44TH TURBOMACHINERY & 31ST PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 14 – 17 2015
GEORGE R. BROWN CONVENTION CENTER

# CAUSES OF SUBSYNCHRONOUS VIBRATION IN INTEGRALLY GEARED COMPRESSORS

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## ABSTRACT

Integrally geared centrifugal compressors (IGC's) are a well-established and continually growing segment of the turbomachinery industry. They have proven to provide a high level of reliability, efficiency and design flexibility. Although the applications and technology of IGC's have continued to grow and advance, there are few publications that address sub-synchronous vibration (SSV) specific to IGC's. Subsynchronous vibration can be either destructive or harmless depending on the root cause. Misdiagnosing the root cause can lead to destructive failure or unnecessary design changes. As integrally geared compressor design and applications evolve traditional design limits are tested. Pressure ratios, rotor flexibility, bearing loads and compressor operating range continue to be increased to satisfy the end users process requirements in the most efficient way possible. When addressing a SSV issue on an IGC one must keep in mind that there can be multiple root causes and each call for a different and sometimes unique approach to eliminate. Although some consider interpreting and diagnosing machinery vibration is an 'art', following a systematic approach will save time and money. This paper will review some of the known causes of SSV in IGC's, typical characteristics of each, and how vibration monitoring techniques can be applied and interpreted in order to determine the root cause and solution. Several examples will be presented to illustrate the vibration characteristics associated with the root cause.

## INTRODUCTION

Subsynchronous vibration has received a large amount of attention and research due to its potentially destructive nature and often complex origin. Understanding this phenomenon has expanded with continual advances in computing power, vibration monitoring capabilities and of course firsthand experience. This understanding must continue to grow as new technologies and designs are implemented. Historically, subsynchronous vibration issues become more frequent as the driven rotors are operated at higher speeds, become more flexible and lighter.

There are several sources of subsynchronous vibration in IGC's and most causes will have common vibration signatures specific to it that can be used as a means to identify the root cause or causes. In some cases the solution can be simple and inexpensive, and others much more involved. Although SSV can be destructive, research has shown that not all SSV is necessarily unstable. Integrally geared compressors also have several design features that allow the variation of key process variables that will aid in identifying a solution to the SSV. Below are four causes of SSV that are the primary focus of this paper.

1. Rotordynamic Instability (Self-Excited)
2. Looseness of bearings in their housings
3. Insufficient Lubrication (Dry Friction)
4. Transmitted Vibration Interaction

The last section covers additional causes covered in less detail. There are resources available further detailing the theory involved with each of the topics discussed, but very few addressing them specific to IGC's. This paper will provide an overview of causes of SSV specific to IGC's, and better familiarize end users and machinery engineers with the symptoms and possible solutions.

## BASICS & TERMINOLOGY

Before these vibration phenomena can be discussed in detail a brief review of rotordynamic terminology is necessary. Figure 1 shows a diagram of the basic forces that act on a shaft rotating counter-clockwise within a lubricated clearance. The speed at which the shaft rotates is called the synchronous speed (N, rpm's). The shaft center travels around its synchronous orbit once per shaft revolution. Each dot represents the completion of one shaft revolution. The size of the synchronous orbit is equal to the shafts synchronous vibration amplitude. The distance from the center of the bearing to the shaft center is referred to as the eccentricity.

Figure 1 shows the condition in which the shaft moves along another orbit path while it rotates, also known as the whirl path. The speed that the shaft moves around the whirl path is called the non-synchronous speed or frequency. If the non-synchronous speed is slower than the synchronous speed, then it is called subsynchronous. During subsynchronous whirl the shaft will make several synchronous revolutions before making one full rotation around the whirl path. For the case shown in Figure 1 the shaft makes 5 synchronous revolutions while completing one full whirl (illustrated by the 5 dots around the whirl path). The frequency of non-synchronous vibration is often referred to in terms of its whirl frequency ratio. Therefore the subsynchronous whirl shown in Figure 1 is at a whirl frequency ratio of 1/5th the synchronous, or 0.20N.

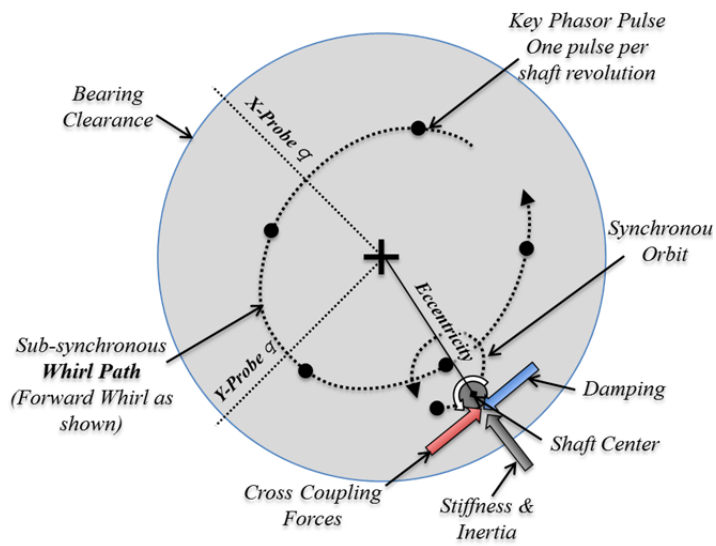



*Figure 1: Forces Acting on a Rotor*

The non-synchronous orbit may or may not be larger than the synchronous orbit, and may or may not be around the center of the bearing clearance as it's shown in Figure 1. The size of both the orbits combines to create the overall orbit amplitude, which is also referred to as the overall vibration amplitude. When the rotation of non-synchronous orbit is in the same direction as the synchronous rotation it is called 'forward whirl'. If it is in the opposite direction it is called 'backward whirl'.

This is similar to the orbiting of the earth around the sun. The earth rotates at a frequency of approximately one revolution each day (N=6.9x$10^{-4}$ rpm's), which we will consider to be its synchronous speed. The earth also moves around a larger orbit, around the sun once a year. For every 365.25 revolutions the earth completes, 1 full orbit around the sun is completed (1.9x$10^{-6}$ rpm's). Therefore the earth is subsynchronously whirling and the whirl frequency ratio is 0.0027N (1.9x$10^{-6}$/6.9x$10^{-4}$). Since the earth whirls in the same rotational direction as its synchronous rotation, the subsynchronous orbit is considered to be forward whirling.

In the turbomachinery industry there are several different terms that can be used to refer to the same thing. The bulk of confusion comes from the general terms 'critical speed' and 'natural frequency'. There is of course variation in terminology across the industry so the definitions below are for clarifying the terminology used in this paper only.

- *Free-Free Natural Frequencies*. Refers to the natural frequency of the rotor, without consideration of the bearings and support structure. These frequencies can be experimentally measured and can be used to validate the analytical model of the rotor.
- *Undamped Critical Speeds on Infinitely Rigid Supports*, sometimes referred to as the Prohl Criticals. Refers to the natural frequencies of the system, excluding damping, when the rotor is rigidly fixed at the bearing locations. They are mainly used as a preliminary design parameter and do not directly predict the critical speeds.
- *Damped Natural Frequencies (DNF)*, also known as the Eigen frequencies, refers to the natural frequencies of the rotor-bearing system including damping effects. The damped natural frequencies are mainly a function of rotational speed, rotor stiffness, bearing damping and the support stiffness.
- *Critical Speed*, refers to the condition when the rotor speed coincides with a forward whirling DNF of the system and produces an increase in the synchronous response. This is also known as resonance.
- *Cross-Coupled Forces* are, in the simplest of terms, negative damping. Cross-Coupled stiffness is a force that acts tangential to the orbit, perpendicular to the principle stiffness and opposite to the damping, see Figure 1. These forces are a result of energy being 'fed' into the system and act to destabilize the rotor. Cross-coupled forces are generated in tight clearance gaps such as seals and impellers.

Harmonics refer to integer multiples of the synchronous speed such as 2N, 3N and so on. It is also important to note that the term 'synchronous speed' used throughout this paper is always in reference to the speed of the driven rotors unless otherwise noted. The terms rotor and pinion are used interchangeably. When beginning a SSV investigation it is suggested that the various parties clarify the terminology that is being used to avoid confusion.

## TYPICAL MACHINE CONFIGURATION

The primary components and features of IGC's will be briefly reviewed with a focus on how they relate to the topic of this paper. Figure 2 shows a diagram of a typical integrally

geared centrifugal compressor. In its most common configuration the central gear, known as the bullgear, drives one or several rotors at a higher speed based on each rotors gear ratio. Fixed speed electric motors are the most common driver as well as steam and gas turbines. Each rotor is driven by the bullgear via the helical gear mesh which is located between the two bearings. Each rotor can accommodate a single impeller outboard the bearings on both ends. This type of rotor is referred to as an overhung or cantilevered rotor design. The machine shown in Figure 2 has four stages on two rotors.

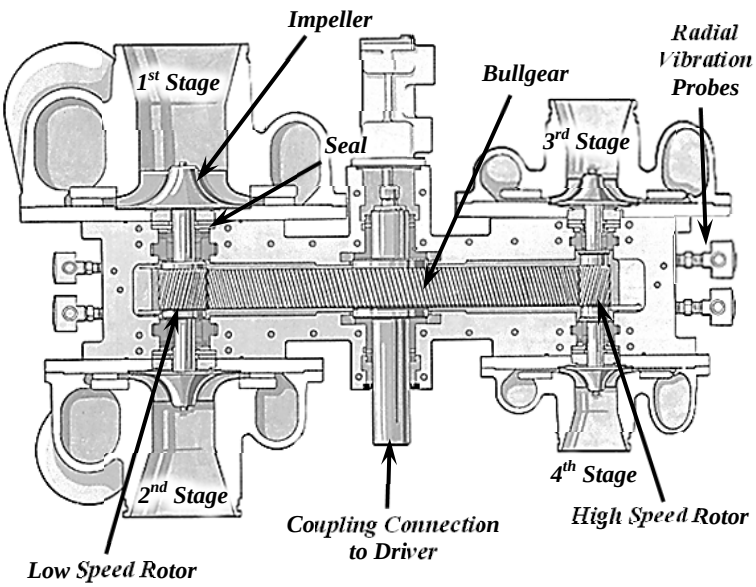



*Figure 2: Example of a Four Stage Configuration*

A single gearbox can also accommodate dual processes which can be controlled independent of each other. One or several stages can be used for booster compression applications in addition to the atmospheric inlet compression process.

There are three types of impellers used in IGC's; open, semi-open, and covered or shrouded. Regardless of the specific design of the impeller, due to the overhung design of the rotors, the impeller is almost always at the point of maximum deflection. Open impellers require a minimal clearance between the impeller blades and stationary shroud for aerodynamic performance purposes. The gas passage width of covered impellers is enclosed within the impeller. Covered impellers however require seals at the inducer, also known as the 'eye'. Figure 3 shows an open and closed impeller; semi-open is a combination of the two.

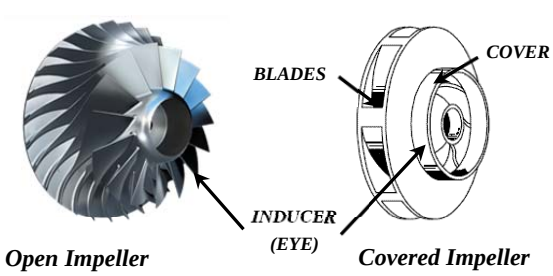



*Figure 3: Typical Impeller Designs*

Seals can also play an important role in the rotordynamic behavior of the machine. There are a variety of different seal designs available on IGC's depending on the OEM and application. Seals are located behind the impeller to prevent the high pressure gas from entering the gear box. Seals are also used to prevent oil migration from the gear box into the process flow. Typical seals used for air and nitrogen applications are labyrinth seals and carbon ring seals or a combination of the two. Dry gas face seals are used mainly in process gas applications. Labyrinth seals can play a significant role in rotor stability with high pressure applications because of their cross coupling effects. Dry gas face seals, brush seals and carbon ring seals are often considered to be dynamically neutral (API 684 2005) but the risk of rubs still exists in all seals.

The pinions typically utilize either a tilting or fixed pad thrust bearing or thrust collars (rider rings) to counteract the impeller and gear thrust. High speed thrust bearings and rider rings are both used extensively and each offer specific advantages. A typical cross section of a thrust collar design is shown in Figure 4. A ring on the pinion with a conical face contacts a similarly conical face on the bullgear. Thrust collars offer a much smaller contact area and therefore less mechanical losses versus traditional high speed thrust bearings. The gear thrust force is also consumed by the thrust collar and only the resultant aerodynamic thrust is seen by the bullgear thrust bearing. The downside is that they produce more radial and axial movement of the pinion. Tilting of the bullgear within its bearing clearance can cause a significant change in axial impeller clearance. The tilting of the bullgear also effects the angular alignment of the thrust collar surface.

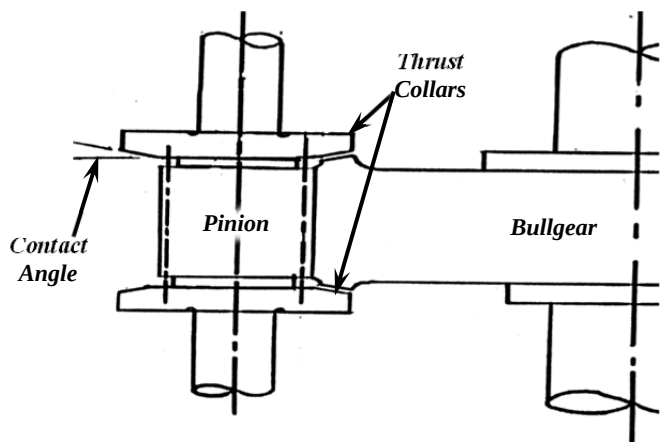
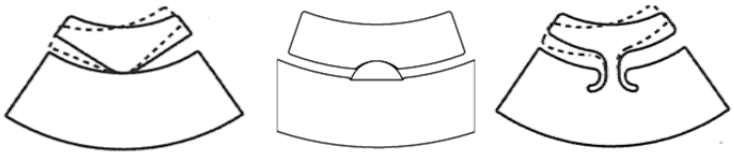



*Figure 4: Typical Rider Ring (Thrust Collar) Design*

With respect to the topic of this paper thrust bearings and thrust collars are not often considered in the rotordynamic model but should be considered in an investigation. Thrust collars particularly because the angle and nature of the contact surface couple lateral and axial vibration. Runout of the bullgear contact surface could cause an excitation source which is at a subsynchronous frequency relative to the pinion. The thrust collar will also affect the lateral response but this is an area for future research. High speed thrust bearings do contribute a moment stiffness that can have a strong effect on the first critical speed, whirl amplitude and speed of instability (Mittwollen 1991). Their effects are normally negligible but

can be more significant on rigid rotors with short bearing spans.

IGC's have typically utilized tilting pad journal bearings on the high speed rotors which have contributed to their high degree of stability. Tilting pad bearings contribute little to no cross coupling forces on the rotor which are the main source of destabilizing effects. There are several different types of tilting pad bearings that are used on IGC's including rocker type, spherical seat, Flexure Pivot™, and keyed (not shown).



*Rocker Pad* *Spherical Seat* *Flexure Pivot™*

*Figure 5: Common Types of Tilting Pad Bearing Designs*

Figure 5 shows the bearing pad centered on the pivot point. When the pad is not centered on the pivot it is referred to as an offset pivot design. The pivot is moved away from the leading edge of the pad by a certain fraction of the pad arc length, usually 0.55-0.65. There are several benefits to an offset pivot design but with regards to SSV an offset pivot design has been shown to reduce the sensitivity of the system to bearing clearance variation (Chen et al. 1994). While there can be negative effects of an offset pivot design they are often used in bearings with high unit loads or surface velocities to reduce the pad temperature.

Rocker pads create a line contact against the housing and allow the pad to rotate circumferentially. The Flexure Pivot™ bearing is similar to the rocker bearing in that it rotates circumferentially but it is machined from one piece of metal which reduces the tolerance stack up. The spherical seat type bearing uses a ball and socket design which creates a point or spherical contact and allows the pad to rotate in all directions. The spherical seat bearing can also become locked on its seat and act similar to a sleeve bearing which can lead to SSV (Vance et al. 2010).

An unbalance of forces on each side of the rotor can cause angular misalignment of the shaft and pad. If the pad is not aligned to the shaft edge loading will occur. Edge loading causes a decrease in the effective width of the bearing which reduces its damping. Edge loading can also lead to dry friction if the oil film becomes too thin. A bearings capability to maintain its designed clearance and preload value is also critical. An increase in assembled clearance decreases the preload which results in a lower stiffness. Causes of SSV such as oil whip and oil whirl will not be discussed in great detail because most IGC's utilize tilting pad bearings which are not susceptible to these phenomena when functioning correctly.

## DYNAMICS OF AN OVERHUNG ROTOR

As mentioned earlier, the driven rotors of IGC's are commonly referred to as an overhung or cantilevered design. This type of rotor has a different dynamic response than a between bearing rotor, particularly the mode shapes. One of the key factors affecting what the mode shapes are is the stiffness of the rotor relative to the stiffness of the bearings. The first two mode shapes of any shaft are 'rigid body' mode shapes. The rotors of IGC's are generally stiffer relative to the bearing stiffness. Therefore the first mode shape is typically a 'rigid rotor' mode shape. Regardless of the design the first true bending mode is always the third mode. Figure 7 shows an example of an undamped natural frequency map and the mode shapes associated with a double overhung rotor. In this example the left side of the rotor has more overhung weight then the right side. Although the mode shapes labeled 'D' and 'E' in Figure 7 does have some curvature to it, it is not a true bending mode.

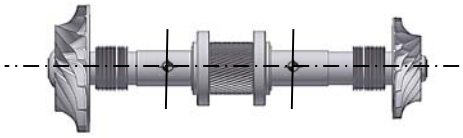


*Figure 6: Example of a Double Overhung IGC Rotor*

It is a general assumption that the first mode shape of a stiff rotor is a rigid cylindrical (bouncing) mode. This is not always the case. It is quite common that the first mode shape is a rigid conical shape (see 'A' in Figure 7) and the second a rigid cylindrical (see 'B' in Figure 7). In this example mode shape 'C' would be the first bending mode. Each of the mode lines reach a maximum critical speed value which would not change if the support stiffness was increased further. These values are referred to as the undamped critical speed on infinitely rigid supports or Prohl Critical's. The flexible mode shapes associated with high bearing stiffness values are not shown because they very dependent on the specifics of the design.

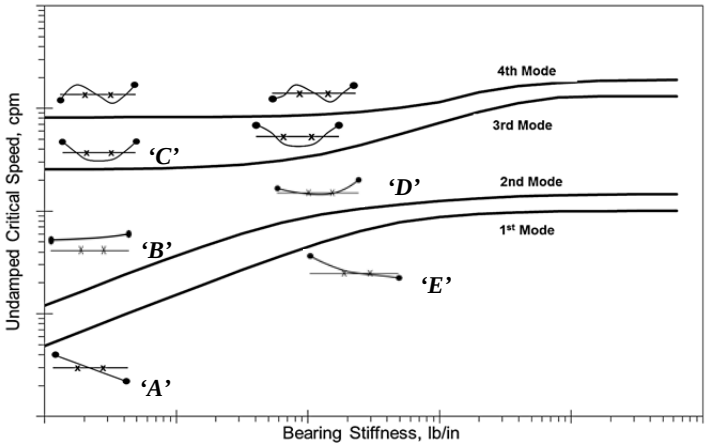



*Figure 7: Example of Undamped Critical Speed Map for a Double Overhung Rotor with undamped mode shapes*

One of the most critical aspects of accurately calculating the natural frequencies of overhung rotors is accurately modeling the overhung geometry. Since the impellers are cantilevered they have a significant impact on the stiffness of the rotor and therefore the natural frequencies as well. Three dimensional CAD software can be used to predict the mass and inertia but there will be some small variation in production. The accuracy of the part and model can be verified by experimentally measuring the free-free natural frequencies. This is done by measuring the response of the rotor while it is hung freely and impacted with a known force. An accelerometer is used to measure the response and a specialized

impact hammer is used to measure the force. Vazquez et al. (2012) provide a detailed guide of this method.

Another assumption made with between bearing compressors is that decreasing the bearing span will increase the rotor stiffness. This is not always the case with a double overhung rotor though. Depending on the original bearing span and the rotor design, decreasing the bearing span can descrease the rotor stiffness, such as with the design shown in Figure 6.

### *Radial Load and Load Angle Variation*

The stiffness and damping characteristics of the bearings are in part a function of the radial load applied to them. Due to the large variation in aerodynamic loading achievable with IGC's there can also be a fairly large variation in the DNF's. In addition to transmitting the torque, helical gears also generate an axial and radial force on both the bullgear and pinion. Figure 8 shows the orientation of the gear loading applied on each pinion for a given direction of rotation. Depending on the tooth geometry the radial gear load will typically be at a 20 degree angle from the vertical, known as the pressure angle. With a centrally located bull gear, one of the pinions will be 'upward driven' and the other 'downward driven'. The gear load and the weight of the rotor are oriented approximately 20 degrees apart on a downward driven pinion. On the other hand, upward driven pinions have the gear load and weight vectors oriented approximately 160 degrees apart. This creates a larger variation in the resultant load angle in off design conditions. The downward driven pinion has shown to be more predictable in part because of a reduced variation in the load angle.

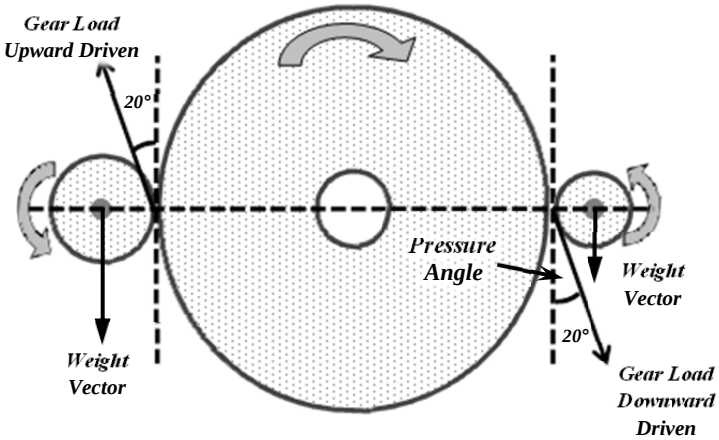



*Figure 8: 2D Diagram of Gear Load*

Operating in off design conditions, such as reduced load can unload the bearings of the upward driven pinion. Another factor that should be considered is the effect of reduced load shop testing. Sometimes OEM's will not have the capability to perform a full power or pressure test at their facility and will operate at a reduced load. This change in load will affect the rotordynamics and should be considered.

## TESTING VARIABLES

IGC's have features that allow the variation of several parameters that may be contributing to the SSV. Observing the changes to the response that occurs as a result of the variation can be a very insightful tool. The aerodynamic load can be varied using the inlet valve and/or the discharge valve. Observing the effect of varying pressure and flow should be considered. The oil temperature and pressure can also be varied without major modifications. Decreasing the oil temperature or increasing the oil flow will increase the damping provided. When performing these variation tests it is important to remember forces radial to the shaft, such as direct stiffness and cross-coupled damping, mainly affect the natural frequencies and critical speeds but have little effect on stability. On the other hand forces tangential to the orbit whirl, such as cross-coupled stiffness and direct damping, affect the stability and amplification factor but have little effect on the natural frequencies (Vance et al. 2010).

The gearcase lubrication design does not always allow the oil flow or temperature to be varied for individual bearings. Changes to the oil pressure and temperature are often seen by all of the bearings as well as the gear mesh spray, not just the rotor being investigated. The oil pressure is set by a pressure control valve upstream all of the bearings. Entrance effects, contractions, poor machining, and sharp 90 degree angles in the oil flow path can cause the oil pressure to be much lower at the bearing entrance than at the pressure regulating valve. This results in less flow to the bearings and gear mesh. Measuring the oil pressure at the bearing to confirm adequate flow is ideal.

## DATA INTERPRETAION TECHNIQUES

Taking the time to set up the proper instrumentation and software to observe the response in the time and frequency domains is fundamental to the investigation. This can ultimately save time and produce a practical long term solution. Having the proper tools available to record and view the data is an obvious requirement in this process. The data shown in this paper was collected with a multi-channel dynamic signal analyzer and processed and filtered in a program made for monitoring the vibration of rotating machinery.

One of the first steps in addressing a SSV issue is identifying the vibration frequency. One should also establish the frequencies that are associated with possible sources of excitation. These frequencies include but are not limited to the following:

- The lateral DNF's of the rotor bearing system
- The speed of the bullgear as well as its harmonics below the rotor speed
- Accompanying rotor speeds
- Electrical frequencies (50 or 60Hz.)
- Torsional natural frequencies
- Support structure natural frequency

During an investigation it is recommended that each stage be instrumented with X-Y radial proximity probes which are oriented 90deg from each other. The location and orientation of the probes are shown in Figure 1 and 2. It is very important that the actual orientation of the probes be correctly configured in the vibration measurement software. Confusing the X and Y probes can lead to false conclusions and fixes that only make things worse. A key phasor can be mounted radial to the pinion and generate a signal pulse every time a notch or dimple on the pinion moves past the probe. This signal is used to measure the rotational speed and phase angle. An example of a key phasor

configuration and signal is shown in Figure 9. Key phasors and X-Y probes are needed to generate the orbit plot and shaft center plot. X-Y probes for each overhung impeller and key phasors for each rotor are suggested on machines with untested components and operating ranges.

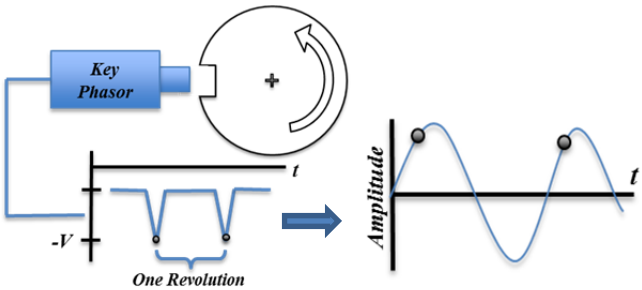



*Figure 9: Key Phasor Configuration and Signal*

Figure 10 shows an example of a spectrum plot from a single probe showing the synchronous response, second harmonic and the transmitted response from the bullgear.

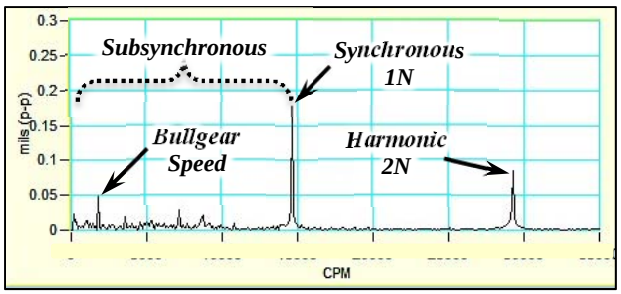



*Figure 10: Spectrum Plot from a Single Probe*

Figure 11 shows the time wave form plot of a single probe for a case where the SSV is larger than the synchronous and dominates the overall response. The overall signal is filtered to show the SSV component and the 1N component. Each dot represents a key phasor pulse which represents one shaft revolution.

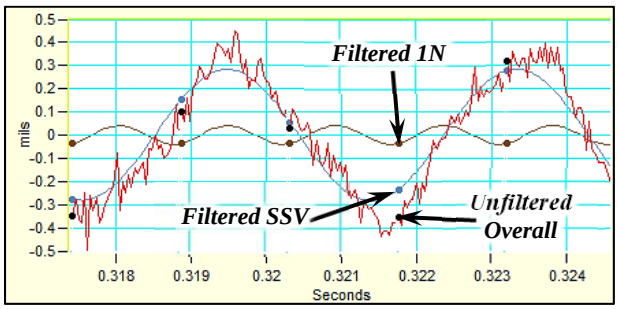



*Figure 11: Time wave Form Plot showing Unfiltered, Filtered SSV and Filtered Synchronous of a Single Probe*

Knowing the orientation of the X and Y probes for a particular stage one can determine whether the whirl is forward or backward. For example, in Figure 1, the shaft is rotating CCW. Therefore the X probe should see the peak vibration first and then the Y probe, this would indicate forward whirl.

The orbit plot filtered at the subsynchronous frequency is also a useful tool. As described by Kar and Vance (2007), the filtered orbit shape can help indicate whether the subsynchronous response is due to instability or a forced response, although this is not an absolute indicator. Extremely elliptical orbits cannot be caused by instability. By observing similarly oriented probes on both ends of the rotor one can determine if each side is whirling in phase or out of phase with each other. This gives an indication of the whirl mode shape when orbit plots are not available.

## CAUSES OF SUBSYNCHRONOUS VIBRATION

### ROTORDYNAMIC INSTABILITY

Rotordynamic instability is also referred to as 'self-excited' vibration. Self-excited vibration is often initiated by a disturbance or force acting on or in the system which disturbs its equilibrium. The resulting response is motion (vibration) that is further created and sustained by the feedback of the motion itself. In other words, the initial displacement creates new forces or force unbalances which act on the rotor and in turn causes more motion.

The stability of a system is based on its ability to return to equilibrium after an initial disturbance. This stability is quantified by the logarithmic decrement (log dec). Since damping is the force that acts to suppress motion the log dec is analogous to the amount of damping present. The destabilizing forces that contribute to instability are cross-coupled forces. The cross-coupling forces are related to the shafts eccentricity in the tight clearance area. Positive log dec values indicate a stable system and negative values indicate an unstable one. Each DNF has a log dec associated with it. A log dec value greater than 0.1 is required to satisfy the stability criteria of API 617.

Rotordynamic instability is characterized by sudden excitation of a DNF of the system. The frequency of the excitation should not track with running speed. Observing the running speed as it passes through the excited frequency range should reveal a critical speed if the system isn't critically damped. Figure 12 shows an example of a rotordynamic instability characterized by a sudden excitation at the first DNF. The SSV dropped off almost immediately after shut down and the 1N clearly goes through a critical at the same frequency on coast down. Another characteristic of instability is that the orbits should be very similar to the predicted orbit shape at the subsynchronous frequency, which is usually circular or very slightly elliptical. The unfiltered orbit plot during this excitation, shown in Figure 13, is very circular and is dominated by the SSV response.

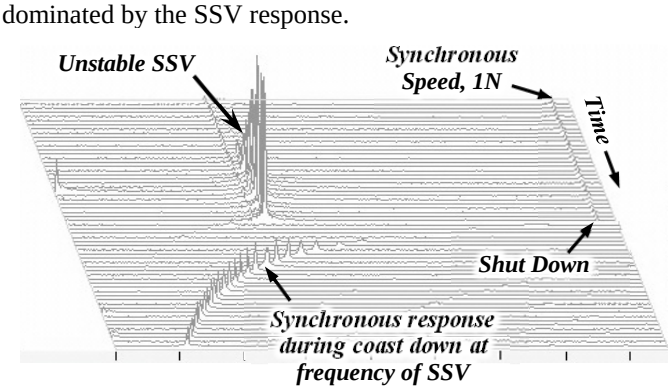



*Figure 12: Rotordynamic Instability Excitation w/ 1N critical confirmation.*

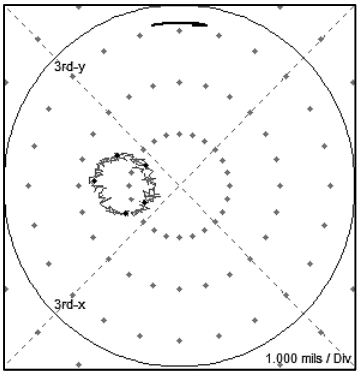



*Figure 13: Unfiltered Circular Orbit During Unstable SSV*

The case shown in Figure 12 and 13 is a good example where multiple root causes contributed to the subsynchronous response. In this case the rotor which experienced the SSV has a first DNF with less than 4% separation from the synchronous speed of the accompanying rotor. Although the first DNF was predicted to be very stable, it was found that the bearing housing was 0.004" larger than the acceptable which caused them to be loose. This looseness combined with an external excitation source near the first DNF cause the rotor to become unstable. Once proper crush was obtained this instability was eliminated. While the root cause of this SSV was not rotordynamic instability, the response was classic instability.

Further discussion of instability can be found in API 684 (2005) and in the majority of publications referenced in this paper. The mechanism that will be focused on in this paper is aerodynamically induced instability produced by flow through tight clearance gaps.

### *Aerodynamically Induced Instability*

The origins of today's rotordynamic stability criteria, as defined by API Standard 617 (2002), date back to the 1970's when several famous centrifugal compressors experienced high SSV whose frequency coincided with the first DNF. These unstable vibrations were a result of cross-coupled forces generated in the tight clearance gaps of labyrinth seals. Fluid flowing through tight clearance gaps such as in labyrinth seals will tend to whirl in the direction of rotation but at a slower rotational speed. Since the fluid will have a whirl velocity of zero at the stator component and a velocity equal to the rotational speed the rotor surface than the mean whirl velocity will be slightly less than 0.5 the rotational velocity. The fluid will typically already be whirling before it enters the seal, which is called pre-swirl. Pre-swirl can have a strong influence on the stability of a rotor. The higher the pre-swirl is the larger the cross-coupling effects are. This is why a popular solution to this problem is to incorporate a de-swirling device or design to decrease the pre-swirl.

This fluid behavior is similar to that of oil whirl in fluid film sleeve bearings. With oil whirl, the fluid forces are strong enough to create a subsynchronous response at 40-49% of the running speed. Fluid such as air does not have as strong of an effect as oil. Its dynamic effect does however increase with density and can be strong enough to excite a DNF of the rotor if one exists between 40-49% of running speed. Therefore instability will not track with running speed like oil whirl will but instead stay at the DNF. In this sense it is similar to oil whip because it will 'lock on' a particular frequency.

Gruntfest et al. (2001) presents one of the first documented rotordynamic instability issues experienced on an IGC. In that particular case study the rotor being investigated was a double overhung rotor with closed impellers on each side. Labyrinth seals were located at the impeller eye and behind each of the closed impellers. A rotor DNF was at approximately 47% of the operating speed and was becoming excited. During the investigation the variation of oil supply temperature was used to identify the SSV's sensitivity to damping. Cooler oil provides more damping. Because the SSV was suppressed by decreasing the oil temperature it was clear that additional damping or less cross-coupled stiffness would yield a stable system. "Swirl breaks" were incorporated into the labyrinth seals at the eyes of the impellers which reduced the pre-swirl and therefore decreased the cross-coupled stiffness. Gruntfest et al. points out that labyrinth eye seals at the eye of the impeller are at the optimal location to excite cantilevered modes of the rotor. This is similar to between-bearing compressors in that close clearance areas at the locations of maximum deflection of a mode shape are more likely to excite that particular mode.

Accounting for cross-couplings not previously considered in the analysis will usually increase the predicted DNF slightly, and decrease the log dec. This is important to keep in mind during the initial investigation because if the SSV frequency is lower than the first predicted DNF, it is unlikely that the inclusion of additional cross-coupled stiffness in the analysis will make the predicted frequency match the actual response frequency. Other factors must be considered in this case, such as structural stiffness.

There are several ways to reduce the cross-coupling effects or increase the stability of a rotor. Increasing the stiffness or natural frequency will improve the stability. Increasing the stiffness can be done by increasing the bearing span (to a certain degree), increasing the bearing diameter and decreasing the overhung weight. These are not always the most practical solutions though. Decreasing the bearing clearance will increase the direct damping and stiffness but will also increase the pad temperature and power loss. As mentioned earlier, cross-coupling forces are related to eccentricity, particularly in the seals or impellers. Decreasing the eccentricity will decrease the destabilizing force. Decreasing the tangential velocity of the gas in the seals will also help to stabilize an unstable rotor. This can be done with swirl brakes, shunt holes or simply by applying seal buffer gas to reduce the tangential velocity.

## LOOSENESS

Mechanical looseness of the rotor-bearing support structure is one of the more difficult causes to diagnose because its signature can be almost unexplainable or similar to other causes. Because of this, and the fact that it can be a relatively simple problem to solve, this is a good thing to check for early on if the root cause is not apparent. SSV caused by looseness of the bearings in their housing has traditionally been considered to occur at integer harmonics, sub harmonics, and their integer

multiples (2N, 3N, N/2, N/3, 2N/3, 4N/3…) (Adams 2001). Rajagopalan and Vance (2007) have shown experimentally with sleeve bearings that a very small amount of clearance can produce a subsynchronous response at the damped natural frequency that does not track.

Looseness is very unpredictable and isn't necessarily limited to the frequencies discussed above. The authors have seen loose bearings excite 1N amplitudes, the first DNF, as well as a broad band low frequency response. If multiple subsynchronous frequencies are seen during a particular test, random peaks come in and out, or if the subsynchronous frequency is not repeatable then loose bearings should be suspected. The shaft center and orbit plot can be very helpful tools because the orbit plot tends to be very elliptical and directionally biased. The shaft center can also suddenly 'jump'. Figure 14 shows an example of the waterfall plot when a bearing suddenly became loose. Figure 15 shows the shaft center plot that resulted from the loose bearing.

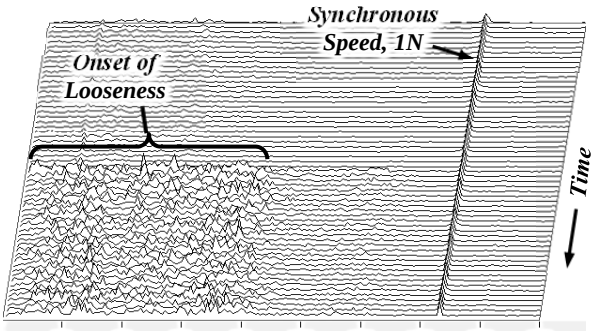



*Figure 14: Sudden change in response due to loose bearing*

Many times the effects of a loose bearing can be seen right after start up and decrease as the machine heats up because the bearings grows more thermally than the housing. Other times it may suddenly become loose as the load increases or as the bearings heat up. The surface finish and roundness of the housing may be contributing in this case.

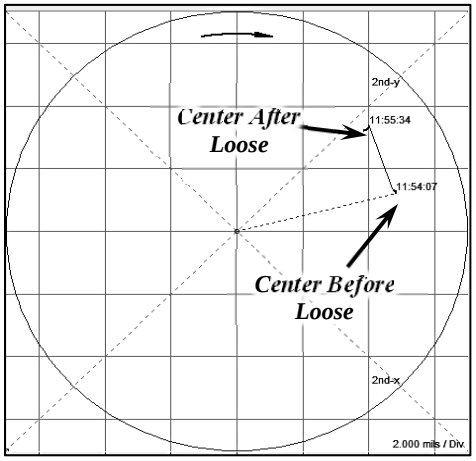



*Figure 15: Change in Shaft Center Due to Loose Bearing*

Two causes of looseness are excessive sealant on the split line or out of tolerance parts. The obvious solution to the latter is to replace the part that is out of tolerance. However, this solution can be very time consuming if spare bearings are not on hand or, in the worst case, the gearbox needs to be replaced. An alternative solution is to apply a shim between the bearing and its housing to remove the looseness. This is not a universally accepted practice because it is often incorrectly executed. The first challenge is determining the amount of looseness that exists. One common method is to place fuse or lead wire on top of the bearing OD and shim stock of a slightly smaller thickness on the split line. Put the top half of the gear box back on and tighten the split line bolts with the shim stock still between the two halves. After the top half of the gear box is removed the crushed thickness of the wire can be measured. Subtract the shim stock thickness from the measured wire thickness to determine how loose the bearing is. This practice is discussed in further detail by Zeidan and Paquette (1994). It is difficult to obtain repeatable results if this isn't done correctly. It is common to glue the shim to the gearbox but it is important not to apply the glue in an area that would increase the overall thickness of the shim.

## INSUFFICIENT LUBRICATION (DRY FRICTION)

The contact of a rotating component with a stator creates a friction force. The friction force is always opposite the direction of rotation and therefore creates a vibration component that is backward. Dry friction can occur as whirl or whip. The main difference between whirl and whip, similar to oil whirl/whip, is that whirl tracks at a fraction of running speed while whip 'locks on' to a DNF and does not track. Unlike a momentary rub, dry friction whirl is more likely to be from a light continuous annular rub. The dry friction can occur in seals, impellers and bearings. Dry friction is the only known cause of backward whirling SSV currently (Rajagopalan and Vance 2007).

The following is an example of SSV that was caused by insufficient lubrication in the journal pads. A double overhung rotor was supported with flexure pivot journal bearings with labyrinth seals outboard each bearing and thrust collars to absorb the axial thrust. The shop testing used an electric motor and a VFD with the motor operating at 1,490rpm. The low speed rotor's operating speed was 14,557rpm. The bearing unit load was 170psi with a journal velocity of 319ft/s. The original rotordynamic analysis indicated a very stable rotor with a first DNF of 0.3N and a log decrement of 0.7. The two stages on this rotor were the first two stages of the compressor therefore had relatively low gas density. During the shop testing SSV occurred at 0.22N.

Electrical interference was the first cause investigated during the test since it was very close to 50Hz, which was the motor electrical frequency. The SSV also had relatively high amplitude and no audible indication of such high vibration. After further testing it was found that the SSV did not appear until the oil temperature reached approximately 100°F. Testing also revealed that increasing the oil pressure 10-15psig would suppress or eliminate the SSV. These changes indicated that there was either inadequate oil flow or inadequate damping in the rotor bearing system.

The SSV did not become unstable and continuously increase. Rather it reached a 'limit cycle' where its amplitude became stable. This stable amplitude was less than the bearing clearance therefore it was suspected that the increased vibration

amplitude caused an increase in damping which stabilized the system. Although decreasing the oil temperature or increasing the pressure decreased the SSV to acceptable levels they were not considered to be a long term solution since the root cause was still unknown. Once power to the motor was disconnected the SSV instantly disappeared and did not track with 1N during coast down.

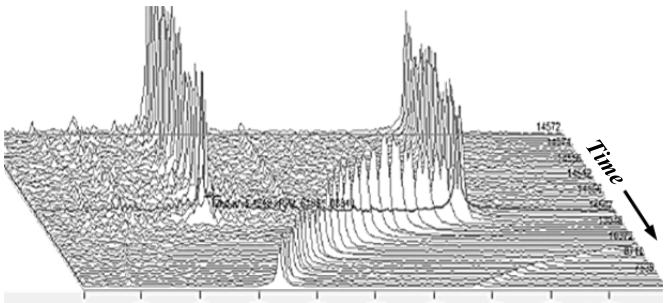



*Figure 16: Waterfall plot during shut down and coast down*

It is difficult to make any conclusions based on the response of the SSV when the load is removed at the same time the speed is decreased. If SSV tracks the running speed at a fixed fraction *with load* it could be an indication that the cause is forced vibration while non-tracking SSV indicates instability or resonance but this is not an absolute indicator. Even though IGC's are fixed speed machines the OEM typically uses a VFD during shop testing which can be used to vary the speed. In order to determine if the SSV tracked with load the pinion speed was decreased 1,000rpm, which eliminated the SSV as shown in Figure 17. The SSV returned immediately after the speed was increased back to the normal running speed.

Next the filtered and unfiltered time wave form and orbit plots were considered in order to determine the whirl direction and orbit shape. The filtered orbit plots were elliptical and the filtered time wave form showed that the SSV vibration was backward whirling. Based on this observation it was concluded that dry friction within the bearing was the cause.

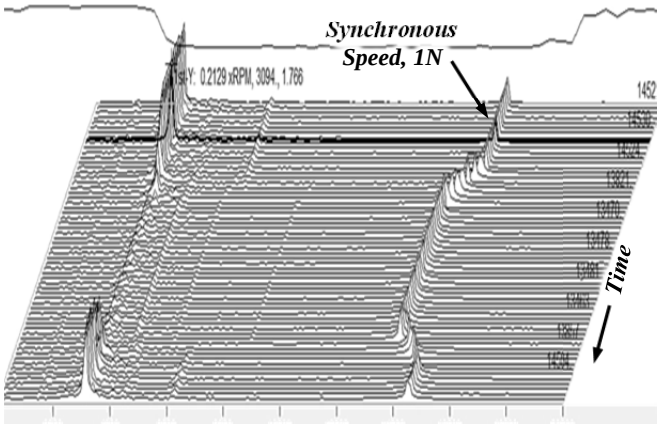



*Figure 17: SSV response to decreasing the rotor speed 1,000rpm*

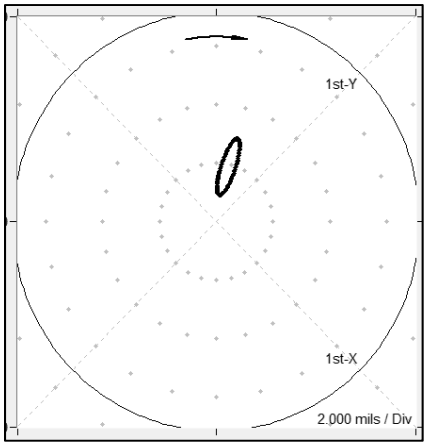



*Figure 18: Orbit plot filtered at the SSV frequency*

In order to determine if the bearings were receiving the recommended flow, the oil pressure at the bearing was measured. The pressure was found to be 6psig lower than the design pressure during the initial testing as was corrected. Bearing predictions showed that an adequate film thickness was still expected even with the reduced flow. Finally the angular tilting of the rotor was considered in the bearing predictions. It was suspected that dry friction was occurring at a localized area of the pads. The final solution was to decrease the clearance of the bearing in order to decrease the angular tilting of the rotor. Decreasing the clearance also increased the effective damping from the bearing. Shaft center plots from before and after this change showed that the eccentricity of the shaft within the bearing was decreased and uniform loading of the pad across the width was achieved.

## TRANSMITTED VIBRATION

The transmission of vibrations across mating gears is a phenomena found in most gearbox applications. Transmitted vibration is a special case because it is not always the root cause of the vibration, just the method in which the vibration is transmitted. Since it is common to only monitor the lateral vibration of the driven pinions, many vibration issues are only discovered if they affect the pinion even though they may not necessarily originate from the pinion.

### *Bullgear Misalignment & Unbalance*

Bullgear misalignment or unbalance is one of the more common causes of SSV in IGC's and isn't considered harmful at acceptable amplitudes. The excitation might not necessarily cause the rotor to become 'unstable' but instead cause the overall vibration amplitudes to exceed acceptable limits. Since the bullgear operates at a speed lower than the pinions, the bullgear synchronous speed is at subsynchronous frequency relative to the pinion. Therefore synchronous vibration and harmonics of the bullgear would show on the pinions vibration spectrum as being subsynchronous at the bullgears rotational frequency.

The most common source of misalignment in IGC's is alignment of the bullgear and the motor shaft, connected via a flexible coupling. If a SSV peak exists at 1N, 2N and even 3N of the bullgear speed, the cause is possibly bullgear shaft

misalignment to the motor. A response at only 1N of the bullgear is more likely to be due to unbalance though. Thermal effects play a significant role in shaft misalignment because the gearbox tends to get hotter than the motor during normal operation. Therefore it is common to align the bullgear shaft to be lower than the motor shaft in cold conditions. Obviously this effect can be more apparent on larger machines. Multiples of driver speed may be seen during start up and while the machine is heating up. The amplitudes of the peaks should decrease as steady state thermal conditions are reached with proper alignment. Improperly seated bullgear bearings and shaft angular deviation also cause misalignment and produce similar symptoms. Lateral vibration originating from the bullgear can also excite a DNF of a rotor. Decreasing the vibration of the bullgear would eliminate the source of the excitation.

### *Pinion To Pinion Crosstalk*

Similar to the case above where synchronous and harmonic lateral vibrations of the bullgear are transmitted to the pinion, the vibration of one pinion can be transmitted to other pinions. Crosstalk is where the vibration energy from one rotor is transmitted to the other rotor (Smith 2011). Smith describes a case with an IGC where the high speed pinion had a DNF very close to the operating speed of the low speed pinion. Synchronous vibration of the low speed pinion was transmitted through the bullgear and excited the high speed pinions DNF causing SSV which lead to bearing wear. It was discovered that excessive pitchline and thrust collar runout on the low speed pinion contributed to transmitting the vibration energy across the bullgear to the other pinion. This is a good example of how the vibration of one pinion can act as a source of vibration on another. The authors have also seen this phenomena occur.

The lateral vibration transmissibility of a gear train is dependent on the design of the pinion and bullgear geometry (inertia), gear teeth geometry, support stiffness and damping provided by all of the bearings. Based on the authors' experience, synchronous lateral vibration of one pinion is more likely to transmit through the bullgear to the accompanying pinion than a subsynchronous vibration of equal or greater amplitude.

Figure 19 shows an example of just such a case. The top graph shows the waterfall plot of the low speed rotor with the synchronous speed labeled LSR 1N. The second graph is the waterfall plot for the high speed rotor with its synchronous speed labeled HSR 1N. The synchronous vibration of the low speed rotor can be seen in the high speed rotors spectrum, which is crosstalk. More interestingly the unstable SSV of the low speed rotor, which is much larger than the synchronous, does not transmit to the high speed rotor. The unstable SSV of the low speed rotor was not caused by crosstalk. This is a clear indication that the transmissibility of lateral vibration is frequency dependent.

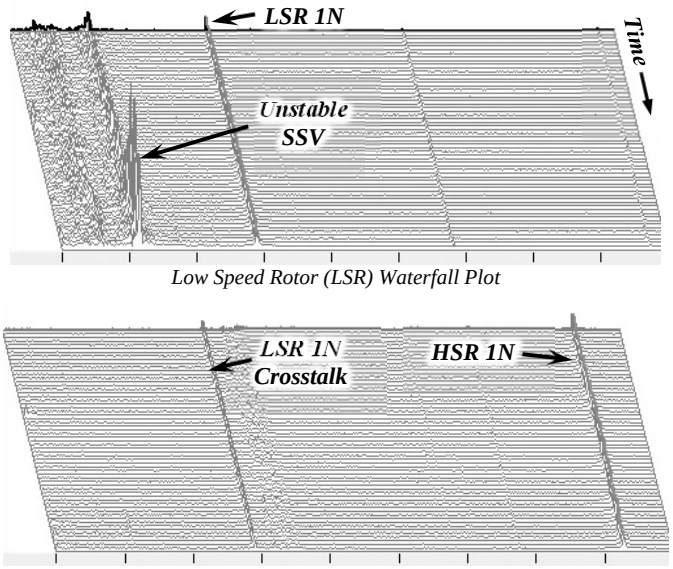



*Figure 19: Lateral Vibration Transmission*

### *Lateral and Torsional Vibration Interaction*

The study of how torsional and lateral vibrations interact in various types of turbomachinery is a thoroughly researched topic but is also an area of continual development with regards to the analytical tools available. Ample publications exist covering the interaction of lateral and torsional vibration with geared applications so the topic will only be covered briefly.

Currently it is common practice to analyze lateral and torsional response separately but methods and software exist to analyze the response of a rotor system with coupled lateral and torsional vibration. Experience has shown that the coupling of these vibrations can lead to possible excitation or increased stable amplitudes. In the case of IGC's with fixed speed operation, new designs should avoid the intersection of rotational speeds with torsional natural frequencies (TNF) of the gear train when possible.

Known sources of torsional excitation include, but are not limited to, lateral vibration of any of the rotating shafts, torque fluctuations from motors during start up, variable frequency drives (VFD's), electrical line frequency resonance and, although rare, aerodynamic flow excitations. Torsional issues become more of a concern with long strings of machinery coupled together. They also tend to affect all of the coupled machines instead of just one machine. When torsional vibration is suspected keep in mind that torsional natural frequencies are not dependent on the rotating speed unlike lateral natural frequencies. Torsional excitation usually does not become suddenly 'unstable' like lateral instabilities as well (Vance et al. 2010). Resonant interference can usually be avoided simply by 'tuning' the TNF with the main driver coupling stiffness.

When TNF's are excited they cause speed fluctuation about the mean operating speed with the magnitude of the fluctuation being equal to the frequency of the torsional natural frequency being excited. This effect can be detected by observing the side bands around the gear mesh frequency response. If a TNF is being excited the distance between the

mean response and the side band would be equal to the torsional natural frequency being excited. This is the simplest method to determine if a torsional excitation exists but it can be challenging to do on IGC's because the gear mesh frequency is often greater than the frequency range of the proximity probes.

ADDITIONAL CAUSES OF SSV

There are multiple other causes of SSV in IGC's that are worth noting.

- Internal Friction

  Internal friction is a rotordynamic instability that is caused by loose fit components on the rotor. Interference fit connections, such as on impellers or thrust collars, can become loose during operation and cause internal friction. The looseness can be very slight and only in local areas of the fit and still have an effect. This is partially why some interference fits are relieved in areas that will grow apart during operation.

- Aerodynamic Stall Cells; rotating stall and static stall.

  Stall is an aerodynamic flow phenomena characterized by localized flow recirculation created by large angles of incidence between the fluid flow and a surface. A localized area of recirculation is referred to as a stall cell. As the volume flow of the compressor moves away from the design condition the angle of the flow exiting the impeller changes. This is why stall is usually associated with off design conditions, particularly low flow conditions.

- Turbulent Flow Distortion.

  Sharp bends in the interstage piping and inlet guide vanes can cause turbulent flow at the inlet of a stage. Srinivasan (2012) presents such a case which was resolved by using a flow straightening device.

- Subsynchronous 'Hash' (see DeCamillio, 2008)

  'Hash' is vibration characterized by low frequency, low amplitude, broadband subsynchronous vibrations that fluctuate randomly. This response is more common on evacuated, direct lubrication bearings but can occur in flooded bearings as well. It was determined that the low frequency, low amplitude hash was caused by pad vibration.

- Structural resonance
- Forced vibration from neighboring machinery

There are also misleading sources of SSV that can also cause false readings such as vibration probe resonance and electrical interference.

## SUMMARY

Several causes of subsynchronous vibration have been reviewed specific to integrally geared compressors. The general design of the rotors in IGC's creates a response unlike between bearing single shaft centrifugal compressors and must be considered. Vibration monitoring techniques are reviewed with a focus on how they can be used to identify common symptoms of each cause. A systematic approach should be taken when making changes to solve SSV. It is important to keep in mind that the source of SSV may not be from a single root cause and can be from a combination of several. Several examples were presented which illustrate this point. Caution should be taken when reading any paper that offers generalities with machine vibration because each individual case is often unique. If a systematic approach is not taken up front the time it takes to solve the problem could grow exponentially or return in the future.

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## ACKNOWLEDGEMENTS

The authors would like to thank FS-Elliott for providing the resources and encouragement to write this paper. Great thanks to all the authors throughout the years that took the time to publish their research and experiences so that we may all continue to mutually benefit. The authors would particularly like to thank Dr. Brian Murphy for his expertise and guidance over the years.