HERMETICALLY SEALED OIL-FREE TURBOCOMPRESSOR TECHNOLOGY

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ABSTRACT
The reciprocating compressor and the traditional centrifugal compressor have been the workhorses of the natural gas industry. However, new demands placed on gas distribution infrastructure by spot market activity necessitates the development of a more flexible and reliable unit that can efficiently and quickly respond to fluctuating market requirements.

This paper presents a hermetically sealed, oil-free turbocompressor system that was developed to fulfill the current natural gas compression demands of flexibility and reliability.

The original targeted market for this system was the gas storage industry. However, the resulting design provides an excellent platform for compression applications that are constrained by low emission requirements, available space, location, or health and safety concerns.

INTRODUCTION
The production, distribution, and underground storage of natural gas in North America and Europe requires reliable gas compression. As the marketing and utilization of natural gas has evolved over the past 20 years, so have the demands on the compression equipment. From the wellhead to the end user, these changes
demand economically superior solutions with greater reliability and flexibility in order to provide a cost competitive commodity.

Traditionally, natural gas consumption has been seen as a seasonal occurrence with the greatest demand for residential and commercial heating during the winter. Therefore, the infrastructure of the distribution systems has included storage of natural gas directed toward accumulating a sufficient supply to meet these seasonal base load requirements. Additionally, a smaller segment of the market provided high deliverability storage to meet peak load demands.

Over the past 20 years more stringent environmental regulations have driven many large users of energy to select natural gas as the fuel of choice, particularly for new power plants. The increased use of energy efficient gas turbines for base load and peaking services has caused the demand for natural gas to rise significantly. Additionally many users with fuel switching capability have moved to natural gas because of the reduced emissions profile. In addition to the increased demand, the usage patterns have changed to include a summer demand peak in many areas. Hence, today’s natural gas market demands an adequate supply of competitively priced natural gas on a year round basis.

Deregulation of the natural gas markets in the United States and liberalization in Europe has also facilitated this transition to increased gas consumption and has allowed the creation of a lightly regulated commodity market for natural gas. This market encourages participating users to manage their cost of gas by the use of flexible storage capacity. Consequently, gas transmission and storage facilities have to adapt and react to these market demands. This means that instead of the traditional yearly storage cycle, storage facilities need to provide high performance, multi-cycle per year capability with fast turnaround times.

One of the most critical components required to meet these new requirements for natural gas distribution is high performance gas compressors. From the facility operator’s perspective these compressors must have the following attributes:

- **Reliability**—Minimal unscheduled downtime
- **Availability**—Infrequent scheduled maintenance required
- **Flexibility**—Wide operating range (flow rates and pressures)
- **Responsiveness**—Able to react quickly to changing conditions
- **Competitiveness**—Low life-cycle cost

A new turbocompressor system has been developed to meet the demands of the natural gas industry. This paper includes an indepth discussion of this new turbocompressor system’s heritage, system design, operational functionality, and static and dynamic characteristics. Operating experiences both during factory testing and onsite are presented. Finally, a discussion of how this new turbocompressor system’s compact, hermetically sealed, and all-electric attributes can provide enhanced performance in various other applications is presented.

**HERMETICALLY SEALED OIL-FREE TURBCOMPRESSOR TECHNOLOGY**

**Background and Heritage**

The realization of a hermetically sealed oil-free turbocompressor that fulfills the operational flexibility, reliability, availability, and environmental requirements of a modern gas distribution system comes from the experience gained in more than 12 years of operation and more than 30 installations of oil-free compressors for pipeline, gas storage, gas injection, and gas depletion applications. Examples of how such equipment addresses these requirements and how that has led to the evolution of a hermetically sealed oil-free turbocompressor are discussed below.

**Flexibility**

An operationally flexible compressor must be capable of a wide operating range. This can be accomplished by a large operating speed range and a compressor with multiple stages that can deliver sufficient pressure ratio. However, a large operating speed range can only be obtained if the rotor system’s rotordynamic margins to critical speeds, such as those stipulated by API 617, Seventh Edition (2002), are respected. This is possible if there are no critical speeds within the operating speed range or if the critical speeds within the operating speed range can be critically damped. Due to the high speeds and long length of multistage compressor rotors it is inevitable that a critical speed will be present below the maximum continuous operating speed. The presence of this critical speed prevents, in the case of conventional oil-film bearings, continuous operation at or near this critical speed. This is not the case for a rotor supported by active magnetic bearings. Active magnetic bearings can critically damp critical speeds and provide the freedom of continuous operation at any speed. Figure 1 shows a gas storage compressor system where the compressor is directly driven by a high speed motor with both motor and compressor rotors supported by magnetic bearings. A description of the high speed motor technology is provided by Gilon and Boutriau (1998). Since both the high speed motor and compressor are supported by active magnetic bearings, the operating speed range for this system is typically 30 to 105 percent. If the process duties require compression through two compressors the operating range can be further enhanced by carefully matching the two compressors’ flow and pressure ratio design so that they can operate in series and parallel mode. The advantages of series and parallel mode are illustrated on the example compressor performance map shown in Figure 2. This series/parallel operation capability of a compressor system is advantageous for gas storage requirements because the pressure ratio can range from nearly one and upwards. For low pressure ratios, i.e., when the storage pressure is close to pipeline pressure, operating in parallel mode offers both increased flow and efficiency and realizes optimal gas deliverability for the storage facility. For higher pressure ratios operating in series mode effectively doubles the pressure ratio of the system.

**Reliability**

The reliability of a compression system can be greatly improved if the system comprises fewer components. This follows the sound equipment operator philosophy: “the less there is, the less there is to go wrong!” Therefore, heeding this sound philosophy, it is best to keep only what is essentially necessary. A system that employs this only-what-is-necessary philosophy is shown in Figure 3. This compression system is dedicated to pipeline boosting duties and consists of simply a high speed motor, supported via active magnetic bearings, with two compression stages overhung off each side of the motor. The use of magnetic bearings eliminates wear due to contact between rotor and stator and the need for a lube-oil system. Unique to this compression system’s design is the cooling of the motor and magnetic bearings with the process gas. The cooling system is very simple and: (a) extracts gas after the first stage, (b) passes this gas through a filter, (c) feeds the filtered gas...
Figure 2. Series and Parallel Operation Mode Compressor Characteristics.

to the motor and bearings, and (d) collects the heated gas and returns it back to the suction pipe. This very simple cooling system using process gas is only possible because the motor and active magnetic bearings require nothing more than electrical supply currents and a cooling medium to remove heat. Since the process gas is pressurized natural gas, the cooling effectiveness of this gas is particularly high due to its high density and specific heat. Because the motor and bearings are immersed in process gas the combined compressor and motor casing is hermetically sealed. This eliminates the need for shaft seals to atmosphere and their associated conditioning, monitoring, and buffer systems. The elimination of the need for shaft seals, a seal gas system, and a lube-oil system removes system components that have the potential to fail and, thereby, enhances the system reliability.

Figure 3. Hermetically Sealed Pipeline Turbocompressor System.

Availability

The ability of a system to react spontaneously to changing market needs demands a compression system that has high availability. This requires that the compressor system must not only be reliable, but maintain this reliability over long periods of time, i.e., downtime for routine maintenance must be minimized. An all electric simplified system as described above for the hermetically sealed pipeline compressor offers this ability. Here, again, a minimization of system components results in improved system availability. Evidence of this is provided by experience gained from a pipeline boosting facility in the United Kingdom with three hermetically sealed pipeline compressors as described above and illustrated in Figure 3. A photograph of the facility is shown in Figure 4. For this facility three identical machines rated at 8 MW (10,728 hp) and 10,000 rpm serve the purpose of peaking gas boosting or pipeline pressure “packing.” Their typical operation profile has been to perform more than one start per day, accelerate to maximum speed, operate at maximum speed and power for a limited time (typically an hour or less), then shut down. For the first four years of operation these units have been able to accumulate more than 7000 starts and 9500 hours of operation. In spite of this demanding multiple start operating profile the complete compressor station availability has been greater than 94 percent. The pipeline pressure “packing” duty of these machines has much in common with gas storage because the “packing” of the gas is effectively storage of gas in the pipelines. This positive experience demonstrates that the hermetically sealed pipeline compressor technology can fulfill the operating availability demanded by a modern gas distribution system.

Figure 4. Hermetically Sealed Pipeline Turbocompressors in a Pipeline Boosting Compressor Facility.

Environmental

An increasingly important requirement for any gas compression facility is a minimization of all aspects that can pollute the environment. This not only applies to emissions of pollutive substances such as liquids and gases, but also noise. This has led to an increased implementation of electric motors as drivers in instances where the environmental aspects are particularly stringent. An electric motor drive eliminates local exhaust emissions from a combusted fuel source and greatly reduces the noise generated by the equipment. Furthermore, when the motor current is fed via a variable frequency drive (VSD) improved partial load efficiency and compressor operation near optimum efficiency is available. Implementation of magnetic bearings brings the environmental benefit of absolutely no lube-oil system and the associated environmental problems of oil spills and used oil disposal. Finally, a hermetically sealed compressor design and its absence of shaft seals assures that there are no continuous local emissions of process gas via a flare or extra requirements to boost the shaft seal gas back to the suction that can adversely affect system reliability and availability.

The Hermetically Sealed Oil-Free Turbocompressor

The above discussion introduced two compressor system types and their associated components that can fulfill modern gas distribution demands. This has led to the exercise to select the best components and aspects of each system, which when combined, would best serve these demands. To this end Table 1 lists the aspects of various components from these systems and labels them as either positive (+), negative (−), or neutral (0) in their ability to fulfill the particular demands associated with modern gas distribution.
Table 1. Comparison of Existing Components and Aspects Best Suited for Modern Gas Storage Demands.

<table>
<thead>
<tr>
<th>Component or Aspect</th>
<th>A - Hermetic Sealed Pipeline Compressor</th>
<th>B - Multistage Oil-Free Compressor</th>
<th>Best for Modern Gas Storage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor</td>
<td>Integrated</td>
<td>Separated, coupled [-]</td>
<td>A</td>
</tr>
<tr>
<td>Magnetic Bearings</td>
<td>In process gas [-]</td>
<td>Neutral [0]</td>
<td>A</td>
</tr>
<tr>
<td>Shaft Seal System</td>
<td>None required [-]</td>
<td>Required [-]</td>
<td>A</td>
</tr>
<tr>
<td>Number of Stages</td>
<td>Maximum of two [-]</td>
<td>Multiple, more than two [+]</td>
<td>B</td>
</tr>
<tr>
<td>Motor/Bearings</td>
<td>Process gas [-]</td>
<td>Externally Forced Cooled [-]</td>
<td>A</td>
</tr>
<tr>
<td>Cooling Medium</td>
<td>Medium, up to 150 bar [-]</td>
<td>High, more than 150 bar [+]</td>
<td>B</td>
</tr>
<tr>
<td>Coupling</td>
<td>None required [-]</td>
<td>Required [-]</td>
<td>A</td>
</tr>
<tr>
<td>Casing Temperature</td>
<td>Low to moderate design [-]</td>
<td>Moderate to high [+]</td>
<td>B</td>
</tr>
<tr>
<td>Level</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pressure Ratio</td>
<td>Low [-]</td>
<td>High [+]</td>
<td>B</td>
</tr>
<tr>
<td>Flexibility</td>
<td>Good, series and parallel operation and wide speed range [+</td>
<td>Good, series and parallel operation if two compression stages, high pressure ratio and wide speed range [+]</td>
<td>A+B</td>
</tr>
<tr>
<td>Reliability</td>
<td>High, very few components [+</td>
<td>Moderate, more components [0]</td>
<td>A</td>
</tr>
<tr>
<td>Availability</td>
<td>High, simplified system with few components [+</td>
<td>Moderate, more complicated system with more components [0]</td>
<td>A</td>
</tr>
<tr>
<td>Environmental</td>
<td>Excellent, no local emissions, low noise [+</td>
<td>Moderate, local emissions via flare, higher noise level [0]</td>
<td>A</td>
</tr>
</tbody>
</table>

The result of this selection is a compressor system that integrates the best suited components from the hermetic pipeline and the multistage oil-free compressors and is shown in Figure 5. This new compressor system has most commonality with the hermetic pipeline compressor and integrates the multiple stages and high pressure casing from a multiple stage oil-free compressor system. Hence, the new compressor system is a direct connection of the multistage compressor rotor and casing with the motor rotor and casing. This turbocompressor system can be further enhanced by adding an additional compressor rotor and casing on the opposite side of the motor yielding a tandem arrangement (refer to Figure 6). With a tandem arrangement higher pressure ratios can be achieved as well as the enhanced compressor flexibility made possible with series and parallel operation. What is distinctly different from the previous systems is the presence of a single radial magnetic bearing in the compressor casing and a larger axial magnetic bearing located between the motor and compressor. Both these differences have distinct reasons and are described below.

The use of only a single magnetic bearing in the compressor section in a single compressor configuration is possible because the adjacent motor bearing serves to help support the combined motor and compressor rotor. This configuration yields a combined rotor system that is supported by three radial bearings. The primary benefit is the absence of a fourth radial bearing, which yields a multistage turbocompressor system that is very compact. If the casings were coupled together via a coupling yet still hermetically sealed, the resulting footprint of the turbocompressor system would be unnecessarily large. Because the fourth radial bearing is not required there is space available in the area of the interface between motor and compressor rotors. Hence, this is the logical location to place the axial bearing for further space optimization along with other benefits.

The benefits of locating the axial bearing between the motor and compressor rotors are the following:

- The thrust disk can be manufactured from a single forged disk that is not shrunk onto the shaft. With a solid disk higher tip speeds can be tolerated allowing a larger thrust disk to be implemented with the benefit of increased thrust capacity for high pressure applications.
- The location minimizes the negative effect upon the rotordynamic characteristics by avoiding having a heavy overhung weight and a longer than necessary bearing span. This location therefore provides the stiffest rotor and best rotordynamic performance.
- The location near the low pressure compressor realizes the lowest possible pressure (slightly above suction pressure) in the axial bearing cavity. The lowest pressure possible is desirable for minimizing windage losses and, consequently, increased system efficiency. Furthermore, the location also facilitates the protection of the axial bearing by allowing it to be exposed only to filtered and cleaned process gas as is also the case for the adjacent motor.
- In the case of the tandem arrangement of two compressors, this location also minimizes the effect of thermal expansion that has to be taken into account in the sensor target and radial bearing laminarization stack length on the opposite side of the motor. A centralized axial bearing thus allows optimized control or differential shaft thermal expansion.

Figure 5. Multistage Hermetically Sealed Turbocompressor System.

Figure 6 Tandem Multistage Hermetically Sealed Turbocompressor System.
The fundamental components of the new hermetically sealed oil-free turbocompressor system are not new because it is an integration of already existing and proven technologies. However, the integration of these components has created some aspects that are new and these aspects required some detailed analytical inspection. These new aspects concern the static (bearing capacity and misalignment influences) and dynamic (lateral and torsional rotordynamics) characteristics due to the multiple (more than two) bearing design of the combined motor and compressor(s) rotor and are the topic of discussion in the next main section.

**STATIC AND DYNAMIC ANALYSES**

*Rotor Model and Static Design*

The compressor design has directly coupled rotors with either three (single) or four (tandem) radial bearings and this makes the rotor statically indeterminate. Moreover, static forces, or bearing loads, can arise from gravitational, misalignment, and rotor/stator interaction gas influences. This, coupled with the fact that active magnetic bearings have limited capacity, means that careful consideration of static forces is required. Therefore, a finite element based static analysis is performed to ensure that the rotor model (shaft and bearings) is adequately designed and the bearings have sufficient capacity to cover the expected static loads while having sufficient remaining capacity to tolerate the dynamic forces. The dynamic forces are primarily due to rotor imbalance and can also include gas forces from seals and acoustic pulsations, e.g., rotating stall or surge. A minimum rule-of-thumb is to reserve 50 percent of total bearing capacity for dynamic forces.

Figure 7 shows a rotordynamic model for a single (three bearing) configuration. This rotor model has six stages, whose relative center of gravity locations are indicated by the solid dots, modeled via concentrated masses and inertias. The calculated gravitational bearing loads and rotor deflection shape are shown in Figure 8, and the calculated bearing loads and rotor deflection shape when the compressor bearing stator is misaligned by one mm (39.4 mils) are shown in Figure 9. The two calculations in Figure 8 and Figure 9 enable a misalignment load diagram to be drawn (refer to Figure 10), which depicts the bearing loads as a function of compressor bearing stator misalignment with the respective bearing 50 percent capacity limit superimposed. Figure 10 portrays the extent of misalignment that is possible before exceeding the minimum rule-of-thumb 50 percent bearing load capacity limit. It is interesting to note that for optimum misalignment tolerance the compressor bearing should be laterally offset approximately 0.2 mm (7.9 mils) in the vertical (negative) direction, i.e., in the center of the misalignment tolerance range. The tolerance range of approximately ±0.4 mm (16 mils) is well within the better than 0.05 mm (two mils) lateral alignment capability achieved in practice. It is important to note that a very stiff rotor, which is typically beneficial for rotordynamic considerations, will have negative impact on misalignment tolerance, i.e., a small misalignment of a stiff rotor will cause large bearing loads resulting in a narrow tolerable misalignment range.

Since the above discussion determined that the rotor design fulfills the static requirements, what follows is verification that the positive static characteristics do not have an adverse effect on the rotordynamic characteristics.

**Figures 7-9:**

- **Figure 7 Multistage Hermetically Sealed Turbocompressor Rotor Model.**
- **Figure 8. Bearing Loads and Rotor Catenary Shape under Gravitation Load.**
- **Figure 9. Bearing Loads and Rotor Catenary Shape for Compressor Bearing Misalignment.**

**Rotordynamic Design**

**Lateral**

*Undamped Eigenvalue Analysis.* The undamped eigenvalue analysis serves the purpose to identify the critical speed locations and mode shapes as a function of bearing stiffness and hence to map the rotor to bearing stiffness relationship. Figure 11 shows the lateral critical speed map for the rotor model in Figure 7. Superimposed upon the lateral critical speed map is a typical effective stiffness path for magnetic bearings. Below the maximum speed this stiffness path intersects the first three modes at approximately 2400, 3500, and 7000 rpm, respectively. These speeds are the expected critical speed locations. Figure 12 shows the associated mode shapes of the curves at the approximate effective bearing stiffness (30 million N/m, 170,000 lb/in). Classical two bearing rotors with low bearing stiffness have first two rigid body
modes (cylindrical and conical) and, thereafter, the first bending mode. Due to the nonclassical mass-elastic nature of the combined three bearing motor and compressor rotor in Figure 7 there is a “tilting” mode (Mode 3) between the rigid body modes (Mode 1 and 2) and the first bending mode (Mode 4). The tilting mode is effectively a rigid body mode because it has much movement at the bearing locations and functions like a combined motor and compressor rotor conical mode. Note that the tilting mode (Mode 3) is the only critical speed within the operating speed range and therefore must be critically damped.

A further purpose of the undamped critical speed analysis is to determine if sufficient margin is present between maximum continuous speed and the first bending mode. Due to the relatively low movement at the bearing locations and high speed for the first bending mode large damping is necessary to critically damp this mode should operation at or through this mode be desired. Since magnetic bearings are dynamically soft, the margin between first bending mode and maximum operating speed is based on the effectively low bearing stiffness of one million N/m (5710 lb/in). A margin greater than 15 percent is required to avoid increased vibration due to operation near a critical speed.

Damping Parametric Study. A damping parametric study is conducted to determine the influence of damping upon the eigenfrequency and damping ratio of a particular mode. The particular mode of interest here is the tilting mode (Mode 3) because it is in the operating speed range. Therefore, a damping parametric study applied to this mode identifies whether it is possible to critically damp this mode and, if so, how much damping is required. The rigid body modes (Mode 1 and 2 in Figure 12) are not of interest because they are very easily critically damped and more so, due to the low bearing stiffness, their eigenfrequencies are outside the operating speed range. Figure 13 shows the results of a damping parametric study for the tilting mode. For this study the stiffness for all bearings was held constant at one million N/m (5710 lb/in) and the damping coefficient for all bearings was parametrically increased from 0 to 100,000 Ns/m (0 to 571 lb-sec/in) with the knowledge that an advanced algorithm (discussed in the following section) can provide these conditions. The results show that, as expected, very little damping is required to yield damping ratios above the API critical level (0.2, which yields an amplification factor of 2.5). With damping ratios above 0.2 the mode is considered critically damped and no separation margin is required. This enables operation at the critical speed and realizes the wide operating speed range required for gas storage applications. The damping parametric study also provides valuable information to determine optimum settings for advanced algorithms to optimize rotordynamic response.

Advanced Algorithms. Modern active magnetic bearings have the distinct advantage that they have digital controls and that these controls can be readily adapted to optimize the rotordynamic characteristics of a particular machine thus enhancing machine reliability and availability. The particular characteristics of the machine in question here (rotor model shown in Figure 7) are the need to critically damp the tilting mode, identified above, guided by the mode’s sensitivity to damping detailed in the previous section. The forces necessary to critically damp the tilting mode (Mode 3) can then be provided via a dedicated transfer function that manages the gain and phase of the actuator to fulfill this requirement. However, since the control is digital based, the opportunity is available to further enhance the rotordynamic characteristics via advanced algorithms. The advanced algorithms
can enhance the rotordynamics in the following two ways: (a) apply a pure damping force at the synchronous frequency to minimize or eliminate rotor response to unbalance while traversing a critical speed, and (b) reduce the controller response to the synchronous frequency at speeds away from critical speeds to minimize the transmitted unbalance reaction force between the rotor and the stator. For the later advanced algorithm the controller allows the rotor to precess about its inertial axes much like the behavior of a satellite in free space. In this scenario there is a minimum stiffness path through which the unbalance force in the rotor is able to be transmitted to the stator. A comparison of the normal and optimized rotordynamic features from advanced algorithms for response at the compressor bearing of the rotor in Figure 7 is shown in Figure 14. In the range from 4000 to 9000 rpm an advanced algorithm is employed to eliminate the peak response associated with the tilting mode (Mode 3) critical speed. Above 9000 rpm another advanced algorithm is employed to reduce the controller response to the synchronous unbalance force. The unbalance response comparison in Figure 14 shows that both instances of advanced algorithm applications improve the rotordynamic response versus the case without advanced algorithms.

Thus, the lateral rotordynamic characteristics of the hermetically sealed oil-free turbocompressor have been qualified and optimized via the advanced algorithms for the magnetic bearings. What remains is a final qualification of the torsional rotordynamic characteristics so that the machine reliability and availability can be guaranteed.

Torsional

A torsional analysis is always conducted to screen the compressor train for potential torsional resonance problems. Figure 15 shows the calculated undamped torsional eigenfrequencies in a Campbell diagram for the rotor in Figure 7. The associated mode shapes are depicted in Figure 16. Note, mode one is the rigid body mode at zero frequency and is not of interest here. A minimum 10 percent margin between an eigenfrequency and a one times excitation over the complete operating speed range is respected for this train. Due to the wide operating speed range, 42 to 105 percent, it is impossible to avoid coincidence of any higher, e.g., two, three, etc., times, excitation of an eigenfrequency. Since modern harmonic drives have very smooth torsional excitation signatures, harmonic excitations much less than 2 percent of rated torque, a harmonic response analysis at resonance is not necessary as the resulting torque amplitudes are negligibly small. Therefore, the only remaining potential source of torsional excitation remains a two-phase short circuit at the motor terminals. A two-phase terminal short circuit contains both one and two times driven frequency transient excitations and can excite mode two in a resonance condition, i.e., with high amplification.

Two-Phase Short Circuit Analysis. Short circuit calculations are carried out at discrete speeds over the complete operating speed range, 4800 to 12,000 rpm, to identify the peak torque responses. This analysis assumes full motor rated torque operation at each discrete speed and a conservative dynamic amplifier of Q = 80, i.e., material damping ratio of 0.00625 (D = 1/(2Q)). The occurring maximum responses are at the motor drive end and are shown in Figure 17. The absolute maximum response obviously occurs at the true resonance speed, 6815 rpm. At this resonance speed the torque response attains a level of 10.67 pu (1 pu = 4945 Nm, 6705 ftlb), which is above the hub shrink fit capacity but below the shaft stress limit. Therefore, in the unlikely event of a two-phase terminal short circuit at resonant speed the coupling hub may slip but the shaft will not break. Note that a hub slip due to a sudden over torque of short duration does not result in a catastrophic failure. Since the shaft stress limit is above that excited by the worst case transient torque excitation there is sufficient safety to
guarantee that the shaft system will operate free from any problems in field operation.

![Figure 17. Two-Phase Terminal Short Circuit Response Curve.](image)

This section has analyzed the static and dynamic characteristics of the hermetically sealed oil-free turbocompressor and has demonstrated that the turbocompressor design will operate free of any static or dynamic problems. These analyses were important to guarantee the reliability and availability demanded by modern gas storage needs. What follows is evidence of the application of hermetically sealed oil-free turbocompressors to fulfill modern gas storage demands from the initial shop test experience to the successful implementation of this compression technology in two gas storage facilities.

**SYSTEM APPLICATIONS AND EXPERIENCE**

The first order for a hermetically sealed oil-free turbocompressor was realized in 2000. This initial order was for a gas storage facility. After design, manufacture, and assembly this first turbocompressor system was brought to the shop test facility for factory testing in August 2001.

**Initial Shop Test Experience**

The primary goal of the shop testing program was to verify the predicted dynamic behavior and gain experience with the sensitivity of the turbocompressor system when it was subjected to various changes, among others, alignment, motor cooling variation, and balance weights. The initial verification of the predicted dynamic behavior of the turbocompressor system was accomplished via bearing excitation tests conducted upon the rotor both at standstill and in operation. These tests served the purpose of verifying and comparing the location and stability of measured eigenfrequencies with the predictions and confirming that prescribed digital controller parameters yielded optimum results. The initial shop test timeline is shown in Figure 18.

An example of a bearing excitation test is shown in Figure 19 that compares the location of predicted and identified eigenfrequencies (peak amplitudes) in the closed loop transfer functions. The measured transfer function in Figure 19 contains lower frequency adjustments to enhance the overall dynamic characteristics and, hence, differs slightly with the predicted transfer function. The other dynamic verification test for the compressor system is a comparison of the measured and predicted rotor response to unbalance under the influence of the advanced algorithms. Here, the use of an advanced algorithm to eliminate rotor sensitivity to the tilting mode is sought for increased compressor robustness and enhanced reliability and availability. The measured rotor response at the compressor bearing location is shown in Figure 20, which confirms the rotor insensitivity to the tilting mode and compares favorably with the predicted response in Figure 14. Note that at the approximate tilting mode (Mode 3) critical speed of 7000 rpm there is no peak response to unbalance where it would be expected without the advanced algorithm. Hence, the confirmation of predicted dynamic behavior with measurement proved that the compressor system concept functioned as robustly as expected and gave confidence for a successful turbocompressor system implementation for field operation.

![Figure 18. Initial Shop Test Timeline.](chart)

![Figure 19. Predicted and Measured Bearing Closed Loop Response.](chart)

![Figure 20. Measured Compressor End Shaft Vibration During Shop Test.](chart)
The application of hermetically sealed oil-free turbo compressor technology for natural gas storage has been realized for two facilities and a third is being commissioned at the time of writing. The following sections provide information relating to the characteristics of gas storage, plant configuration, initial experience, and some operational aspects of two existing gas storage facilities from the user's perspective.

Development and Characteristics of the Gas Storage Service Industry

The first natural gas storage facilities were developed in the early 1900s in Canada and the United States. The primary purpose of these facilities was to provide localized natural gas supply during the winter heating season. This was necessary because frequently the demand exceeded the capability of the pipeline and production infrastructure to deliver adequate natural gas into the consuming market areas.

The significant milestones related to the development of gas storage in North America and Europe are shown on the timeline in Figure 21. Of particular importance is the creation of the spot market over the past 20 years. This has lead to a significant increase in the demand for gas storage. The result being that there are now more than 400 facilities in North America and over 130 in Europe.

Gas Storage Functionality

The traditional cycle for natural gas storage has been to inject gas during nonwinter months (April through October) and withdraw gas in the winter months (November through March). This annual cycle used seasonal demand in market areas together with the long haul pipeline capacity to dictate storage operations. Over the past 20 years storage facilities have been developed with the flexibility to accommodate multiple cycles per year. This greater flexibility allows users of natural gas not only to have security of seasonal supply, but also to manage their cost of gas by buying, storing, selling, or using gas based on current market rates. The growth of the natural gas commodity market, when coupled with more flexible storage, has also created many opportunities for gas marketing companies to increase their profitability.

Gas Storage Container Types

To be effective, the underground storage of natural gas requires containers that do not leak or cause any safety risk to nearby population centers or property. The vast majority of the storage facilities utilize depleted hydrocarbon reservoirs, salt caverns, or aquifers for storage containers (Figure 22).

Gas Storage Compression Requirements and Duties

One of the most critical components in any gas storage facility is its compressors. Compressors are required to inject the gas underground for storage and can also be used to extract the gas for withdrawal. For gas injection duties the compressors must increase the gas pressure from the pipeline (typically 40 to 80 bars, 580 to 1160 psi) to the final storage pressure of the facility (typically between 100 and 200 bars, 1450 and 2900 psi, but can exceed 300 bars, 4350 psi). For withdrawal the compressor must deliver gas at the pipeline pressure with a suction pressure that can be lower than 30 bars, 435 psi.

Traditionally the duty of gas storage compressors has been cyclical in nature. Seasonal cycles created the overall requirement for injection or withdrawal. Compression was primarily used for injection with limited use at the end of the withdrawal season. Both the pipeline pressure and the storage pressure fluctuate in real time and the overall storage pressure gradually changes during the course of the seasons. These changes require compressors to provide a large operating envelope to cover the varying flow rates and pressure ratios.

In contrast to the traditional storage compression duties, spot market activity has created different duties for storage compressors. Daily demand cycles, market conditions, and short term weather patterns can require a facility to change its operational mode many times a week. Under spot market activity conditions a storage compressor may be required to be stopped and started frequently to either inject or withdraw gas. This has exerted pressure on the compressor manufacturers to yield a compressor that is: (a) flexible, must have a wide operating envelope, (b) reliable, downtime for maintenance must be reduced or eliminated, and (c) available, be able to react spontaneously to market demands and tolerate numerous stops and starts.

An additional requirement that is becoming increasingly important is the environmental aspect. Permitting fees and/or emission taxes have forced modern storage facility operators to require equipment that has reduced emissions of pollutive substances and lower noise levels.

Gas Storage Compression Equipment

The most common type of compressor dedicated to gas storage is the reciprocating compressor. This is mainly due to a low capital cost for this type of compression equipment and good performance at partial load operation. Furthermore, many reciprocating compressors are an integral configuration where the driver and compressor are in a single casing. In many instances the driver is a gas-fired engine that utilizes process gas as the fuel. This is particularly convenient in remote locations where electricity supply may be limited, expensive, or not available. The drawbacks of reciprocating compressors are the need for lubricating systems and the significant maintenance they require that limits the reliability and availability of this type of compressor. Modern gas storage demands stretch the ability of reciprocating compressors particularly...
in the areas of availability and reliability. For storage facilities to perform effectively in spot markets, a compressor that combines the attributes of high reliability and availability is essential. In addition this compressor must meet modern environmental demands.

Gas Storage Facility in New York

In July of 2002, a New York oil and gas company commenced commercial operations of Phase I of a natural gas storage facility located in Tioga County, New York. This location was selected because of the existence of reservoirs suitable for conversion to storage, the proximity to a large market (150 miles (241 km) northwest of New York City) and the proximity to a gas transportation infrastructure. This storage facility is the eastern most gas storage facility in the United States. In order to provide a high performance facility it was developed utilizing a technology to access the reservoirs with horizontal wells for storage purposes. This resulted in a high performance, multicycle facility having withdrawal capabilities up to 500 MMcf (14 million m³) per day, and injection capabilities exceeding 250 MMcf (7 million m³) per day. The working gas capacity is approximately 12 Bcf (34 million m³) at the maximum reservoir pressure of 3250 psia (224.1 bar).

The primary components of the gas storage facility are listed below.

- Two reservoirs, 10 horizontal storage wells, eight observation wells
- Pipeline gathering system
- Compressor station (compression, dehydration, filtration, and pressure letdown)
- 115 kV switch station and power line
- Pipeline lateral (connects the compressor station to the pipeline)

The compressor station is a modern, state-of-the-art facility. It combines unique architecture, advanced compression, and automated control systems. Operation of the storage facility, including the 10 storage wells, is accomplished from the operator’s console. The overall compression facility (Figure 23) is served by three identical hermetically sealed oil-free turbocompressors (Figure 24) that total 25,000 horsepower (18.6 MW). These turbocompressor systems were selected because of their highly flexible performance envelope, quick start-stop capability, low maintenance requirement, and environmentally friendly features. The operating envelope when “compressing in” with a suction pressure of 800 psig (55.2 barg) is shown in Figure 25. With the pipeline normally operating in the 800 to 1100 psig (55.2 to 75.8 barg) range, the predominant mode of operation for the storage facility is to “compress in” and “free flow out.” Toward the end of the withdrawal season, when the reservoir pressure drops below the pipeline pressure, the mode changes to “compress out.”

Commissioning and Site Acceptance Tests

Commissioning of the compressors started in early April 2002 and concluded with a successful site acceptance test of the turbocompressors in late July 2002. As is normal with a complex installation various program delays were experienced that lengthened the time required for this process. Additionally, since most of the systems associated with the compressors were either new or significantly modified, the normal types of commissioning problems were encountered.

Operations

The estimated cumulative operating time for the three turbocompressors is on the order of magnitude of 10,000 hours. Due to the nature of this facility’s operation the number of starts and stops are estimated to be in the neighborhood of 750, which are relatively high when compared to a base-load type application for compression. It is important to note that there were only a very few hours during which the availability of the turbocompressors had a negative impact on the station’s ability to meet its operational requirements.

The majority of issues that have made the units unavailable for service were related to one of the following systems or components:

- Magnetic bearing power and control wiring inside the turbocompressor.
- Motor power wiring inside the compressor.
- Auxiliary bearings resisting rotation.
• Cooling system leaks on the variable speed drives.

The availability of the three compressors from June 2002 through December 2004 is shown in Figure 26. A chronological summary of the main causes for loss of availability indicated in Figure 26 follows:

• August 2002—Magnetic bearing wiring attachment problem
• November 2002—Variable speed drive leaking coolant issues
• January 2003—Magnetic bearing wiring attachment issues
• February 2003—Motor wiring (unit 1)
• March 2003—Motor wiring (unit 2)
• March 2003—Motor wiring (unit 3)
• May 2003—Permanent fix of magnetic bearing wiring on all units
• May 2003—Variable frequency drive (VFD) ground fault likely caused by expended deionized water canister not recognized because of a faulty probe
• June 2003—Auxiliary bearing
• July 2003—Auxiliary bearing
• October 2003—VFD program upgrade
• February 2004—Magnetic bearing connector
• December 2004—Motor wiring failure

There have been some issues related to availability of repair parts, which, when combined with the need to limit facility downtime, have created lengthy periods of unavailability. However, based on the data, the authors conclude that the operational availability for these units has approached 99 percent while the overall availability from Figure 26 has been 85 percent. Most of the availability issues can be attributed to the new integrated design for the turbocompressor system and a scale up of the variable speed drives. As these early production issues are resolved, the authors believe that the overall availability of the compressors will improve markedly.

Routine periodic maintenance such as replacement of the cooling gas filters, recharging of the deionization canisters for the variable speed drives, and calibration of transmitters has been performed. No additional maintenance issues beyond those described above have arisen.

Since the system is electric driven, there are no emissions issues for the facility that relate to the turbocompressors. This allows the facility to utilize all of the emissions allotment to cover other systems, such as dehydration and boilers, for which zero emissions technology is not currently available or practical.

With the exception of the few issues described above the compression system has performed as expected. It has even performed at suction pressure levels of below 300 psi (20.7 bar), significantly below the minimum design suction pressure of 600 psi (41.4 bar), which has given even more flexibility in operation of the facility. Overall, the turbocompressor system has met the goals as a system that is flexible, reliable, responsive, environmentally friendly, and available.

**Figure 26. New York Gas Storage Facility Compression System Availability.**

Nüttermoor Gas Storage Facility in Germany

An energy service company in northwest Germany with one million electric power and 70,000 gas customers, operates two gas storage facilities in Nüttermoor and Huntorf. These two gas storage facilities provide the basis of the energy service company having been one of the lowest cost gas suppliers in Germany for the past decades. This low cost gas supply requires optimum gas supply efficiency via a supply infrastructure capacity, both of gas storage operator and the gas producers, that has the highest possible constant usage factor throughout the whole year. Otherwise, the complete technical infrastructure for production, treatment, and transport would have to be laid out for the peak usage capacity and result in costs that would lead to significantly higher user prices.

The Nüttermoor facility realizes lowest cost gas supply via the following:

• The facility provides a buffer between constant production and varying demand. Relatively constant production and contractually agreed gas purchase by the energy service company on the one side compare with enormous weather dependent demand variations. On a cold winter day, gas usage can be five to 10 times higher than in August. This storage facility compensates between constant production and seasonal demand fluctuation.

• Increased supply security. In the event of an interruption to the gas production, this storage facility makes sufficient gas quantities available to the market.

• In the liberalized European gas markets, gas trading is becoming increasingly important. This storage facility makes it possible to take low-cost excess gas, which cannot be sold at short notice, store it, and then make it immediately available again upon demand.

**Locations and Storage Capacities**

The Nüttermoor facility is integrated in a 34,000 km (21,000 mile) long supply network in the concession area and has 18 storage caverns in operation. The caverns are large voids, leached in salt domes, with geometric volumes of up to 700,000 m³ (24.7 Mcf) and maximum cavern gas storage pressure up to 170 bar. A further five caverns are currently under construction. The facility has a total of 1.1 million m³ (38.8 Mcf) of stored working gas with a total gas turnover of 4 billion m³ (173 Bcf) per year. Maximum extraction capacity is 1.75 million m³/h (61.8 Mcf/h) compared with a storage capacity of 530,000 m³/h (18.7 Mcf/h). The total installed compression inventory comprises 12 reciprocating and two centrifugal compressors with a combined power of 23.7 MW (31,782 hp). An aerial view of the facility is shown in Figure 27. The complete storage facility is remotely controlled from the energy service company’s headquarters in Oldenburg, Germany, some 70 km (43.5 miles) away.

**Figure 27. Aerial View of Nüttermoor Gas Storage Facility.**
Compressor Selection

Confronted with vibration problems on the existing reciprocating compressors as well as equipment problems caused by the ingress of lube oil into the gas stream, the energy service company made the decision to participate in the original high speed motor driven oil-free multistage compression system development program in 1990. Following the successful installation and completion of the test program, the energy service company purchased this system. After more than 10 years of operation this compression system still fulfills the requirements with respect to service and reliability.

Today’s liberalizing European gas market presents storage facility operators with completely new challenges. This facility is now required to be able to store or extract gas quantities between 10,000 and 125,000 m³/h (0.35 to 4.4 Mcf/h) with high precision and at very short notice. This is a fundamental requirement in order to participate in the gas trading market and has placed increased demands on the operation reliability and availability of the existing machinery. Furthermore, there are continuous increasing environmental requirements to adhere to.

The new hermetically sealed oil-free turbocompressor system fulfills these modern gas storage requirements perfectly. With the elimination of dry gas seals, the unit has practically no components subject to wear. Since the compressor system is hermetically sealed, with the motor cooled via process gas, the system is compact and there is no gas released to atmosphere. Due to these improvements the installation time has been reduced significantly and has made the system more manageable. A picture of the installed hermetically sealed oil-free compression system with tandem arrangement is shown in Figure 28. A tandem arrangement was chosen for this compressor system so that a wide operating range, made possible with both series and parallel mode operation, is available. The operating range is similar to that shown previously in Figure 2.

Operating Experience

The commissioning problems and issues were few and limited mainly to slight coolant leakage and winterization of the variable speed drive cooling system and the adaptation of the control system to fulfill the accuracy and startup time demand by the spot market activity. After successful control system setup the initial operating experience shows that the nominal injection quantity per hour can be injected with a maximum deviation of 1 percent. The first year and a half operation has demonstrated that the hermetically sealed oil-free compression system more than fulfills the specified requirements and has increased plant availability. The service requirements and resulting downtime have thus been extremely low. At the time of writing this turbocompressor system has operated for almost 3000 hours and has experienced more than 500 starts.

OTHER POTENTIAL APPLICATIONS

The previous sections have described how the demanding characteristics of the modern gas storage business have led to the development and successful installation of hermetically sealed oil-free turbocompressor systems for two gas storage applications. However, this turbocompressor system is not limited to solely gas storage applications but can be applied to gas compression applications that have particular constraints that conventional gas compression systems have difficulty complying with. The particular constraints that the hermetically sealed oil-free turbocompressor system can best fulfill, compared to a conventional compressor system, are environmental constraints.

Typically, environmental constraints relate to emissions of pollutive substances and hermetically sealed oil-free turbocompressor system’s positive attributes to eliminating these constraints has been described previously in the section, HERMETICALLY SEALED OIL-FREE TURBCOMPRESSOR TECHNOLOGY and the subsection, Environmental. Here, environmental constraints are extended to encompass possible restrictions pertaining to: (a) footprint or space available, (b) location and climate, and (c) health and safety aspects that can make a compression system difficult to implement. A discussion of how the hermetically sealed oil-free turbocompressor system’s compact, hermetically sealed, and all-electric attributes described above can address these additional environmental constraints are discussed below.

Footprint

The footprint or space that a turbocompressor system occupies has a direct influence upon the total system cost. This is certainly the case for offshore upstream oil and gas applications where severe space and weight constraints can be present to keep size and cost of the platform under control. The hermetically sealed oil-free turbocompressor system can address this constraint by occupying a much smaller footprint than a conventional low-speed motor driven turbocompressor skid. The extent of the difference is illustrated in Figure 29, which compares side by side a similarly powered conventional motor driven compressor skid with the hermetically sealed oil-free turbocompressor system skid. The hermetically sealed oil-free turbocompressor system skid occupies less than half the footprint and is approximately two thirds the total mass. This considerable space and weight savings is gained due to the following reasons:

- Motor size — The power of a motor is directly proportional to the motor’s torque and operating speed. Therefore, a motor that operates at a high speed will have a comparably low torque for a given amount of power. Since a motor rotor’s size, or active volume, is proportional to the torque it is rated for, a high speed motor will have a rotor size that is much smaller. For the example low-speed and high-speed motors shown in Figure 29, the high-speed motor operates at approximately three times the grid frequency and therefore, for the same power, has a rotor that is approximately three times smaller in volume than the conventional motor.
- No gearbox — The high speed motor directly drives the compressors and, therefore, there is no need for a speed increasing gear.
- No couplings — The compressors are directly coupled to the motor and thus save considerable train length.
HERMETICALLY SEALED OIL-FREE TURBOCOMPRESSOR TECHNOLOGY

- Single bearing per compressor—There is no compressor bearing on the compressor driven end and this translates into reduced compressor rotor length.
- No lube-oil system—Due to the use of magnetic bearings there is no lube-oil system with its voluminous holding tank, pumps, heat exchangers, filters, etc.
- No shaft seal instrumentation rack—Since there are no shaft seals, no instrumentation and ancillary equipment associated with such systems are required. Instead, a much smaller instrumentation rack for flow and pressure transmitters is needed.

Figure 29. Comparison of a Conventional Turbocompressor System with a Hermetically Sealed Oil-Free Turbocompressor System (All Dimensions in mm).

The footprint of the hermetically sealed oil-free turbocompressor system can be further reduced by mounting the system vertically—the footprint will then be one half or less that of a horizontal arrangement. This can be accomplished without any rotordynamic drawbacks or concerns because the magnetic bearings are neutral to the rotordynamic behavior whether the rotor is horizontal or vertical.

Since the motor operates at frequencies that are higher than the common grid frequencies (50 and 60 Hz) it requires a variable frequency drive and the size of the VSD must be considered as part of the system. A VSD increases the overall system footprint and weight until it becomes the same size or slightly larger than a conventional fixed speed solution. Nevertheless, the VSD provides the benefits of variable speed operation, higher partial load efficiency, and low motor starting currents that enable multiple start operation profiles. Hence, if a gas compression duty can be fulfilled by a fixed speed compression system, the application of a hermetically sealed oil-free turbocompressor is a feasibility balance of the additional benefits it brings versus its additional cost—where the additional cost is mainly due to the difference that the hermetically sealed oil-free turbocompressor system has a VSD while the conventional does not.

Location

The location of a turbocompressor system can cover a variety of aspects such as: (a) climate, (b) remoteness, (c) neighborhood, and (d) physical boundaries that can constrain a turbocompressor system installation. A discussion of the above identified aspects follows:

- Climate—Certain climates, such as arctic, can severely limit what type of equipment can be deployed or can require additional subsystems to keep the equipment at acceptable temperatures for sound operation. A notably sensitive component to arctic climates is a lube-oil system because it requires additional heaters and, sometimes, very expensive synthetic oil. The most typical problem with machinery in arctic climates is a difficulty to start the system—multiple start-stop operation profiles are very difficult to achieve. The hermetically sealed oil-free turbocompressor system addresses this problem by containing no parts or subsystems that are severely affected by temperature extremes—either hot or cold—and can reliably be stopped and started at will.
- Remoteness—Underground gas bearing formations are quite often found in regions that are quite inhospitable to humans; therefore, it can be advantageous to have a completely unmanned compression station that is remotely controlled. Furthermore, the cost of a compressor installation that does not have to make provisions for continuous human presence is dramatically reduced when compared to a manned installation. The hermetically sealed oil-free turbocompressor facilitates remote control by providing an all-electric system and contains no substances that need to be routinely maintained or replenished by human intervention such as can be found in lube-oil or shaft-seal systems. Several turbocompressor systems of the hermetically sealed oil-free pipeline type are already remotely controlled and some from a distance 5000 km (almost 3000 miles) away.
- Neighborhood—In contrast to the location remoteness discussed above it is possible that a gas compression plant is located near a residential area. In this situation emissions via a flare line, noise, and visual impact play important roles regarding what type of compression equipment can be installed or not. The hermetically sealed oil-free turbocompressor system’s attributes facilitate the acceptance of compression equipment in or near a populated area by reducing or eliminating the undesirable features of: (a) flares, due to the hermetic sealed design, (b) noise, due to the all-electric aspects, and (c) visual impact, by providing a compact solution that does not require large infrastructures. An installation of a turbocompressor system of the hermetically sealed oil-free pipeline type has already been realized in an affluent neighborhood in North America where the compression station appears as nothing more than a typical residential home.
- Physical boundaries—Consider a compressor system that is surrounded by fluid that is not air—suddenly, there is a completely new and different physical boundary the compressor system is subject to. This is just the case when subsea compression is considered and the physical boundary is that the complete compression system is submerged in seawater. Subsea compression is the application that has generated the most interest for the hermetically sealed oil-free turbocompressor system because its attributes of all-electric and remote operation, hermetic sealing, and few as possible train components or subsystems create a good basis for this demanding application. Also, the ability of the magnetic bearings to rotordynamically control a rotor mounted vertically or horizontally is seen as an additional benefit to optimize the system size and configuration for deployment and service by ship.

Health and Safety

Optimum health and safety aspects of a gas compression system can best be achieved when the system is remotely controlled, i.e., unmanned. Clearly if there is no continuous human presence in the vicinity of the machinery, health and safety aspects are reduced to the few occasions when human intervention is necessary for inspection or maintenance. The above discussed application of a hermetically sealed oil-free turbocompressor system to subsea
compression effectively removes health and safety constraints because intervention can only be performed on the surface (and the compressor system is completely removed from the process) or by a remotely operated vehicle (where there is no human contact with the system). Nevertheless, upon human intervention, all the necessary precautions have to be adhered to regardless of the equipment type.

FUTURE DEVELOPMENTS

The above discussions have shown that the hermetically sealed oil-free turbocompressor system can provide an attractive solution for gas compression applications that are environmentally constrained. Many of these applications are located in the upstream oil and gas market where the process gas is quite often corrosive. Because the motor and magnetic bearings (active electric components) are submerged in the process gas, a corrosive process gas can adversely affect their performance to the point that the compression system is no longer reliable and available. Therefore, the original hermetically sealed oil-free turbocompressor system that was developed to solve modern gas storage demands must be adapted to handle potential corrosion problems of active electric components. The first step in this adaptation has been realized by the magnetic bearing manufacturer via the design and introduction of a canned magnetic bearing.

The canned magnetic bearing design encapsulates the complete magnetic bearing within a corrosive resistant shell (can) that hermetically seals the bearing internals from the process gas. A comparison of the canned bearing with a standard bearing is shown in Figure 30. The can prevents contact of corrosive process gas with the electrically active parts of the magnetic bearing.

![Figure 30. Standard (Left) and Canned (Right) Magnetic Bearings.](image)

A canned bearing was installed in the compressor system for the Nüttermoor storage facility to gain industrial experience and prove that the canned bearing performance does not differ from the standard magnetic bearing. Prior to installation the canned bearing was subjected to rigorous thermal and pressure tests to qualify it for industrial use. This bearing was installed in the compressor system after a successful 500 hour test with standard magnetic bearings. The 500 hour test provided baseline information with which the canned bearing could be compared. After more than 2000 hours of successful operation the canned bearing was removed for inspection. The inspection of the canned bearing revealed no damage and a completely gas tight can. This positive industrial experience provides confidence that the canned bearing can be installed for future applications in corrosive environments without loss of magnetic bearing performance.

CONCLUSION

A hermetically sealed oil-free turbocompressor system has been developed and introduced to answer the multicycle high flow-in/flow-out capability on very short notice demanded by the modern gas storage market. This compressor system has been installed in two gas storage facilities and fulfilled the demands and expectations of the market and storage operators. The initial installation experienced several issues that temporarily reduced the availability for the first years of operation. These issues have been addressed and permanently fixed. The second installation did not experience these issues proving that the lessons were learned, solved, and successfully implemented.

The hermetically sealed oil-free turbocompressor system was also shown to provide an attractive solution to gas compression applications for which environmental constraints of emissions, footprint, location, or health and safety are present. Because of the positive environmental attributes there is increasing demand to apply this hermetically sealed oil-free compression system to upstream oil and gas process such as offshore unmanned platforms and subsea gas compression installations. To answer these demands the hermetically sealed oil-free compressor system will be adapted to handle corrosive gases typical of these upstream processes. The initial step of this adaptation has already been accomplished with the introduction of a canned magnetic bearing.

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