Ensuring integrally-geared compressor reliability with API 617

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Summary

Integrally geared centrifugal compressors (IGCC) are widely used in various industries. In some they are state of the art, in others they are accepted. However, there still are reservations against IGCCs. API 617 addressed these reservations by extending the original scope such that it also covers integrally geared technology. Some examples of how the intent of this standard is put into existing designs are presented in this paper. When comparing single shaft centrifugal compressors it shows where IGCCs are best used, i.e. where they have advantages over other technologies and where their limitations are. This report summarizes, based on some examples, which components and procedures are addressed by API: The gearing within the compressor is an integral part of this compressor type as are the overhung rotors. The lateral analysis during the design phase safeguards against excessive vibration levels, bearing design supports the dynamic behavior and helps to prevent instabilities. Seals, usually dry face seals, minimize the leaks. All these elements, and many more, are considered by API. This series of standards provides assurance for minimal downtime and safe operation.

1. Introduction

IGCC was conceptualized in 1930's, commercialized in 1950's and have been used by various industries ever since then including industrial gases in which they serve critical plant process without redundancy. Beginning with the seventh edition, released in 2002, the standard API 617 explicitly considers in part 3 the peculiarities of IGCCs for the hydrocarbon market segment. This part complements the parts for expander-compressors (part 4) and centrifugal and axial compressors (part 2), both of which do not exhibit gears as an integral part of their design.

The design of the compressors handled in part 2 differs from the design of those addressed in part 3, In *Inline compressors* (IC) or *beam-type compressors*, all the impellers are located on one rotor and rotate with the same angular speed. There is only one rotor, two bearings, and one seal separating the process to the environment. All other seals prevent the gas from leaking to adjacent stages. Figure 1 exhibits an *integrally geared compressor*: Each rotor has one or two stages, a seal for each stage and two bearings for each rotor. The gear which determines the speed of the rotors is an integral part of the compressor, for which is accounted for by its name.

Both configurations use *centrifugal compressors* as opposed to *axial compressors*. While for both compressors the gas enters the impeller in axial direction, in a centrifugal compressor the gas exists the impeller in radial direction while in an axial compressor, the gas exits in axial direction.

An axial compressor stage is capable of a higher volume flow than a centrifugal stage of the same diameter and running at the same speed. On the other hand the centrifugal stage has a much higher head rise (which translates to higher pressure ratios) for the same diameter and running speed. For a given overall pressure ratio, more stages are needed for an axial compressor compared to a centrifugal compressor. Axial compressors have a smaller range of operation and are usually more expensive, see (Boyce, 2003).

It is hardly possible to make universally applicable statements regarding the differences between ICs and IGCCs: There are always cases in which the usual differences between the two designs become negligible. Hence, the following statements have to be taken with some caution. While there are operation ranges where an IGCC is the clear choice over an IC and vice versa, there are also ranges where both technologies are technically viable.

2. Preferred range of operation

The impeller design of IGCCs and ICs is very similar. The difference of these technology predominantly lies in the rotor configurations which, in turn, cause some other differences in the design. These differences present themselves in multistage machines – the more stages, the greater the difference.

The task of each compressor is to increase the pressure of a given gas flow. With each stage, head is added to the gas. With ICs the stages have about the same impeller diameters and the stages run at the same speed. Therefore the flow coefficient decreases from stage to stage. For IGCC the rotational speed increases from stage to stage (speed remains the same, if two stages are on one common rotor), consequently, the impeller diameter is progressively getting smaller. This enables the impellers run at their optimal flow coefficient, i.e. each impeller is running more efficiently.

Figure 1: Packaged integrally geared compressor.

For IGCCs, it is also easier to incorporate interstage cooling. In ICs, the discharge of one stage is often directly fed into the inlet of the next stage. Interstage cooling helps keeping the compression closer to an isothermal compression, which is the most efficient process.

In many cases the additional efficiency outweighs the power losses in the additional bearings, gear losses, and the losses through the seals. For example refrigeration compressors, fuel gas boosters, and air separation units, nowadays integrally geared technology is widely used.

For a given gas and pressure ratio but with increasing mass flow, the speed ratio of the first and last stage decreases. Hence, the larger the mass flow through the machine is, the lower is the gain in efficiency due to integrally geared technology. Above a certain mass flow, the use of integrally geared technology does not add value.

The advantage of integrally geared technology over a broad range of pressures and mass flows is accepted in most industries. However, sometimes the reliability is questioned. The usual argument points to the higher complexity of the machine which is, supposedly, more prone to failure. The demands of API standards and the track record of integrally geared centrifugal compressors running for decades prove the reliability of the IGCC technology.

3. Selected solutions in IGCC design

3.1 – Gear design

By definition, the gear is a central part of an integrally geared compressor. In most cases the compressor is driven at a low speed, i.e. by a two or four pole induction or synchronous motor. Depending upon the country of operation, this results in a speed between 1480 rpm and 3600 rpm. For custom made machines, any other input speed can be used. Oftentimes, steam turbine drives do not need an intermediate gear to drive the compressor.

The gear fulfils two tasks: In the first place it transforms the driver speed to the required rotor speeds, but it also moves the rotors' center lines apart from each other to avoid collision of the stages. In Figure 1 it can be seen, that the two stages on the non-drive-end-side of the gear box need a minimum space for housing clearances and would not fit on a smaller gear box. Particularly when a second split line is needed, space can become an issue, see also Figure 8.

In part 3, section 4.11 API-617 deals with integral gearing in general and in subsection 4.11.3 with gear rating. This is where the standard refers to API-613 "special purpose gear units for petroleum, chemical and gas industry services".

API-613 is broad: It covers not only the gear rating, but also lubrication systems, instrumentation, and controls. In addition it is also very conservative. API-613 is based on the AGMA standard which was in effect in 1977. This standard was based on so-called "grade 1" material. Today, better grades are used, but API still uses the old data. Hence, API-613 is the most conservative. For a detailed discussion see (Rinaldo, October 2016). Being conservative is not always helpful. According to the same source, API meanwhile acknowledges its being conservative and allows alternative ratings, if the pitch line velocity becomes too high or the required face width is too wide.

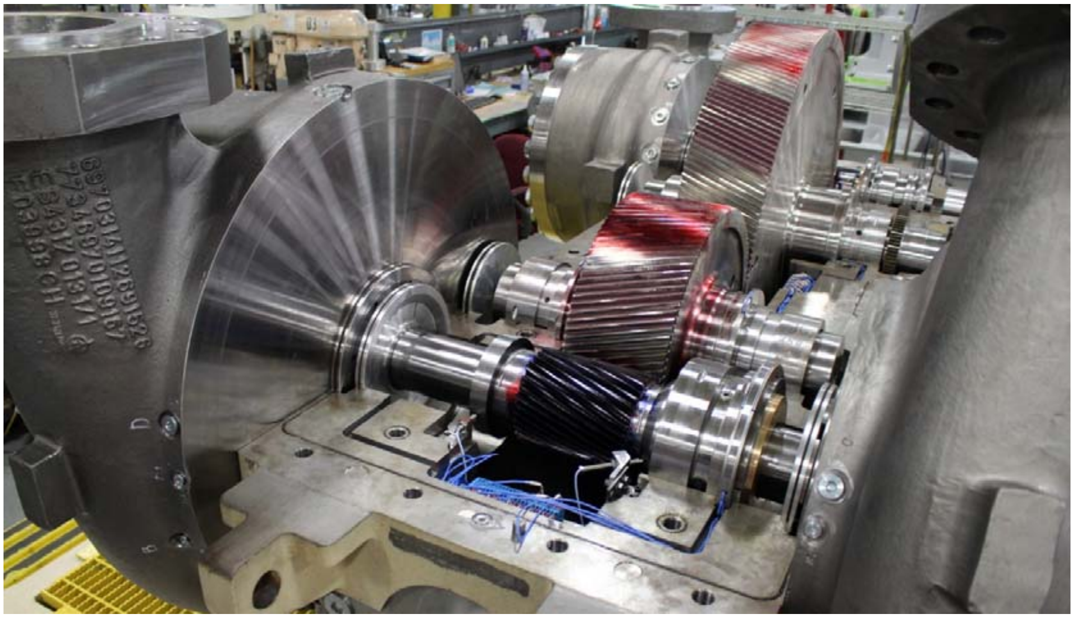

Figure 2: Overhung rotors in an integrally geared compressor (during assembly).

One input parameter in the design is the transmitted power. In addition to the normal operation, the gear design also considers transient operation of compressors. This is particularly important with the main driver being a synchronous motor. While turbine driven machines start-up without significant peaks in torque, induction motors will show an elevated torque during start-up. This peak is usually about 3 times higher than the nominal torque.

The highest transient peak during start-up occurs with synchronous motors. The reason is an unavoidable resonance which the drive train has to drive through. API-617 does not go into detail, but expects a transient torsional analysis to be performed if synchronous motors are used. The resulting torque-peaks can, for example, be reduced in the motor design and the coupling stiffness. Without going into further detail, API-compliant machines are safe to run for 20 years even if peaks loads occur during start-up.

3.2 – Lateral behavior

In integrally geared compressors the rotors are designed with the impellers on the end of the rotors. The bearings are between the impellers, and between the bearings is the gear through which the rotor is driven. This rotor design is called an overhung design, because the impellers are not between the bearings but hang out to both sides of the support. The physical design is shown in Figure 2. A typical model for the lateral analysis in Figure 3.

Usually, rotors with two impellers operate above the second critical speed and rotors with one impeller above the first critical speed. The resonance frequencies of the rotor depend upon the rotor speed. Critical speeds are resonances of the rotor with respect to unbalance excitation. Hence, during start-up of the compressor the rotor has to pass one or two critical speeds. Various API standards, amongst them API-617 and the general rotordynamic tutorial API-684, educate about lateral dynamics and provide requirements for rotors with regards to stability, unbalance response and separation margins.

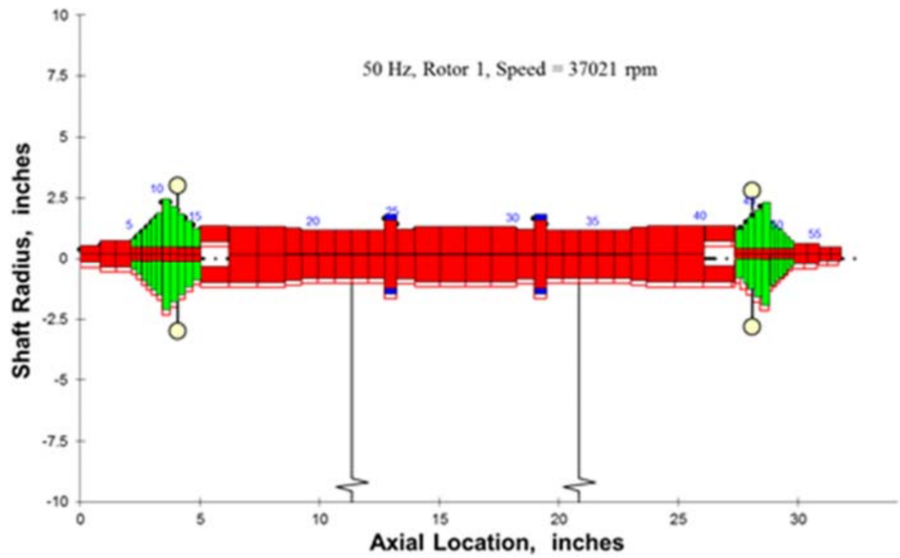

Figure 3: Model of an overhung rotor with two impellers.

Usually, the first step in a lateral analysis after modeling the rotor bearing system is a critical speed map. For this representation no damping is considered and the bearing stiffness is equal for both bearings and equal in every direction of the bearings. This is a strong simplification. However, the purpose is a graphical representation, where the rotor is operating relative to the critical speeds and how sensitive the rotor-bearing system is with regards to the bearing stiffness. Figure 4 shows a critical speed map of the overhung rotor from Figure 3. The horizontal axis represents the bearing stiffness and the vertical axis the undamped critical speeds. The four black vertical curves mark the actual bearing stiffnesses. The black horizontal line represents the operating speed. The map shows that the rotor will run above the second critical speed but below the third one. (Note, that the axes are scaled logarithmically.)

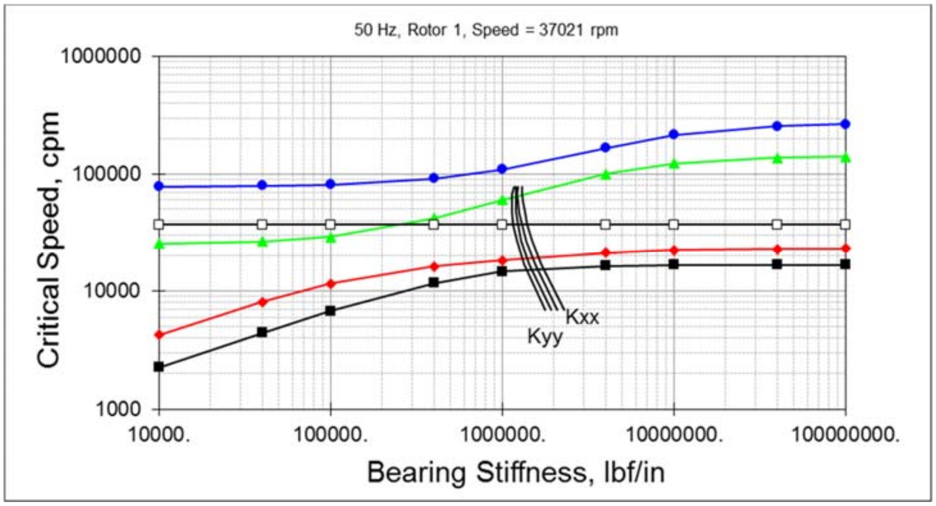

Figure 4: Critical speed map.

The concept of stability focuses on free vibrations at the natural frequencies at running speed of the rotor. The first value of interest is the damping. To quantify it a variety of measures are available. API uses the logarithmic decrement (log-dec): It measures the ratio of displacements of two subsequent peaks of a free vibrations. A rotor should have a log-dec of at least 0.1, which means that each vibration peak experiences about a 10% decrease in displacement compared to the previous peak.

Other results from the stability analysis are the natural frequencies of each mode. Here is where confusion usually begins, mainly for two reasons: natural frequencies refer to free vibration and resonance frequencies refer to a forced vibration. In addition, different excitations (external force, support-displacement, or unbalance) result in different resonance frequencies. The higher the damping gets, the bigger the difference between natural frequency and the resonance frequency. So, the result of the stability analysis according to API-617 gives a hard criterion for the damping and some information regarding the location of the natural frequencies and resonances.

The single most important excitation mechanism is unbalance. Resonance frequencies for unbalance are higher than the natural frequencies of the excited mode, and even higher than the undamped natural frequencies. The analytic assessment of a rotor-bearing system with regards to unbalance is based on a model which comprises the mass and elasticity of the rotor, especially the gyroscopic effects of the rotor, and a linearized model of the bearings. The bearing model yields linear stiffness and damping coefficients, which are dependent upon the direction – i.e. they are inhomogeneous. In combination, the damping, the gyroscopic effects, and inhomogeneous coefficients are the reasons why rotors exhibit a particular vibration behavior. While a non-rotating axial-symmetric beam always has a set of two modes at one frequency, the gyroscopic effect splits this pair of modes into two modes with different frequencies. At each mode, any point of the rotors centerline describes an elliptical orbit. Depending upon whether the point follows the orbit in the direction of the rotor's rotation or not, it is called forward mode or backward mode.

Computer software is used to model the rotor and calculate the rotor's reaction on various scenarios of unbalances which can occur at a rotor. API-617 defines these cases in terms of absolute values for the unbalances, locations where they occur, and in which direction they act relative to each other. Usually, three cases are calculated: In the first case, two unbalances act from each impeller in the same direction. In the second, both act in opposite directions. The purpose is to excite two modes to which often are referred to as rigid body modes – the conical and cylindrical modes. A closer look reveals, that at these modes the rotor is by no means rigid. The names are rather traditionally used or help visualize the rotor behavior. The third case, which is part of the unbalance response calculation, simulates three unbalances acting on the rotor. They are distributed such, that they bend the rotor, which simulates the worst case to excite the third mode. Two unbalances are acting on the impellers in identical direction, while the third unbalance acts on the gear in the opposite direction.

Results of the unbalance response are the resonance frequencies at running speed, the amplification factor (which is just another way of quantifying damping), and the response of the rotor to unbalance. API-617 defines limits of how close the operation speed may be to a resonance. At low amplification factors (less than 2.5) it is even allowed to operate directly on a resonance, because the rotor's response will be tolerable.

An unbalance response plot of the rotor from Figure 3 is shown in Figure 5: The left side graph shows the unbalance response with cylindrical excitation and the right side graph shows conical excitation. As seen in the undamped critical speed map, Figure 4, the rotor passes its two resonance frequencies before it reaches its operation speed. During operation the rotor will not be susceptible to unbalance because its operation speed is sufficiently different from the resonance frequency. The relative difference is called separation margin and its limits are defined by API.

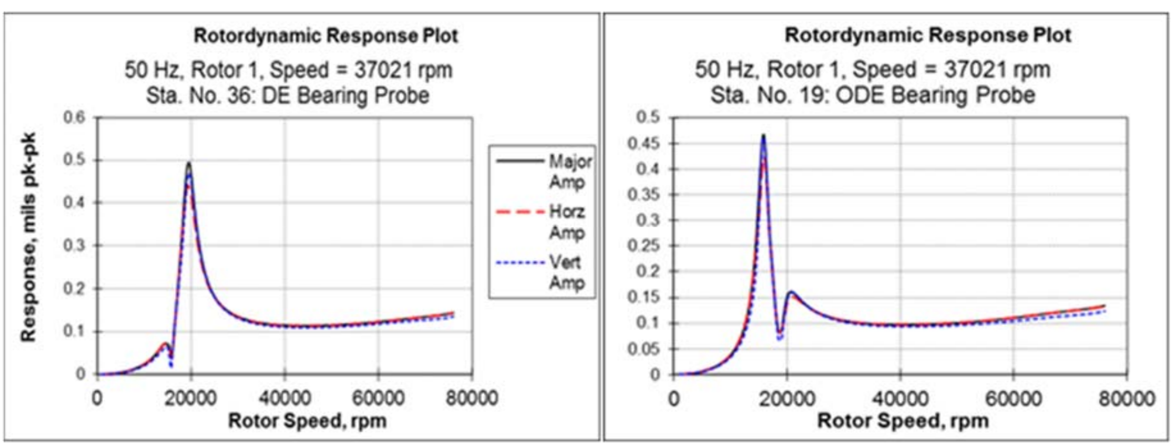

Figure 5: Unbalance response.

The weights used in the analysis correspond to the balance criteria of the physical rotor. By analyzing the rotor in this manner, the simulation reflects a worst case of what the rotor is expected to show during operation. This directly leads to a further check, defined in API-617, section 4.8.3, the "unbalance rotor response verification test". During this test the results of the analytical model are compared to the behavior in the machine.

In summary, API-617 focusses on the peculiarities of overhung rotors as they are used in integrally geared turbomachines. A thorough analytical simulation is complemented by a test of the actual rotor which further evaluates the model and guarantees a safe operation.

3.3 – Bearings

For high speed rotors tilting pad bearings are used for support while for the bull gear fixed journal lobe-bearings are used. Both types are dynamic bearings, which means at standstill the rotor has contact to the bearing. The need for tilting pad bearings arises from cross-coupling effects. These effects act detrimental to the rotor's stability. At the bearings, these effects can be eliminated by using tilting pad bearings. However, cross-coupling still can be caused by seals – especially at gases with high density and high pressures. API-617 provides in section 4.8.3 (Level I stability test) a check as to whether cross-coupling effects have to be considered. If not, a so-called level-I-analysis is sufficient. Otherwise a so-called level-II-analysis is needed which takes into account these effects. – Level II analyses are, for example, used for CO2-compressor in fertilizer plants. These compressors are centrifugal compressors which compress CO2 from atmospheric pressure up to more than 200 bar in eight stages, see Figure 8. The critical pressure of CO2 lies at 73.9 bar, so the last stages work on a very dense fluid.

Bearings are a central element in the rotor design. Each rotor is supported by two journal bearings and usually two axial bearings. It is also possible to use thrust collars to transmit axial forces in the rotors onto the bull gear. In this case, the axial bearings of the bull gear have to carry the resulting axial force of all rotors. For sake of brevity, only journal bearings are further discussed.

In integrally geared compressors the power is transmitted by the gear, and the reaction forces act in the bearings. These forces are quite high: While it can be advantageous to use magnetic bearings in compressor expanders, in integrally geared equipment the forces are too high for this technology.

Lobe bearings exhibit good damping properties and they are inexpensive: the lack of moving parts in their design makes them a rather simple element which should be used as long as it is possible. However, there are limits. At higher speeds, the rotor-bearing-systems equipped with lobe bearings will become unstable, see for example (Robert Gasch, 2013).

The journal bearings for high speed rotors are tilting pad bearings. Their advantage is the absence of destabilizing forces on the rotor. The rotor-bearing-system can be subject to resonances, but this system will not become unstable due to the bearings. In terms of the stability check. With tilting pad bearings the real parts of the eigenvalues of each mode remains negative, while an instability would be reflected by a positive real value.

The general setup of a tilting pad journal bearing is depicted in Figure 6. The bearing comprises a shell, nozzles, and pads. These pads support the rotor. Their bodies are usually made out of stainless steel or a copper/chrome alloy if there is the need of increased heat transport. Both metals have the sufficient strength to carry the load without plastic deformation. This strength is needed, since the radius of the pads' back is smaller then the inner radius of the shell. This enables the pads to tilt and align their position as needed to support the rotor.

On the rotor side, the pads are covered with a soft metal layer, called babbit. Babbit is any of several alloys which is characterized by its resistance to galling. In the case this surface gets damaged, it is quite easy to re-babbit the pads as opposed to replace the whole bearing or shaft.

The babbit comes into contact to the rotor only at standstill. During operation it is separated from the rotor by a thin oil film. The oil is injected through the nozzles between the pads and fulfils two tasks: the obvious is the lubrication, the other is heat convection.

The radius of the pad is slightly bigger than that of the rotor. This difference in their radii and the clearance of the bearing define the so-called pre-load. The pre-load, the clearance, and the bearing diamter are usually the design parameters used to determine which bearing to use. The preload, i.e. the bigger radius of the pad compared to the rotor, enables the pad to ajust. Interacting with the rotor and the oil between the rotor and the pad, the tilting pads move into a position such that the oil is building up a wedge, i.e. the gap between pad and rotor is getting narrower which also increases the oil pressure in the resulting oil film. After a maximum pressure, the oil film increases again and the pressure drops.

Figure 6: Tilting pad journal bearing.

During operation the oil film between the pad and the shaft is sheared with a high rate. This energy lost depends amongst other parameters upon the bearing load, the speed, the oil viscosity, and the geometry of the bearing. The losses associated with it directly lead to the second task of the oil: the heat transmission. The majority of the produced heat is moved out of the bearings not by conduction through the bearing shell, but through the oil.

Usually, the babbit is not the weak spot of a bearing with regards to temperature, but the oil itself. The failure mode is the degradation of the oil and its additives which build up a layer on the bearings. This layer grows and reduces the gap of the bearing. Generally, these cases are detected by temperature sensors in the pads. Their presence is required and the limit temperatures are defined by API-617 (part 3, section 4.9) which refers also to the standard API-670 “Machinery Protection Systems”.

The sufficient supply of oil in the bearings is guaranteed by properly designing the supply nozzles. A mechanical running test of the compressor is required by the standard and defined in detail by API-617 (part 3, section 6.3). During this test, the pad temperature is monitored as well as the oil supply pressure. During operation, both quantities are constantly recorded and will cause alarms and trips when exceeding the corresponding set points.

The single most frequent reason for bearing failures is caused by using oil other than the type recommended by the manufacturer.

3.4 – Seal design

API-617 considers seals in several aspects. These considerations are complemented by API-614 (5th edition) "Lubrication, Shaft-sealing and Oil-control Systems and Auxiliaries". The purpose of the seal on high speed rotors is to separate the oil from the process.

The quantity of seals used in IGCCs might raise a concern in several ways, for example: (1) seals can suffer a catastrophic failure, (2) the accumulated gas leakage over seals must be managed, and (3) seals affect the lateral dynamics of the rotors – usually not favorably.

Seal types used in turbomachinery comprise of labyrinth seals, floating carbon ring seals, liquid film seals, and dry face seals. Often two types of seals are combined at the single compressor stage.

Regarding their influence on the lateral behavior of the rotors API-617 lists in section 4.8.6 labyrinth seals amongst the elements which have to be considered while checking the stability of the rotor. This is referring to the cross-coupling-effect. Especially at high pressures and high densities of the gas within the seal, there act forces in circumferential direction of the seal surface. These forces have a de-stabilizing effect and are referred to as cross-coupling. API-617 provides a way to assess as to whether these effects have to be considered in the lateral analysis.

All seals influence the lateral behavior of rotors by adding mass and length to the rotor. Both, contribute to the dynamic flexibility of the rotor. Flexibility in a rotor is usually not desirable unless it is controlled.

To minimize seal leakage, usually dry face seals are used. Dry face seals are combined with labyrinth seals. The space between the seals build up chambers which, in turn, are connected to ports for seal gas, inert gas, and the flare. Several configurations are possible, but in general filtered and dry process gas (taken e.g. from the stage outlet) is injected between the actual dry gas seal and an inner labyrinth, which separates the process from the dry gas seal. With this technique, no dirt or liquid can enter the dry gas seal from the process side. Only a very small amount of gas travels across the dry face seal. See Figure 7 for a sketch in which the inner labyrinth seal is located behind the impeller.

From the bearing side, the dry face seal is protected against oil by an inert gas which keeps to one side the oil out of the seal and to the other side it keeps the process gas entering the inside of the gear box. The flows of the process gas (coming across the dry face seal) and the inert gas (coming across a labyrinth) are transported to the flare or safe area.

In dry face seals two rings have a circular contact surface. One ring, the primary ring, is fixed to the gear box and stationary while the second ring, the mating ring, is fixed on the rotor. The flow path of the gas is between these two rings. The primary ring is usually made out of a soft carbon graphite material while the mating ring is usually made out of tungsten carbide, silicon carbide, or silicon nitride. Together they build a soft/hard combination.

During standstill, the surfaces are in contact and pressed together by springs. It is the stationary ring which exhibits some flexibility in the axial direction, while the mating ring is rigidly fixed on the rotor. Since the surfaces are in contact, the seal leakage is close to zero.

The mating ring has groves on the seal surface which pumps gas between the rings and cause the ring lifting off as soon as the rotor starts to turn. The gap between the rings is very small, usually around 5 to 10 micrometers. The dimension of the gap and the relative movement of the rings guarantee only a small amount of seal leakage.

Dry face seals come in a variety of configurations, three of which are single, double, and tandem. At tandem seals "During normal operation, the primary seal absorbs the total pressure drop to the user's flare or vent system, and the secondary seal serves as a backup should the primary seal fail", (Stahley, 2005). While "double opposed gas seals are used in process gas applications where it is necessary to eliminate all process emissions, even to flare. This may be required to comply with environmental regulations or in toxic gas applications."

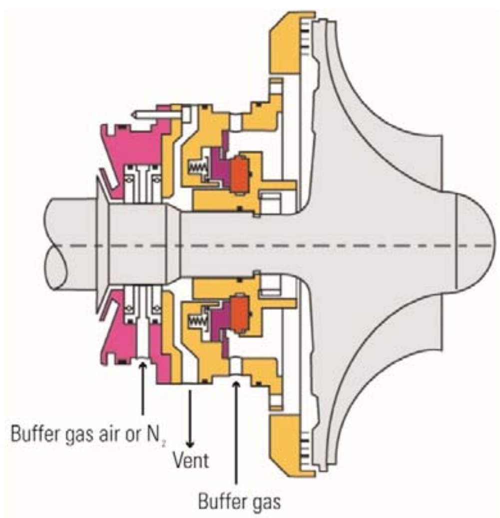

Figure 7: Single dry face seal.

API-617 and API-614 define the required instrumentation for dry gas seals and some details regarding the setup.

Dry face seals are the seal of choice in critical applications, like in fuel gas boosters, as well as in high pressure applications – see Figure 8 for an 8-stage CO2 compressor, where dry face seals needed to be developed to deal with gas pressures as high as 205 bar. As any other part in a compressor, during scheduled maintenance events, the dry face seals are inspected. Their reliable operation is guaranteed.

Figure 8: CO2-Compressor with eight stages – using dry face seals.

4. Conclusion

The API standards are the commonly used standard in the hydrocarbon market. They establish rules for design, testing, and operation with the objective of safe and reliable operation for 20 years. Since its seventh edition in 2002, API-617 comprises a part dedicated to integrally geared turbocompressors. By doing so, API acknowledged the practice that this type of compressors became increasingly popular even in the industry for critical processes and services due to its superior efficiency in some ranges – paired with similar reliability compared to centrifugal inline compressors.

References

Boyce, M. P.: Centrifugal compressors: A basic guide. Tulsa, Oklahoma: PennWell Corporation, 2003.

API – American Petroleum Institute: API Recommended Practice 684, API Standard Paragraphs, Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals, and Rotor Balancing, second edition, August 2005.

API – American Petroleum Institute: API Standard 617, eighth edition, Axial and Centrifugal Compressors and Expander-compressors, September 2014.

Rinaldo, J.: "Gearbox specs: Getting it right", Turbomachinery International, October 2016.

Gasch, R. & Nordmann R.: Rotordynamik, Berlin, Heidelberg, New-York: Springer, 2013.

Stahley, J. S.: Dry gas seals handbook. Tulsa, Oklahoma: PennWell Corporation, 20