are provided with close and "open" impellers (technically semi-open). They are fabricated from castings, milled forgings with and without welded or brazed covers. Because of their heritage as air compressors the *most prevalent material of construction is stainless steel with 17-4ph and similar alloys* being common.

While single shaft compressors rarely use open impellers, they are the dominant configuration for IGCs. The reasons for this are numerous. They are less expensive to manufacture and intrinsically stronger than closed impellers. They more easily accommodate the use of inducers and can be designed for higher tip speeds. A single casting can be optimized for a wide range of aerodynamic conditions by machining the blade profile and matching the stationary shroud to its contour. This can be a significant issue when considering future rerates or upgrades. It can also be beneficial if a replacement impeller is required because of an equipment failure. Even if the user does not have a spare impeller, the supplier may have a casting or forging that can be quickly made

into a replacement in an emergency.

Open impellers are not without their issues. *The principal issue is that in the event of an impeller rub the damage is likely to be much worse than with a closed impeller*. The likelihood of the event can be reduced by having good antisurge protection and by using axial position monitoring of the pinions. Some additional margin of safety is also achieved by the use of high speed thrust bearings instead of thrust collars. Where there are safety issues associated with an impeller rub, material combinations also play an important role. IGCs in oxygen service use open impellers, but the use of bronze stationary shrouds has been demonstrated to prevent ignition even when a serious rub has occurred. Another issue, which has less substance, is that a closed impeller is significantly more efficient than an open impeller because of leakage at the blade tips of open impellers. *There is no apparent penalty assessed against open impellers by suppliers who offer both styles* and, in actual field measurement, changes in tip clearance within reasonable limits do not show a perceptible change in efficiency.

In an effort to improve efficiency and better optimize performance particularly on larger compressors *some suppliers are offering a combination of open and closed impellers*. This permits a better matching of specific speeds by using, for example, an open first stage impeller and a closed second stage impeller on the same pinion. It also allows for the open impeller clearance to be set fairly close, with the closed wheel allowed to move away from the inlet. An example can be seen in Figure 9.
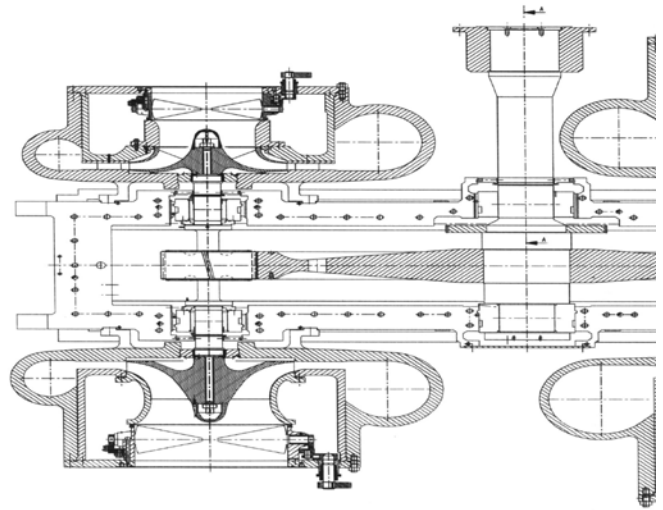

Figure 9. Closed/Open Impeller Configuration.

While it is not a consideration in a single shaft compressor, the ability to be able to remove and replace an impeller in the field can be a service requirement on an IGC. This is because some designs require the removal of one or both of the impellers in order to install or remove the pinion from the gearbox. This is true on any design that uses a dry gas face seal or other cartridge type as a shaft seal. Suppliers use virtually the entire range of impeller fitting techniques to attach the impellers to the pinions. These include thermal interference taper fits, hydraulic interference taper fits, bolted/flanged couplings, toothed couplings, polygons, keyed line-to-line fits, and fits that rely on the friction of the metal-to-metal contact between the end of the pinion and the back of the impeller to drive impeller. The thermal taper fit has largely fallen from favor because it is virtually impossible to remove the impeller in the field. Even when done in the shop, removal can result in bending of the stub-shaft leading to additional rework before a new impeller can be installed. The use of polygons, toothed couplings, or "friction drives" requires precise tightening of the center tension bolt to work properly. Hydraulic bolt tensioning is the most reliable

method and is simple to do with the right fixtures. Hydraulic taper fits also work very well providing the impeller is not overdrawn onto the taper during assembly. Also the tolerances on the taper can complicate the fitting of a replacement impeller. With any of these approaches, the use of an indexing pin allows removal and reassembly without disturbing the rotor balance. *All such methods have been used and accepted in the industrial gas industry, although some are more preferred over the other.*

The major suppliers of IGCs are somewhat divided on the issue of routinely overspeed testing of impellers prior to installation. Two of them overspeed test all impellers prior to installation. One of them only overspeeds them as part of mechanical assembly testing when this testing is done with a variable speed driver. In this last case the primary intent of the practice was to verify that the thermal taper fit on the impellers was properly settled rather than to test impeller integrity. It can, however, be used to satisfy a user's requirement to have the impellers oversped. Spare impellers are not oversped. A fourth supplier does not overspeed his impellers at all. *The point may be moot since, in the experience of the authors, none of these suppliers has had an impeller failure that is attributable to something that an overspeed test would detect.*

Thrust Collars

A common design feature of IGCs is to use thrust collars on the pinions that ride on a thrust fellow on the bullgear instead of high speed thrust bearings. Figure 10 shows a typical thrust collar design. This design approach has several attractive benefits; it also has some downside, which needs to be kept in mind. The benefits are lower first cost and lower mechanical losses. The downside is reduced confidence in the axial position of the pinion and impellers. This is because this position can be affected not only by the axial clearance of the bullgear thrust bearings but also by tipping of the bullgear through the radial clearance in the bullgear journal bearings. Usually this is not an issue because the large mass of the bullgear makes it very resistant to lifting in the bearings. However, in *installations with long, heavy coupling spacers, the outboard bullgear bearing can be unloaded to the point that this is a real concern.* Also, in applications where even a light touch of the impeller to the shroud is deemed unacceptable, designs employing thrust collars should not be used. An example of this would be an IGC in oxygen service. Thrust collar failures have occurred in the industry through the years. The few failures that the authors have experienced are more related to off-design conditions or strong surges.

Figure 10. Pinion Showing Thrust Collars.

Seal Design

IGCs are available with a wide variety of shaft seals although all suppliers do not provide the full range of seals. The most common type of shaft seal is a stationary aluminum labyrinth. An example is shown in Figure 11. This is satisfactory for low pressure, nontoxic, nonflammable applications. These seals can be easily provided with ports to permit the injection of buffer gas or to reduce the amount of gas lost to the atmosphere. At higher pressures, rotating labyrinth seals (Figure 12),

brush seals, or carbon ring seals can be provided depending on the supplier. Any of these seals can also be provided with buffering or gas recovery ports. It should be recognized that the performance of rotating labyrinth seals that are close fit to the stationary (i.e., abradable material or carbon bushings) can deteriorate over time, particularly if the compressor is stopped and started frequently or is subjected to surge. *It is recommended that the user review the design of seals of this type carefully so that future maintenance requirements are understood.* For very specialized applications, such as oxygen compressors, multiported labyrinth seals that prevent process gas leakage along the shaft and atmospheric leakage in along the shaft can be provided. For this type of application it is very important to be able to accurately monitor the integrity of the seal system, and care needs to taken that pressure and pressure differentials in the seal are being accurately measured. Finally, *dry gas face seals* have been successfully used on IGCs by most of the suppliers of IGCs. They *have shown to be highly reliable and very low in maintenance* in industrial gases applications. Their high first cost is the primary inhibitor to their use and so they have been used primarily where there are toxicity or flammability issues or significant savings in loss prevention of the compressed gas. It must be noted that process gas leakage to the atmosphere for normal air separation industry gases is allowable (namely: air, nitrogen, oxygen). In the case of nitrogen or oxygen, however, proper ventilation must be provided.
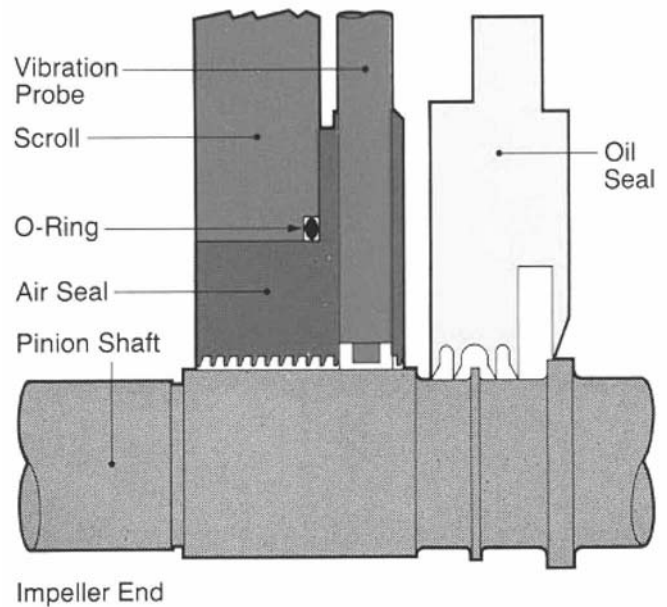

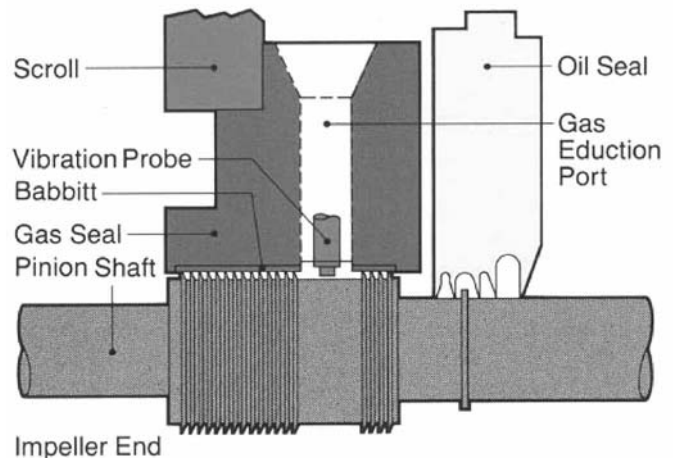
Figure 11. Typical Stationary Labyrinth Seal.

Figure 12. Typical Rotating Labyrinth Seal.

Bearing Design

The life of a bearing is much more complicated in an IGC than in a single shaft compressor. In the latter the bearing loads are only a function of rotor weight and do not change appreciably with load. The pinion bearings in an IGC have to deal with both the rotor weight and its significant overhung masses and the variable loading from the gear teeth. The bearings tend to be more highly loaded, and the load vector on the bearing can shift significantly, particularly on the updriven pinion. In a fouling gas service the balance condition of the rotor can change with time. This means the bearing must have good damping and sufficient load capability to deal with the vibratory load, in addition to the steady-state loads, without fatigue damage. Today the majority of IGC suppliers use tilting-pad bearings with positive preload that meet these design requirements very well. One supplier has been very successful with tilting-pad bearings with zero preload. Given this as a starting point, there are some additional realities of the application to be considered.

Good performance of the gearing is highly dependent on having good tooth contact, and good long-term performance of the bearings is highly dependent on the bearings not being edge loaded. Even if gearing exhibits good static contact, it may not mean that it will have good running contact, which is really what one wants to achieve. Today the technology exists to be able to predict what the static contact must be in order to have good running contact when the gearing is running at design loads. Unfortunately this prediction assumes that the stackup of manufacturing tolerances works out well and that no distortion of the gearbox occurs because of things like piping strain applied to the compressor volutes. These issues become more important as the compressor increases in size, and/or in designs with single ended pinions that do not have offsetting loads at either end. *Therefore, a means of adjusting the centerline of the bearing relative to the bearing bore can be a valuable design feature.* Using eccentric bearing housings or shimming pad supports are some of the ways of doing this. In the authors experience it is not uncommon to see evidence of edge loading on pinion bearing pads in IGCs over time, particularly when the pads have a line contact to the bearing housing. For that reason bearing designs that allow compliance in both directions are preferable. *For these same reasons, gear contact pattern should be checked on any integrally geared compressor fairly soon after the initial startup.*

On larger integrally geared compressors, the suppliers are often balancing the bearing loads, rotordynamics, and bearing surface velocity. The bearing manufacturers often show designs with high allowable loads and/or high surface velocities. These loads and velocities should be reviewed. *Bearing loads at the design point above 320 to 350 psi (22 to 25 bar) should be reviewed* with long-term maintenance in mind. Also, *any bearing surface velocity above 260 to 295 ft/sec (80 to 90 m/sec) should be similarly reviewed.*

Another area of some controversy is whether the load should be on the pad or between the pads in a tilting-pad bearing. In fact both conditions can exist in a given compressor and can change due to changes in load. As a practical matter, *there is little if any difference in the observed performance of the compressor from a rotordynamics standpoint whether the load is on the pad or between the pads on an IGC*, based on periodic vibration surveys over a long period of time.

Rotordynamics

Rotordynamics is always an issue on these compressors, as it is on all rotating pieces of equipment. One of the differences for most (or all) IGCs is the fact that they are inherently single speed machines. For motor driven IGCs, the speed is defined by the motor speed. Even for steam turbine drivers, the speed range is very small.

The industrial gas companies rarely see any rotordynamics issues on rotors on the split line. The suppliers all have standard guides used to predict and design rotors without many issues. *There have been some occasions with three pinion designs where the top rotor may have some rotordynamic issues.* The authors can think of a handful of designs with field issues over the last 10 years. The top pinion designs tend to run the highest in speed due to aero needs. Because of this, the rotors will have fewer teeth, thus smaller diameters. The gearbox design generally determines the bearing span, and for the top pinion, this tends to be longer than desired by the lower splitline rotors. Thus the top pinion often ends up as a longer and thinner one than optimum for mechanical design purposes. *These physical limits tend to make the rotordynamics design of the top pinions more difficult.*

In terms of the acceptable criteria for the logarithmic decrements, the integrally geared compressor has similar requirements as other compressors. Typically log decrements will be above 0.1, and a warning should be raised *at any pinion that has a log decrement nearing 0.08.*

PERIPHERY DESIGN ISSUES

Gas Coolers

The requirements for a compressor with high efficiency, low cost, and small footprint have led to the development of low pressure drop, close approach, compact gas in shell (element) coolers for IGCs. *This represents the vast majority of the coolers purchased with IGCs.* A typical design is shown in Figure 13.
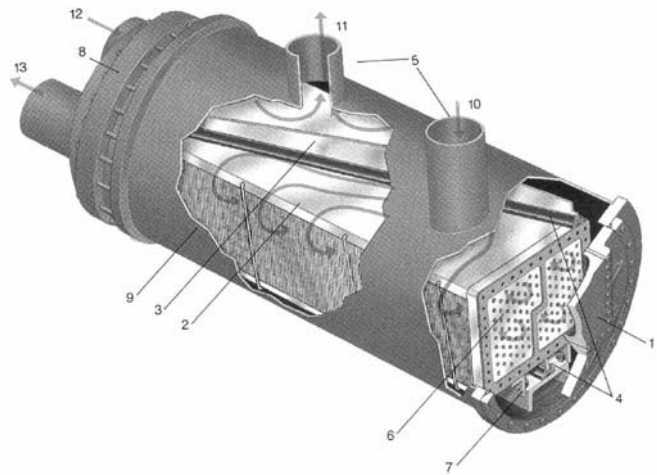

Figure 13. Typical Cooler Design.

Gas-in-tube coolers are generally used only where the operating pressure would result in excessive shell thickness and high material cost. Gas-in-tube coolers are occasionally used for lower pressure aftercooler duties where gas side pressure drop is less critical. *Present design limits for gas on the shell are about 800 psig (55 barg) with tube materials normally used in the industrial gas industry.* However, there is a commercial impact. The gas-in-shell coolers do have a maintenance plus due to the inherent cleaning aspects (tube rodding).

Element type coolers are built with shell diameters between 2.5 ft (800 mm) and 12 ft (3500 mm) and with tube lengths up to 33 ft (10 m). Gas flows in a single pass perpendicular to the tube bundle. External tube fins are always used to enhance the gas side surface area and compensate for the poor gas thermal conductivity. Cooling water is arranged in two or four waterside passes depending on the required cooling water pressure drop and temperature rise. Tube geometry is normally triangular pitch.

Coolers are short relative to their diameter, and water channels are arranged for each intercooler at one side of the IGC package. Gas nozzles are oriented to suit the stage volute positions, minimizing or even eliminating interstage gas piping.

To minimize piping stress and nozzle loads, *coolers may be spring mounted* either from above or below. If the coolers *are rigidly mounted, the interstage piping may require flexible elements* to meet maximum nozzle load criteria. *Suppliers have employed both methods successfully*. Machines that have the flexible elements in the piping have not caused many unnecessary shutdowns.

Approach temperatures of 11°F (6°C) are common. However an approach temperature of 9°F (5°C) can be achieved and may be economically justified. Rarely are the approaches less than this 9°F. Where power consumption is less critical, approach temperatures may be allowed to rise. In order to achieve low approach temperatures, it is essential that little or no gas is allowed to bypass the cooler bundle from the hot to the cold side. Multilayer stainless steel sealing strips have been developed to provide a good seal along the length of the tube bundle.

In atmospheric air or wet gas application, separation of condensate is critical to the efficient operation of the IGC. Condensate carryover in air compressors will increase mass flow and stage power. Impellers become more susceptible to dirt deposition, reducing efficiency and giving rise to rotor unbalance. Condensate may also result in corrosion of the stage diffuser walls and reduction in pressure rise to surge.

Condensate is normally removed using stainless steel mesh pads. These demister pads may be located either horizontally in a dedicated area of the cooler shell or vertically alongside the tube bundle; both are acceptable. In both cases control of the gas velocity and flow profile is essential for good separation. Vertical demister pads should be fitted with a gutter system to drain away condensate at intermediate heights. Vertical demisters result in a considerably shorter cooler shell length and tend to achieve a more even flow distribution. One drawback to the vertical demisters is that the gas/particle velocity through the mesh pad is relatively low. Aftercoolers, where the downstream process may require better liquid separation, must be particularly carefully designed.

Cooler gas side drains must be sized to handle the volume of condensate produced under all operating conditions. To minimize gas losses, drains are usually fitted with condensate traps. Condensate traps should be fitted with bypass valves and strainers. To minimize reentrainment of condensate, cooler gas side drains should ideally be located close to the end of the cooler toward which the air is flowing. Level alarms are rarely used in the industry. Options to traps have included continuously bleeding drilled gate valves and solenoid valve actuation. In all cases, strainers need to be installed to prevent blockage.

Tube and fin material must be carefully chosen to suit the cooling water and gas side fluid composition. A wide selection of fin and tube materials is available. Tubes may be 90/10 CuNi, 70/30 CuNi, admiralty brass, 304 SS, 316 SS, duplex or titanium. *The vast majority of the coolers used by the industrial gases companies use a copper-based material. Present day "standard" designs in air/nitrogen service use 90/10 Cu/Ni*. Fins may be plate type, helical high fin, or bimetallic. Fins may be aluminum, copper, or stainless steel, although *most designs use aluminum or copper*. In recent years, cooler fins made of copper (in wet air service) have been found to have some environmental issues in certain countries due to copper chromate in the condensate. The plate fin design uses a continuous plate, usually aluminum, interconnecting all tubes and providing a bundle that is fully resistant to vibration. Tubes are expanded into the plate fin to produce a tight mechanical contact at the fin to tube interface. Helical high fin, usually copper, is wrapped around the tube. The fins may be soldered to the tube to provide a bonded joint and to increase the corrosion resistance of the fin. Nonmetallic hexagonal spacers are fitted at intervals along the tube length, and the bundle is clamped to ensure a tight fit of the tubes and to avoid any tube vibration. *For corrosive services, a phenolic coating is sometimes applied to the cooler bundle*. This decreases the heat transfer slightly, but will improve corrosion issues.

Cooler shells for atmospheric air duty and dry gas compression are carbon steel. To prevent corrosion in wet air duties internal coatings are usually applied to the shells.

Coolers can be designed to any recognized pressure vessel design code. However some codes require a thicker tube wall than others. Tube wall thickness may become too great to allow expansion of the tubes for a plate fin design, and then a helical high fin design must be used. *Tubular Exchanger Manufacturers Association (TEMA) is not used as the design basis because it is not readily applicable to the gas in shell, crossflow coolers. However manufacturers say they meet the intent of TEMA, and can provide standard exceptions to TEMA (normally TEMA C)*.

Lubrication Systems

Compressor vendors have developed simple low cost lube oil systems that meet the reliability requirements of the industrial gas industry. *API lube oil systems are not the norm and are only provided when specifically requested by a customer*.

The lube oil reservoir may be incorporated into the compressor baseframe or may be freestanding. Some designs incorporate the oil reservoir as a structural part of the compressor support frame. *Retention time is typically five minutes, although some smaller units are three minutes*. Oil pump, filters, cooler, etc., will be mounted on the top of the oil reservoir. *Single water-cooled oil coolers are standard with either single or duplex oil filters*.

The main lube oil pump will typically be mounted on the compressor gearbox and *either driven from an auxiliary gear off the bullgear shaft or directly driven off the bullgear shaft. This has always been the standard in the industrial gas industry*. The gear driven main oil pump will be sized to provide rundown lubrication in the event of a plant electrical power failure. A system of check valves may be installed around the main oil pump to provide positive lubrication in the event of compressor reverse rotation that may occur if, after a trip, the compressor discharge system was to depressurize through the compressor. An electric motor driven auxiliary lube oil pump will be used for prelubrication prior to starting and to provide rundown lubrication in the event of a failure of the main lube oil pump. Sparing of the main oil pump is recommended, as often this pump is a special design due to the gearbox mounting method. In almost all cases, an overhead rundown tank is not provided if there is a gear driven main oil pump.

An oil vapor extraction fan draws a very slight vacuum on the gear casing. This minimizes oil leakage through the shaft labyrinth oil seals and reduces the possibility of oil contamination reaching the process. An acceptable alternative to an extraction fan is to use an air ejector. This is more of a commercial issue as the cost of the motive gas may be higher than the electrical cost of the fan. In either case, the compressor can operate without the oil extraction system, but oil seal leakages will be noticed.

All components in the lubrication system, except for piping between the oil filter and the compressor bearings, are carbon steel as a standard. The industry has not found that the stainless steel reservoir/components are needed, as the reservoir is always hot and purged by the air extraction system.

Couplings

A coupling is required between the electric drive motor and the compressor bullgear. Motor running speed will be between 3600 rpm and 900 rpm. Considerable coupling length may be required to give clearance for volutes and piping mounted on the motor side of the gear case. *The drive for reduced maintenance has led to the replacement of grease lubricated gear type couplings by dry multidisc couplings (although some designs still use the gear type coupling)*. The use of high power synchronous motors with their inherent high pulsating torque during startup must be considered when designing the coupling. A synchronous motor may use a reduced voltage starting method. The coupling stiffness can be adjusted to modify the train torsional natural frequency such that

the train passes through the torsional critical before switching to full voltage. The coupling must be designed to withstand the expected number of compressor starts.

Where a steam turbine is used to drive an IGC through a pinion, the issues associated with coupling design are different. The coupling will be operating at a relatively high speed. Due to the relatively light pinion weight, the coupling may have a significant effect on rotordynamics. The coupling mass is effectively overhung on the pinion. The coupling stiffness may also significantly affect the rotor behavior. *In these instances, a train lateral analysis should be considered.*

Instrumentation

Instrumentation can be divided into three groups: instrumentation required for control, instrumentation required for protection, and instrumentation required for performance monitoring. A balance has to be sought between cost, reliability, and risk of serious mechanical damage. Larger IGCs can be fitted with a full scope of instruments including X and Y vibration, axial position, bearing temperature, interstage pressure, and temperature. Smaller, more standardized IGCs may be limited in the available choice of instrumentation, with additional instrumentation being relatively costly. Industrial gas companies have different philosophies on the scope and function of instrumentation. Many air separation plants can be shut down for short periods when customers are provided with product from cryogenic storage systems, and this may influence decisions on instrumentation.

Surge Control

Although IGCs are robust machines and although some amount of surging can be tolerated without damaging the compressor, most systems will include antisurge control. With IGCs, a flow signal can conveniently be taken by measuring the pressure difference across the reducer at a stage inlet. This may be the first stage but may also be a later stage, where a higher pressure differential will be obtained. Experience shows this flow signal is sufficiently accurate for antisurge control. A signal for discharge pressure or pressure ratio can by taken from pressure measurements in the process piping. Other experience shows that a pressure ratio and a power signal are adequate. Experience has shown that the plant digital control system (DCS) is adequate for antisurge control. Antisurge control may also be carried out locally with a programmable logic controller (plc) based system. Some standardized IGCs use alternative parameters for antisurge control such as stage inlet section pressure difference and stage pressure difference. *Basically, in the industrial gas industry, a simplified surge control has been found to be acceptable.* Part of this reason is that for this business the fluid molecular weight does not change, the compressor speed is constant, and often the suction pressure is relatively constant.

Except for oxygen compression, *it is not usual to fit instrumentation for surge detection.* If required, surge may be detected by reverse pressure difference across a stage inlet reducer, temperature increase at a stage inlet, rate of change of stage discharge pressure, or, more crudely, through motor current or axial vibration.

Vibration Systems

Vibration systems use the proximity type probes found in most turbocompressors. The industrial gas industry will most *often use a single probe at each high speed bearing location, and no axial probes.* As the compressor criticality increases, orthogonal probes will be added, as well as some axial probes and bearing resistance temperature detectors (RTDs). *Accelerometers are used on the gearcase only when needed for predictive maintenance reasons.* Vibration probes are almost always wired directly to the DCS (computer system). This has been an ongoing discussion with the various suppliers as well as the proximity probe suppliers. *DCS scan/response times are less than 0.5 sec range.*

Inlet Guide Vanes

Inlet guide vanes may be fitted to one or more stages of an IGC. High flow *coefficient three-dimensional impellers with axial inlet flow respond well to control by inlet guide vanes. Multiple sets of inlet guide vanes are used to improve turndown efficiency on MACs and product compressors. On multiservice IGCs, inlet guide vanes may be fitted at the first stage of each process section.* Figure 14 shows a typical inlet guide vane. To minimize load torque during start up, to around 25 percent of full load torque, the inlet guide vanes can be almost completely shut leaving only a small central orifice through which flow can pass. The compressor should only be operated in this condition for a minimum period, as the turbulent flow from the guide vanes may excite blade vibration in open impellers leading to blade failure. The guide vanes are cantilevered from the pipe wall. The location of the inlet guide vanes in the stage suction piping allows the use of a fairly simple mechanical guide vane linkage that may be external. However, the guide vane mechanisms must be designed to accept the loads imposed by the guide vane actuator. Guide vane spindles are sealed to prevent leakage of gas. In the case of elevated suction pressures, this leakage is to atmosphere. However in the case of low suction pressure product compressors, air may leak into the system leading to product contamination. For this application guide vane mechanisms must either be internal to the compressor casing or, if external, must be purged with the product gas. To achieve maximum effect the inlet guide vanes should be located close to the impeller and downstream of any inlet reducer. Guide vanes may be subject to damaging vibration if located in an area of turbulent flow. Such areas of turbulence may exist immediately downstream of a swept interstage piping elbow.

Figure 14. Inlet Guide Vane Assembly.

PACKAGING AND INSTALLATION ISSUES

In the past small IGVs were highly packaged while large units were installed on mezzanine style foundations and required a large amount of field installation effort. *Today even very large units (examples up to 20,000 hp) are installed on steel that is anchored to a flat concrete foundation without experiencing any operating problems. More controversial is the limiting size for installation of the motor on steel, particularly when a two-pole drive motor is involved.* This is a consequence of vibration problems, which have been experienced due to inadequate stiffness in the support structure under the motor and unanticipated structural resonant conditions. The situation is aggravated by the fact that many motor frames themselves have been of poor structural design. Although the design tools exist today that should permit extending the limits substantially above current limits, the cost of correcting problems when they occur has prompted caution in this area. *A typical recommendation is to limit steel mounting motors at 3000 hp (2250 kW) for mounting two-pole motors on steel and about 5000 hp (3750 kW) for four-pole and higher motors.*

The interstage piping on IGCs is very compact and often insufficient space exists to accommodate flanged connections. This has lead to the widespread use of grooved pipe joint couplings.

Generally these couplings have been trouble-free and reduce the effort involved in installation and removal. They do however have pressure thrust limitations that may require the use of tie-rods across them. There is also a common misconception that these couplings can act as expansion joints. While they can accommodate more piping misalignment than flanged connections, once they have been made up they are essentially rigid and transmit piping loads accordingly.

When installing the piping, regardless of the connection type involved, it is very important to avoid putting excessive loads into the gearbox because this adversely affects gear tooth contact. *Volute deflection should be monitored with dial indicators during the installation. The coolers should be full of water on their waterside. Careful attention should be paid to the piping flexibility on external gas piping as well for the same reason.*

If the unit is provided with soleplates they should be leveled and grouted in place using a good quality epoxy grout. Smaller skidded compressor systems are normally grouted with a sand cement grout. The equipment installed on the soleplates should be shimmed at the hold-down bolts as required to achieve good alignment and should be checked for "soft foot" conditions. Because larger IGCs often have long spacer couplings, alignment should be done using lasers as opposed to the reverse indicator method with fixtures. If a fixture is used, a calculation of the fixture sag is required. It should be noted also that because of the relatively long coupling spool lengths and relatively low drive speeds, the allowable misalignment is relatively high. When the alignment has been determined to be satisfactory, the equipment should be doweled to the sole plates. If the unit is skid mounted with the drive motor separately mounted on concrete, the compressor skid should be set, leveled, and grouted and the final alignment achieved by moving the motor. *The supplier should always dowel the gearbox to the skid steel while the machine is in the shop.* When setting and leveling the skid it should have temporary shims at the anchor bolts, and the anchor bolts should be tightened and the skid checked to be sure that these shims are supporting the skid evenly before it is grouted. After the grout has cured the shims should be removed. A grout should be put in the resulting cavities. The procedure is the same for setting and grouting the skid when the motor is mounted on the skid. If the compressor supplier, prior to arriving in the field, mounted the motor on the skid, it should have already been aligned. This alignment should be rechecked to verify that it was not changed in the course of installing the skid.

COMMISSIONING

The commissioning activities involved for IGCs are, for the most part, the same as they would be for other turbocompressor installations. Several points are worth considering. Oil flushing is not generally required on fully packaged units that have been run in the factory prior to shipment. It is important to maintain the condition of the compressor from the time it is received in the field until it is put into service. In the case of completely or partially packaged unit it is easy to forget that much or all the shipping preservation may be lost during installation, and deterioration can take place prior to startup. Maintaining a nitrogen atmosphere on the gearbox oil system and compressor seals is a good way of preventing problems. The vibration probes on many IGCs are located in the shaft seals. If the unit has seal gas supply or recovery piping it should be checked for cleanliness. The presence of rust or scale

in the piping can adversely affect the performance of the vibration probes and damage the probe target surface. Prior to running the compressor, machinist bluing should be applied to the pinion teeth and to several segments of the bullgear teeth and the tooth contact verified after a loaded run of the compressor. The verification should be done after only a few hours of loaded operation. If significant edge loading of the gearing exists an excessive number of stress cycles can be accumulated in a short period of time.

OPERATIONS AND MAINTENANCE

There really are very few aspects of predictive and preventive maintenance of IGCs that are fundamentally different from other types of turbomachinery. Good practices in the areas of vibration analysis and lube oil analysis are really no different. Because of the integrated relationship between the compressor and the gearing it is more important to monitor the health of gearing and the pinion bearings than might be done on other compressor systems. Checking tooth contact every few years is a good practice. When doing so, the best practice is to apply new bluing and check the contact after a loaded run in a similar fashion as was done during commissioning. Also, the emergence of significant vibration amplitudes at multiples of pinion speed can be evidence of a wear pattern developing in the gearing, which can result in fretting on the backs of the bearing pads that in turn causes the bearing clearance to increase and tooth contact to deteriorate. When this occurs the gearing can often be recovered by touch up grinding by the gearing supplier provided the wear is not excessive.

There have been instances where field balancing of a pinion was needed to bring down vibration levels. It is often more expedient and less costly to install the spare pinion (more easily done than an inline compressor rotor) and then send the removed pinion for an inspection and shop balance.

CONCLUSION

With due consideration of several design parameters, the integrally geared compressor has been found to be a well established and reliable compressor design for the industrial gas industry. A solid design, a good installation, and adequate instrumentation allow this style of compressor to run successfully for many years. The manufacturers have improved their technology and increased the size and complexity of the machines, without sacrificing reliability.

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