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Applying API 617, 8th Edition to Expander-Compressors with Active Magnetic Bearings

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ABSTRACT

High-speed expander-compressors are commonplace in the gas-processing industry. In recent years, expander-compressors equipped with Active Magnetic Bearings (AMBs) have gained wide acceptance and are now the norm in ethylene plant refrigeration turboexpander applications and increasingly specified for gas-liquid separation. Further improvements in magnetic bearings are simplifying the technology for those responsible to purchase, commission and maintain rotating machines. As well, it is becoming easier to verify that AMB machines comply with accepted design standards.
The Eighth Edition of API617, released in September 2014, includes a new annex (Part 1, Annex E) and other material, specifically addressing AMB-equipped machinery. Unlike previous editions which included only informative material, this new material provides a detailed criteria by which AMB designers, purchasers and users can evaluate API compliance or AMB-equipped machines. These design, analysis and test specifications reduce the technical complexity of AMB technology for expander-compressor stakeholders, and contribute to continued growth in the market place.
This tutorial will discuss and demonstrate the application of API617, 8th Edition to AMB-equipped expander-compressors from a practical, user-oriented point of view.

INTRODUCTION

Previous authors have presented excellent and detailed descriptions of high-speed expander-compressors (E-Cs) (Jumonville, 2010). Other authors (Swanson et al, 2014) have provided a comprehensive description of general AMB theory, and the design, analysis and testing requirements specified in API 617, Eighth Edition. The goal of this tutorial is to contribute a practical, “hands-on” application of criteria in the new API specification, as applied specifically to expander-compressors. This tutorial contains recommendations and observations based on experience gained through the design, analysis and commissioning of many AMB expander-compressors, both prior to and after the publication of the latest API617 edition.

The live tutorial session includes the demonstration of many of the AMB testing and operational techniques discussed herein on a small, portable AMB test stand. The test stand consists of a 20,000 RPM high-speed spindle with a small, commercial magnetic bearing controller. While this system is a much smaller brother to the expander-compressors discussed herein, the principles of control, stability, clearance checks, etc., are the same for all AMB systems.

For the remainder of this paper, all references to API617 will be to the Eighth Edition, unless otherwise indicated.

This tutorial assumes an audience already fully familiar with expander-compressors, and with industrial AMB technology. Notwithstanding, a very brief introduction to AMB expander-compressor applications will be provided, along with a description of the advantages of AMB technology in these applications.

INTRODUCTION TO AMB EXPANDER-COMPRESSORS

An expander-compressor, also referred to as a turboexpander-compressor or simply a turboexpander, as discussed in this paper, refers to a machine with a common shaft, with a centrifugal expander wheel on one end, expanding gas with a temperature of less than 300° C (570°F), and a centrifugal compressor wheel on the opposite end. Table 1 provides typical characteristics of AMB expander-compressors used in gas processing facilities today. Over 550 AMB expander-compressors are installed and in service as of 2015, in gas processing facilities around the world. A typical expander-compressor is shown in cross section in Figure 1. An operational AMB expander-compressor is shown in Figure 2.

Table 1 Typical AMB Expander-Compressor Characteristics

Machine Characteristic	Typical Values
Speed	6 – 70 kRPM
Power	0.2 – 14 MW (.27 – 18.7 kHP)
Radial AMB diameter	50 – 240 mm (2.0 – 9.5”)
Shaft mass	4 – 600 kg (9 – 1320 lb)
Number in Operation	>550

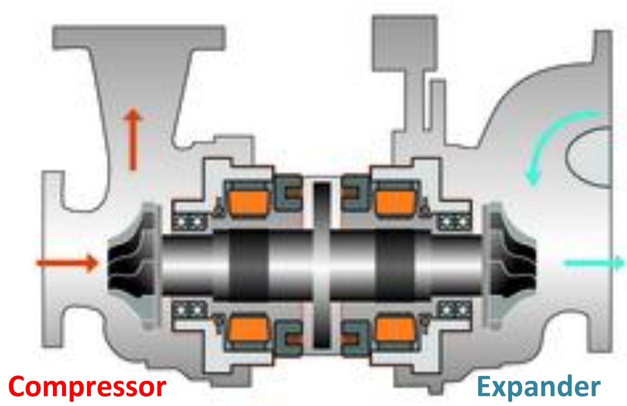

Figure 1 Typical AMB Expander-Compressor

Typically the compressor impeller (left-hand side in Figure 1) is the larger of the two wheels. Gas flow for the expander is radially inwards, axially outwards. Gas flow for the compressor is the opposite – axially inwards, radially outwards.

Expander-Compressor Applications

Applications for AMB expander-compressors fall into two categories:

Figure 2 Operational AMB Expander-Compressor
Note the diffuser cone (far –right), and white ice-ball surrounding the cold expander volute (right), and the thermal protective blanket around the hot compressor volute (left). Photo courtesy the Atlas-Copco Mafi-Trench Corporation.

Refrigeration/Liquefaction Processes

In these applications, the expansion of a gas for refrigeration purposes is the desired output of the machine. By passing through the expander’s inlet guide vanes and impeller, energy is removed from the process gas, resulting in a lower gas temperature and pressure. The process gas is a mixture of hydrocarbons, from which the heavier compounds can liquefy and drop out during the expansion process. These liquids can be condensed and recovered, and are either a profitable by-product or the primary desired output of the refrigeration process. Specific examples of expander-compressor use for refrigeration are natural gas liquid separation, ethylene plant refrigeration and air separation.

In these applications, the (nearly) isentropic expander replaces the traditional (isenthalpic) throttling or J/T (Joule-Thompson) valve. While more complex, the expander is able to expand the gas to a much lower temperature than a J/T valve, thus achieving greater efficiency in the refrigeration cycle.

In refrigeration/liquefaction applications, the compressor is a convenient load device that can accept the energy being removed from the process gas. AMBs are particularly suited for these refrigeration applications.

Pressure Let Down Energy Recovery Processes

In these applications, the recovery of useful work from a high-temperature and/or high-pressure gas is the desired output of the machine. For example, a high-pressure process gas could be expanded to a lower pressure (which may have ancillary process benefits), driving a compressor which provides pressurized air for combustion or some other plant purpose. A familiar example of this type of machine is an automotive turbocharger, where the expansion of hot exhaust gasses drives a compressor which provides pressurized engine combustion air.

HISTORY OF API617, EIGHTH EDITION AND AMBs

The Sixth Edition (February 1995) contained the 1.5- page Appendix J Application Considerations for Active Magnetic Bearings. This was substantially added to in the Seventh Edition (July 2002) to form the 4.5-page Annex 4F, of the same name. While still an "informative annex", this is where we began to see the detailed requirements for AMB machines take shape. It was also here that the API617 committee adopted the dynamic stability evaluation criteria used in the ISO-14839-3 standard. Thus, the 16-page Annex 1E in the Eighth Edition (September 2014) is the result of nearly 20 years of contributions from AMB vendors, users and academicians.

Prior to the Eighth Edition, those engineers performing rotor dynamic analyses and design work for AMB-supported machines applied the LOB-machine-oriented API617 standards and specifications, and negotiated with machine-builders and End-Users on exceptions where the standard was poorly adapted to AMB machines.

With this AMB-focused annex, the need for this negotiation has not been eliminated, but has been reduced and defined. While some standards are made mandatory ("the machine shall meet this specification…"), many topics in the annex begin with "if specified…", indicating it is up to the purchaser of the AMB system or rotating machine to include or omit those particular requirements.

AMB EXPANDER-COMPRESSOR DELIVERY PROCESS

Figure 3 summarizes the steps to comply with API617 for an AMB expander-compressor.

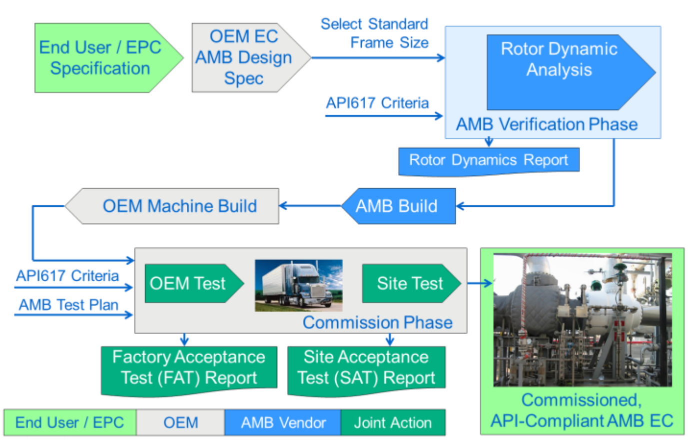

Figure 3 API617 Process of an Expander-Compressor

The End User / EPC Specification generally will follow the format described in API617 Part 4 Annex A. This document specifies operating conditions, gas properties, system component locations, required test options, seal types, etc. In terms of the bearings, it only specifies which technology is to be used (LOB or AMB), and some related bearing diagnostics options.

Based on the End User / EPC Specification, the expander-compressor OEM will select an appropriate frame size to propose to the purchaser of the machine, and populate a standard specification for the AMB supplier.

Selecting a Standard Frame Size

OEMs offer a range of frame sizes of AMB expander-compressors. Nearly all AMB expander-compressors purchased today are built as a standard frame size.

Each frame size can accommodate a range of expander and compressor wheel sizes, loads, and speeds, with some range overlap existing between adjacent frame sizes. Figure 4 shows two extremes of expander-compressor AMB cartridges, while Figure 5 relates radial bearing rotor diameters to machine power and speed.

Based on the End User / EPC specification, the OEM issues an AMB Design Specification. This contains details such as:

- Frame size
- MCOS (maximum continuous operating speed)
- Trip Speed
- Mass, inertia and overhang of expander and compressor wheels
- Magnetic Bearing Controller (MBC) Options (see next section)

Figure 4 Large (240mm) and Small (51mm) E-C AMB Cartridges
Photo courtesy S2M

Selecting an MBC and its Options

As magnetic bearings grow in size, so too does their demand for amplifier power. Each frame size is paired with a specific MBC with an appropriate amplifier power. In other words, once the frame size is selected, the standard MBC for that frame is automatically selected.

Options for the MBC that must be selected at this time include:

- Cable length: the bearing and sensor cables. These can be up to 1000 m (3082 ft) in length.
- Cabinet finishing (site-specific paint colors)
- AC to DC power converter (input power options)
- Customer interface options (serial, digital and analog I/O)
- UPS/Battery options

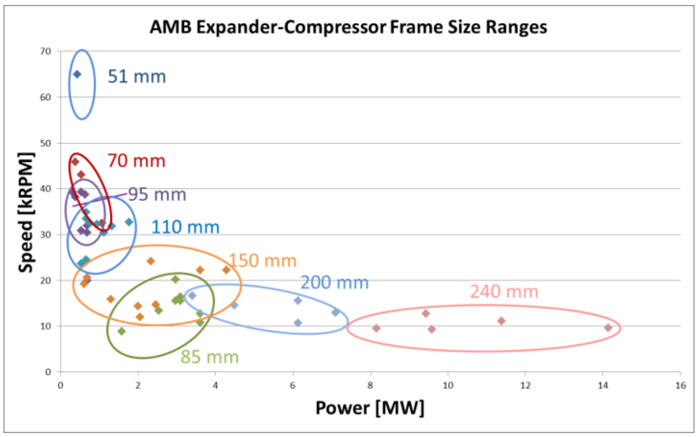

Figure 5 Power, Speed and Bearing Diameters

Once the expander-compressor frame size and MBC are selected, the rest of the delivery process can take place. The major elements of this process are: rotor dynamic analysis, machine build and assembly, and testing (at the OEM and End User sites). These steps will be discussed in detail below.

Design of a New Frame Size

End Users and EPCs generally are not involved directly in the design of a new frame size. They may provide input as to the requirements, but this activity is a collaboration between an AMB vendor and the expander-compressor OEM. The design of a new frame size invariably includes a "frame study" in which the AMB and machine designers map the maximum combinations of power, speed and process loads which the conceptual machine can accommodate.

Now that major expander-compressor OEMs have well-established frame sizes available, the design of new frame sizes is rarely undertaken. Thus, while they are manufactured to order (volumes do not yet allow for "Commercial Off-The Shelf" frame availability), nearly all AMB expander-compressors ordered today are repeat constructions with established reference cases. They vary only in terms of the options and wheel properties, within the bounds set by their previously-done frame study.

End Users and EPCs should specify that expander-compressors designed prior to the release of API617, Eighth Edition must meet the new standards. Because the major AMB vendors and expander-compressor OEMs contributed to the creation of Annex E, the annex reflects the current electrical and mechanical design practices of these OEMs.

Expander-compressor OEMs and AMB vendors may present exceptions for approval or negotiation, detailing where they do not comply with elements of the standard.

ROTOR DYNAMIC AND LOADS ANALYSIS

As mentioned, prior to the publication of Annex 1E, API617 was being applied to AMB rotating machines as closely as possible, with exceptions and additional analyses as preferred and negotiated by the stakeholders. Annex 1E confirms this practice and clarifies many AMB-specific requirements.

This section outlines the analysis for a proposed AMB expander-compressor, focusing on the AMB-specific aspects.

It is important to note that this analysis is a confirmation that the proposed expander and compressor wheels are suitable for the selected frame size, and that the selected frame size is suitable for the process loads. The wheels should have been selected from within the speed, power and wheel mass ranges specified for the frame size. Prior to quotation, OEMs or AMB vendors can perform an abbreviated version of this analysis to ensure proper frame selection.

The output of this process is the traditional API617-compliant rotor dynamics report that all rotor dynamics professionals are familiar with.

Note that all of the sample images are for an expander-compressor utilizing 110mm radial bearings.

The most important point the authors would like to make regarding the rotor dynamic analysis of an AMB expander-compressor is that, with the exception of some noted criteria changes, the process of analysis is identical to that used for all API617-compliant machines.

Lateral Analysis Steps

Creation of the Finite Element Model

The lateral analysis requirements for AMB machines are nearly identical to those of LOB machines, in terms of amplification factors, separation margins, and unbalance response.

A rotor Finite Element Model (FEM) is first created by entering geometry, mass and stiffness properties into a table interface. The resulting table defines the nodes of the FEM, and forms the input for an ordinary-differential-equation solver routine, the output of which is the free-free, undamped rotor model. The format of this model can vary, but its characteristics (free-free natural frequencies, mode shapes, etc.) serve as good criteria with which the OEM and AMB vendor can compare their models, if separate models are used. With the AMB component locations (auxiliary bearings, sensors and magnetic bearings) added to the model, engineers examine the modal visibility (discussed below).

The outputs of this analysis step are the Mode Shape

Diagrams, the free-free Campbell Diagram, and the Undamped Critical Speed Map (Figure 6, Figure 7, and Figure 8). For the Campbell Diagram and Critical Speed Map, API617 dictates the frequency ranges over which they must be plotted.

The free-free Campbell Diagram is the most useful of these outputs, and is often used in isolation as a quick check of the separation margin of the first bending mode.

Accuracy of the inputs from the machine OEM is important as this stage. The base rotor model will exist from the frame design, but the mass, inertia and axial center of gravity data of the wheels will normally be unique to the machine. It is rare that more than a handful of machines have identical wheel properties, as wheel properties are customized for the target process conditions. The dynamic coefficients of the labyrinth seals can also vary widely from machine to machine, although these have a smaller overall significance than the wheels. A common cause of poor agreement between the rotor dynamic model and field measured data is disagreement in the mass of the modeled and real wheels.

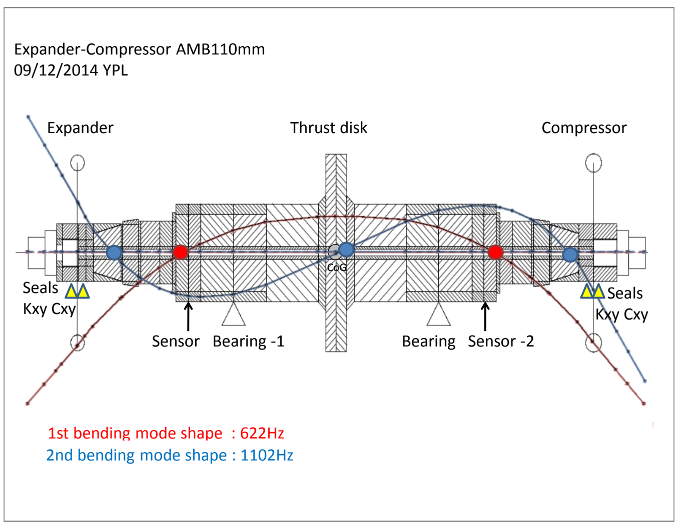

Figure 6 Mode Shape Diagram

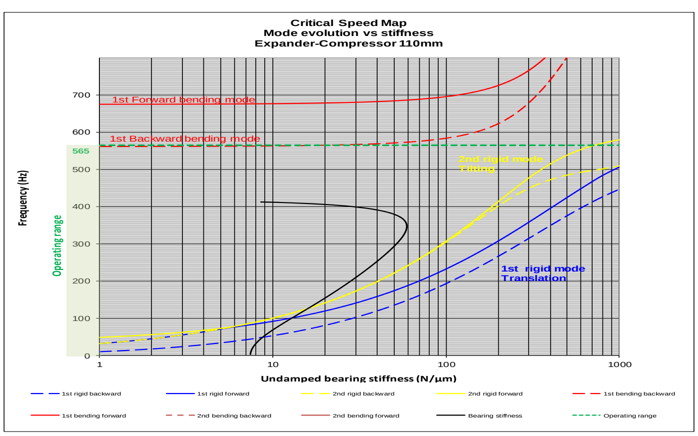

Figure 7 Undamped Critical Speed Map

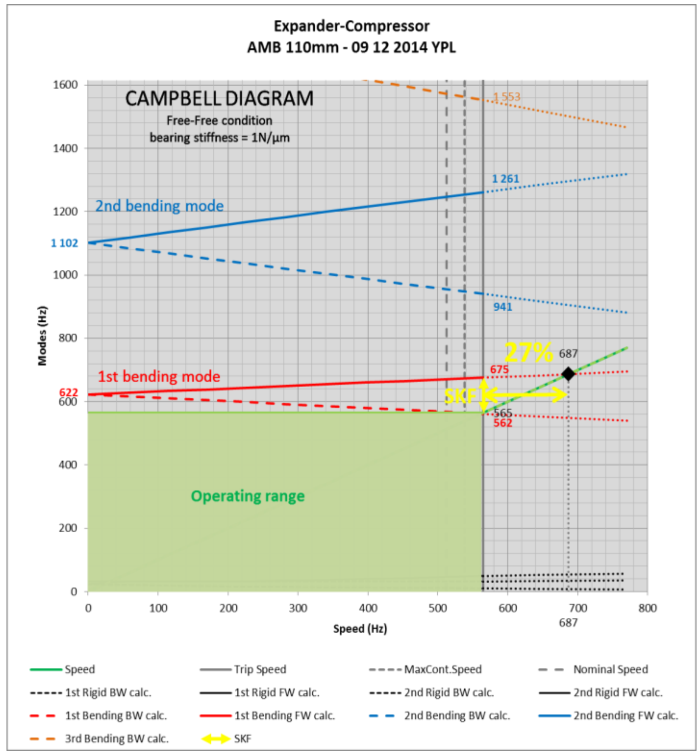

Figure 8 Campbell Diagram

Stability Analysis and Tuning

The rotor model is then incorporated into a closed-loop dynamic model including the other system elements (magnetic bearings, position sensors, DSP, amplifiers, seals, etc.). At this stage, input from the OEM is required for the dynamic coefficients (cross-coupling and direct stiffness) of the seals and (if applicable) the impellers.

The mathematical relationship between shaft position and bearing currents, implemented in the MBC, is known as the bearing tuning, and is also commonly referred to as the control law or the compensator design. The process of optimizing bearing tuning for individual machines is generally performed by the AMB vendor.

Stiffness vs Stability

The criteria for "good" bearing tuning are very simple. First, the tuning must provide sufficient stiffness and damping to provide adequate unbalance response. This is discussed later in this tutorial. API617 provides stiffness criteria indirectly – it provides vibration limits, while the AMB system can only keep vibration within the criteria by supplying adequate stiffness.

Secondly, the closed loop system must be sufficiently stable. In simple terms, the amplifiers can be instructed to act aggressively to position excursions (high gain, or stiff tuning). Or, they can be instructed to react in a lazy manner, providing only small current changes for position variations (low gain, or soft tuning). As a rule, stiffness and stability are mutually competitive. If bearing tuning is too stiff, a rotor will be unstable and vibrate against the auxiliary bearings with the slightest disturbance, or not levitate at all.

Stability Criteria

Unlike for stiffness, API617 provides very clear and direct criteria for tuning stability. The criteria for system stability are taken from ISO14839-3, the ISO standard which addresses the vibration of AMB-equipped machinery. It is convenient that API employed already-accepted criteria rather than establish a new and competing technique.

The authors do not wish to present a detailed explanation of the stability criteria here, but refer the interested reader to API617, the ISO standard, and (Swanson et al), but will give a brief summary of the process.

The process by which stability is measured is simple, reliable and is automated in some in the control software of some MBC platforms. It involves, as a minimum, calculating (or measuring during commissioning) a specific transfer function, known as the "sensitivity transfer function," for each bearing axis, with the machine at zero rotational speed. The customer may also specify that the sensitivity transfer function must be measured while rotating. The amplification, or gain, of this transfer function must lie below a certain value for the system to be considered stable. ISO defines four "stability zones" as described in Table 2.

Figure 9 illustrates a typical analytical sensitivity transfer function, calculated for the sample expander-compressor, at full speed. The three traces correspond to the VW13 (expander side) and VW24 (compressor side) radial bearings, and the axial bearing.

In addition to the sensitivity transfer function requirements, API617 requires that the closed-loop system modes have positive log decrements, with a minimum value depending on the frequency (either above 0, above 0.1, or above a calculated value between 0 and 0.1). These log decrement values can only be calculated, and not compared to measured values at the commissioning stage.

API617 provides criteria for selecting if a Level I or Level II Stability Analysis should be performed. In the experience of the authors, the Level II analysis should be made initially, without considering the less detailed Level I.

Closed Loop Transfer Function

API617 also requires the calculation of the closed loop transfer function. The interested reader should refer to the previously mentioned references for a detailed description of the closed loop transfer function. In short, it is a useful snapshot of the dynamics of a single axis, and is mainly used for model verification purposes at the commissioning stage.

API617 also has "if specified" requirements for the calculation of the open loop transfer functions, as well as cross-coupled transfer functions, which allow a designer to explore the relationships between each of the four radial sensors and each of the four radial actuators.

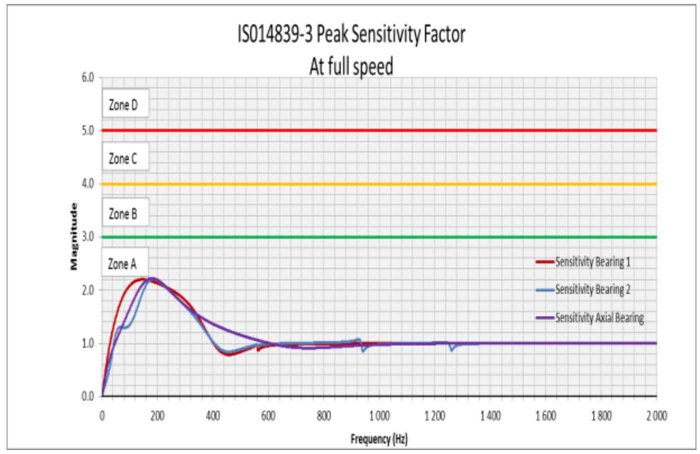

Figure 9 Calculated Sensitivity Transfer Function

Table 2: ISO 14839-3 Stability Zones

Zone	Sensitivity TF Gain	Stability Description
A	< 3.0 (9.5 dB)	The sensitivity functions of newly commissioned machines would normally fall within this zone. Safe to run.
B	< 4.0 (12 dB)	Machines with sensitivity functions within this zone are normally considered acceptable for unrestricted long-term operation.
C	< 5.0 (14 dB)	Machines with sensitivity functions within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.
D	>5.0 (14 dB)	The sensitivity function within this zone are normally considered to be sufficiently sever to cause damage to the machine

A Note about the Backward First Bending Mode

Expander-compressors tend to be quite gyroscopic compared to other machines covered by API617. This means they experience comparatively more mode separation between forward and backward modal frequencies as rotation speed increases. API617 E.4.8.6.2 requires that all modes within the running speed (<N_{mc}) have a log decrement greater than 0.1, and greater than 0 for all modes above 125% of N_{mc}. In practice this can be impossible or require too great of a stiffness compromise to achieve for the backward first bending mode, because the gyroscopics force this mode to enter the running speed. This is an exception that would normally be requested by the author.

Unbalance Response Analysis

The unbalance response analysis requirements for AMB expander-compressors are nearly identical to those of non-AMB machines (regarding amplification factor, separation margin, etc.), with one exception. The mechanical test vibration limit (A_{vl}) is three times greater than for LOB machines. As explained in (Swanson et al), magnetic bearings are less stiff and provide more damping than LOB systems, and thus transfer less force to the machine casing due to unbalance, for an equivalent amount of shaft displacement. A larger vibration limit takes advantage of the unique properties of AMBs with no compromise to machine safety. See Figure 10 for one of the typical outputs of an Unbalance Response Analysis.

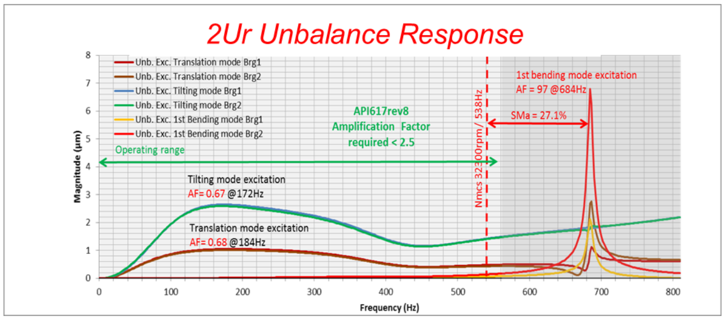

Figure 10 Unbalance Response Analysis Plot

Load Analysis

A rolling-element or hydrodynamic bearing will, if temporarily overloaded (but not so much as to cause immediate failure), continue to operate and support the rotor, but with shortened life and higher running temperature. In contrast, an AMB, if overloaded, will no longer constrain the rotor, and will allow it to contact the auxiliary bearings. Because an overload condition will cause an immediate machine trip, API617 contains the following two specified AMB force factor of safety requirements.

The first is that for each radial unbalance response case, the factor of safety relative to the maximum rated dynamic capacity of the radial bearing shall be 1.5 or greater.

The dynamic capacity of a bearing is frequency dependent. If an AMB system has a static capacity of F_{max}, at low frequencies it will be able to create a sinusoidal force on the rotor, with an amplitude of F_{max} (varying between + and – F_{max}). As the frequency of a sinusoidal force command increases, the commanded time-rate-of-change of the force (dF/dt) will exceed the bearing's ability to produce that force. Above a certain frequency, the bearing system will create a sinusoidal force, the amplitude of which continuously decreases with frequency. This transition frequency is seen on Figure 10 as the point at which the horizontal maximum bearing capacity envelope becomes a slanted line.

The second capacity factor of safety requirement relates to the maximum static axial bearing capacity. Expander-compressors are required in API617 to be equipped with an automatic thrust equalizing valve. This valve is meant to actively maintain a low axial load by venting from or injecting to balancing chambers in the machine. This valve cannot compensate perfectly for thrust loads. Therefore, API617 requires the axial magnetic bearing capacity to be two times greater than the largest anticipated residual loads.

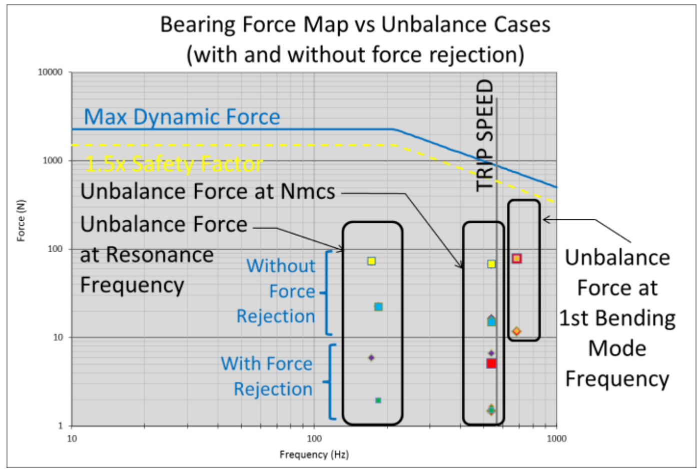

Figure 11 Radial Force Envelop Analysis

Axial Analysis

Just like radial levitation, axial levitation requires feedback control. It therefore requires a full dynamic stability analysis. API617 allows the use of a simple lumped-mass-model. This makes the axial analysis and compensator design much simpler than the radial.

Bearing Stiffness Transfer Function

In summary, once the above is complete for the simulated AMB – rotor – MBC system:

- the stability has been validated (through the sensitivity transfer function)
- the stiffness has been validated (through the unbalance response analysis)
- the static and dynamic load capacity has been validated

The expander-compressor OEM, and potentially the machine End User, will wish to perform their own independent rotor dynamic analysis. API requires AMB vendors to provide sufficient detail to OEMs or End Users / EPCs to do this.

The AMB vendor will provide the combined transfer function of the controller, amplifier, bearing and sensor. This is provided as a *bearing stiffness transfer function*. This is a transfer function that characterizes force response at the bearings to shaft motion at the sensors. An independent analysis can combine the bearing stiffness transfer functions with an independently generated finite element rotor model to simulate the combined rotor/control behavior.

The bearing stiffness transfer function can be provided in pole/zero, state-space or frequency-response data format. Most usefully, it can be provided in engineering units of stiffness (N/m, or lbf/in) and damping (N-s/m, or lbf-s/in) versus disturbance frequency.

The bearing stiffness transfer function can be used to verify the unbalance response, and the response to process loads and conditions. It cannot, however, be used for stability analysis. The bearing stiffness transfer function simulates a bearing which, without a feedback loop, has stiffness and damping properties, at each frequency within the analysis range, matching those of the simulated AMB system.

Auxiliary Bearing Analysis

The AMB vendor is required to demonstrate, by a basis agreed upon by the vendor and OEM, that the auxiliary bearings will maintain zero-contact between the rotor and stationary components during a delevitated coastdown. This analysis will include the effects of ball bearing and damping ring compliance, unbalance and process forces, rotor flexibility, and, if specified, magnetic bearing forces as well.

Minimizing the time required to reach zero speed is critical for maintaining auxiliary bearing performance.

Other AMB-Specific Considerations

Undamped Critical Speed Map Applicability

The UCSM is useful in analysis of LOB bearing systems. The stiffness of hydrodynamic bearings changes with the shaft rotation speed, which in turn modifies the modal frequencies, and the UCSM is a useful way to examine this evolution.

The stiffness and damping of AMBs does not change significantly with rotor speed. It depends on the frequency of the disturbance encountered, not the frequency of the rotor. In other words, while the stiffness and damping to synchronous forces changes with rotation frequency, the response of AMBs to non-synchronous disturbances is independent of rotation speed.

This fact renders the UCSM less useful for AMB systems than for LOB systems. In the experience of the authors, this map is not used by AMB designers or rotor dynamics engineers, and is included in analyses only to comply with API and provide customers with a familiar plot.

Analysis Software

The solver routine software used by AMB vendors, in the experience of the authors, normally is provided by third party software companies or groups (XLRotor, XLTRC, Madyn, university research groups such as ROMAC, etc.). These validated programs are used as a part of a larger application or suite of software tools that are tailored to the AMB vendors' specific analysis. It should be noted that most commercial rotor dynamics software applications now include or are developing AMB modules.

Modal Visibility

In examining the free-free mode shapes, ideally a modal node will not be co-located with a sensor (low modal visibility) or a bearing (low modal controllability), and will not lie between the associated sensor and bearing of an axis. In the experience of the authors, this is not as critical as has been reported – only a minor separation between modal nodes and bearing or sensor locations is sufficient to achieve good control, assuming other good design practices are used. Nodes between sensor and bearing locations can also normally be accommodated in the tuning process, as discussed by (Swanson et al).

Unbalance Force Rejection Control

API617 requires that all levitated lateral analysis be done with Unbalance Force Rejection Control disabled. This is a commonly-used technique that instructs the MBC to NOT react to synchronous rotor displacements. The MBC is instructed to ignore the rotor imbalance. This has the result of allowing the rotor to rotate about its mass center (see Figure 12) rather than trying to force it to rotate about its geometric center. This greatly reduces or eliminates the synchronous vibrations sensed by the bearing housings. This also reduces the amplifier effort by changing its output currents from sinusoids at the rotation frequency, to DC values, which vary only with process loads. In practice, enabling unbalance force rejection can either slightly increase the synchronous position response, or, in the majority of cases, reduce the position response, depending on the machine speed, bearing tuning, and dynamics of the system.

For people encountering this technology for the first time, it is a remarkable experience to feel a high-speed industrial machine operating with the casing vibration due to imbalance reduced to zero.

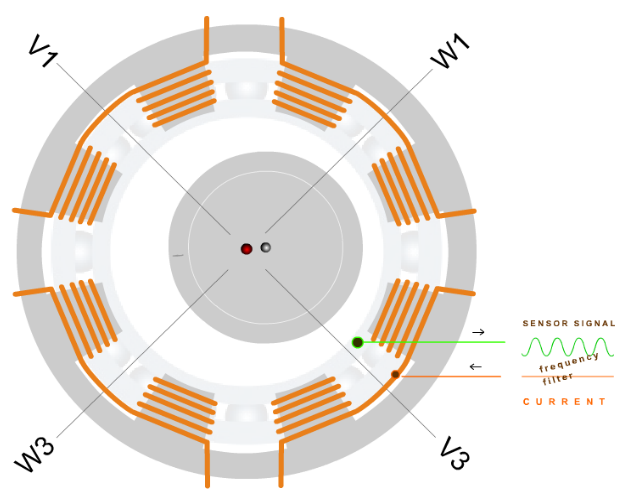

Figure 12 Illustrating Rotation about the Rotor Mass Center

CONSTRUCTION, COMMISSIONING AND TESTING

Figure 13 summarizes the commissioning process for an AMB expander-compressor. Two similar test campaigns are employed for each machine, for the purposes of this paper referred to as the Factory Acceptance Test (FAT) and the Site Acceptance Test (SAT).

A major advantage of AMB technology is the built-in machine intelligence (sensing and computation) which supports the automated verification of API617 compliance.

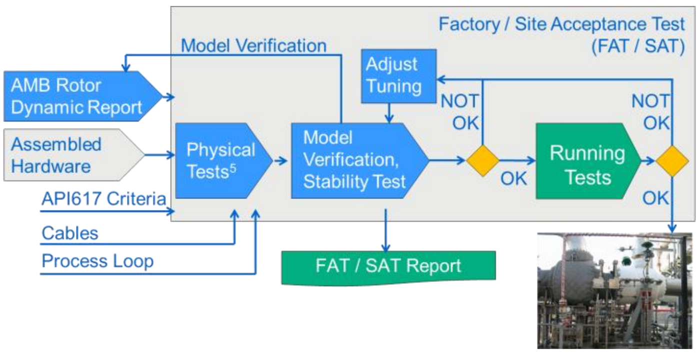

Figure 13 Commissioning and Testing Summary

Construction

AMB hardware supplied by the AMB vendor to the expander-compressor OEM includes:

- two stator cartridges (containing the radial and axial bearings, speed and position sensors, auxiliary bearings and housings)
- rotor cartridges, with the bearing and sensor targets
- auxiliary bearing landing sleeves
- MBC and its options and customizations
- End User cables, test cables (optional)

These components (excluding the End User cables) are standard items, with a lead time dictated by the purchasing agreement in place between the AMB vendor, OEM and End User.

The base rotor, usually including the axial bearing flange, is retained in the machine OEM's scope of supply. There is nothing proprietary about the part that requires construction by the AMB vendor – as long as the material is as specified, it simply a machined item. Also it contains the critical and highly-controlled features for wheel attachment.

The rotor cartridges and auxiliary bearing landing sleeves are installed onto the rotor by the OEM, through either hydraulic or thermal expansion. They can either be installed in a finished state, or finish-machined on the rotor, as the OEM prefers. It is critical to avoid damage to the rotor sensor target surfaces. If a sensor surface is clamped in a lathe's three-jaw chuck, even using machined soft-jaws, it can easily be permanently deflected or scratched, inducing a strong three-time position and current response when rotating.

The pressure housings, volutes, wheels, pressure sensors, piping, and all other non-AMB-related equipment are provided by the expander-compressor OEM.

After construction, the OEM will install the machine into either a test loop (prior to the FAT) or the End User process loop (SAT). The test loop can be as simple as an evacuated pipe loop, or as accurate as a fully pressurized test loop with an inert process gas.

Junction boxes typically lie between the machine and the MBC. It is the responsibility of the OEM (or End User technicians if in the field) to connect the cables, junction boxes and MBC properly.

Physical Tests (FAT and SAT)

Once constructed, a number of physical tests are performed. These include:

Wire and Insulation Checks: using a multimeter and insulation tester, the resistance, inductance and isolation to ground are checked for each wire pair. This process is important to identify mis-labeled or mis-wired connections.

MBC Cable Customizations: if required, the MBC is customized for cable length.

Interlock, Alarm and Communication Checks: prior to levitation and rotation, the machine protection interlocks must be connected and verified, in terms of both hardware and software operation. Bypass valves, temperature and pressure sensors, automatic thrust equalizers, and many other sensors and actuators must have their control and command links established and tested, as with any industrial rotating machine. The AMB position alarms must also be checked at this stage.

Initial Levitation: the rotor must be levitated for the next steps. If the Revision A bearing tuning, created during the rotor dynamic analysis, cannot achieve stable levitation, it must be adjusted. If the model differs significantly from the physical system (because a wheel mass is different, or a wheel is poorly attached, etc.), there can sometimes emerge a high-frequency ring on the rotor. By reducing the overall gain of the tuning, and adding or moving gain reduction tuning filters using software tools, stable levitation is achieved.

Position Sensor Calibration/Sensitivity and Clearance Checks: This allows the clearance to the auxiliary bearings to be examined, as an automated or manual process. This is done by moving the rotor within the auxiliary bearing clearance, looking for signs of contact with the bearings, and adjusting the levitation set point to place the rotor in the middle of its clearances.

A characteristic example of a calibration output is shown in Figure 14 below.

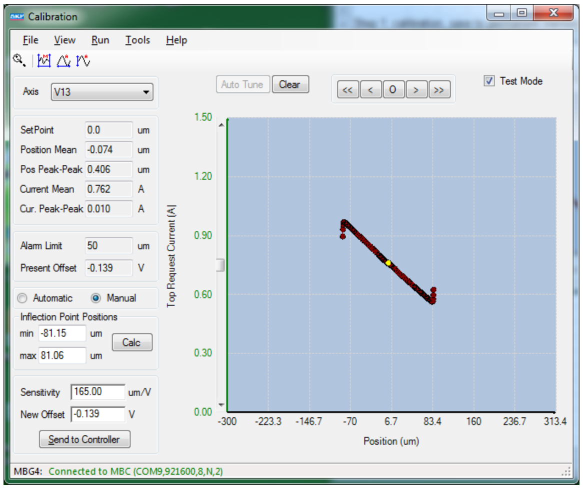

Figure 14 Sensor Calibration Result

Model Verification (FAT and SAT)

API617 requires that, once levitated, the dynamic model be verified against the physical machine. This can be done by one of two methods. The first is the Unbalanced Rotor Response Verification Test (URRVT). This involves measuring multiple unbalance case responses - running the machine over its speed range with controlled amounts and locations of imbalance. This is a time consuming process, normally requiring machine disassembly and risk of damage. Also, the expander-compressor ought not to be rotated until the tuning is qualified and model validated. As a result, in the experience of the authors, the URRVT is rarely, if ever, undertaken for expander-compressor model validation.

The other model verification option given by API617 is the Transfer Function Based Procedure (TFBP). This requires that closed loop transfer functions, measured on each machine levitation axis, be compared to those generated from the rotor dynamic model, while levitated and non-rotating. The standard provides criteria within which the measured and modeled closed-loop transfer functions must match, up to 125% of N_{mc}. Modern MBC platforms can collect the data necessary for this validation automatically.

The standard also includes "if specified" clauses allowing the purchaser to require the measurement of at-speed transfer functions (measured while rotating) and cross-coupled transfer functions. These are the transfer functions between each of the four radial sensors and each of the four radial actuators.

Only the TFBP is applicable to the axial axis.

API617 requires that the model be updated to match the field measurements. This means adjusting the model bearing tuning to match that entered into the MBC, and then adjusting the rotor and/or associated system subcomponents until the closed loop transfer functions match to within the specified criteria. This is discussed further below.

Stability Check (FAT and SAT)

Once the model has been verified to the satisfaction of the AMB vendor and purchaser, the tuning stability is checked. This is done in the same way as at the analysis stage – using the sensitivity transfer function. As with the analysis stage, the peak of radial sensitivity transfer functions must fall within zone A, and for the axial axis may fall within zone A or B. Modern MBC platforms can measure these transfer functions automatically.

AMB vendor field service technicians typically have additional tests (for example, recording a position step response test), not specified in API, which they perform to further characterize the closed-loop stability of the levitation system.

API617 allows axial modes dominated by the properties of a flexible coupling to be excluded in both the model verification and the sensitivity analysis.

Figure 15 shows a comparison of modeled and measured closed loop transfer functions, while Figure 16 shows a measured sensitivity transfer function.

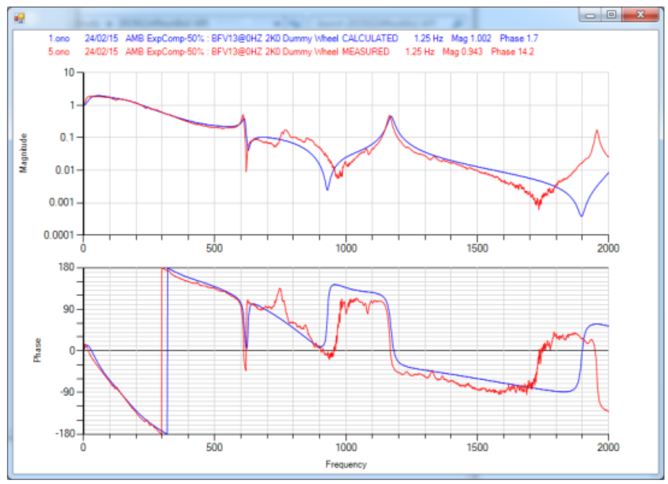

Figure 15 Measured and Modeled Closed Loop Transfer Function

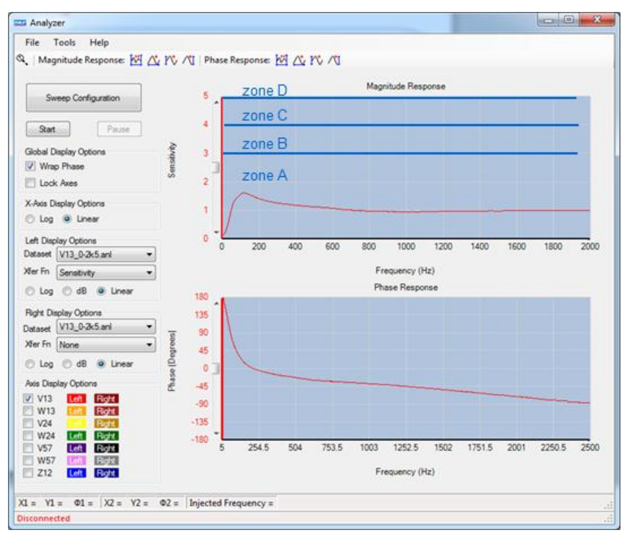

Figure 16 Measured Sensitivity Transfer Function

Running Tests (FAT and SAT)

With the physical checks complete, the process or test loop connected, and the tuning stability and model verified, the machine is ready to run. The main goal of the Mechanical Run Test is the validation of the expander and compressor performance. In terms of the AMB system, running tests are primarily to ensure the residual imbalance and process loads do not result in position excursions exceeding the alarm limits. Position alarm limits are normally set at between 30% and 50% of the mechanical clearance (the nominal radial distance between the rotor landing sleeve and the auxiliary bearing inner race). These are normally larger than the A_{vl} vibration limit set by API617. The specified A_{vl} while running is the same as is used at the analysis stage.

Figure 17 through Figure 19 show typical measurements taken to validate the position performance of an expander-compressor. Note that these three figures were collected on a motor-driven compressor, as the smooth speed control creates a better illustrative example of the measured unbalance response plot.

Verification that the applied forces are within the bearing capacity factor-of-safety can be observed in a spectrum plot of the measured bearing currents, as shown in Figure 18 (the limits are much higher than the scale of the plot and not seen).

If a system is equipped with unbalance force rejection, at least two runs to full speed are normally done, one without and one with rejection enabled.

Potential Differences between FAT and SAT

Differences in test conditions between the FAT and SAT can lead to changes in observed performance, and can also require modification to the bearing tuning. The major sources of FAT to SAT differences are noted below.

For simplicity, the test loop used during the FAT is normally not filled with hydrocarbon gas. Thus the cooling effects of the process gas on the magnetic bearings, and their resulting steady-state temperature, can vary from FAT to SAT.

Differences in mounting of the machine between the FAT and SAT can lead to variation in the measured transfer functions, and can also require modifications to the bearing tuning.

Differences in cable length, between the FAT test cables and SAT job cables, can lead small to variations in the measured transfer functions and sensor sensitivities.

Finally, careful checking of the wiring is required at both the FAT and SAT to ensure no errors have been made during interconnection.

Auxiliary Bearing Validation

API617 requires that the AMB system have some means for verifying the operability of the auxiliary bearings that does not require machine disassembly. One aspect of this is the clearance check – if the clearances to the auxiliary bearing have changed dramatically, barring a sensor malfunction, it normally means that damage has occurred to the bearing race or balls.

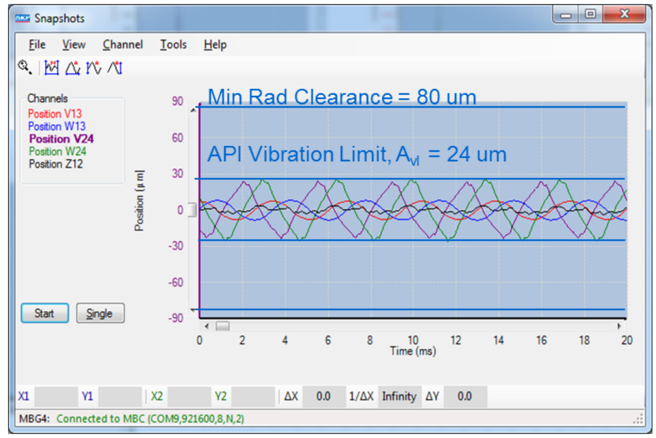

Figure 17 Maximum Position Response, 17 kRPM

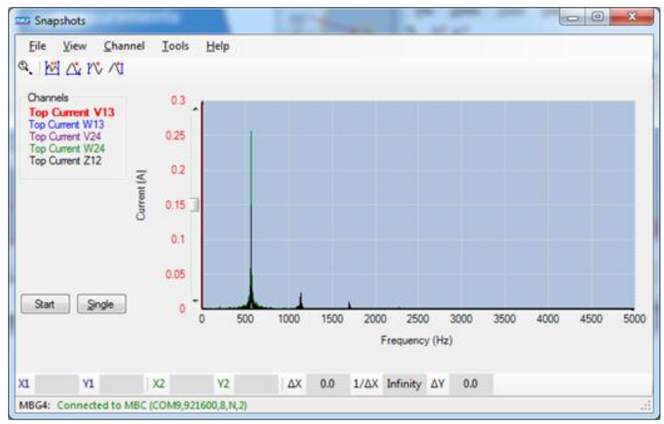

Figure 18 Bearing Current Spectrum

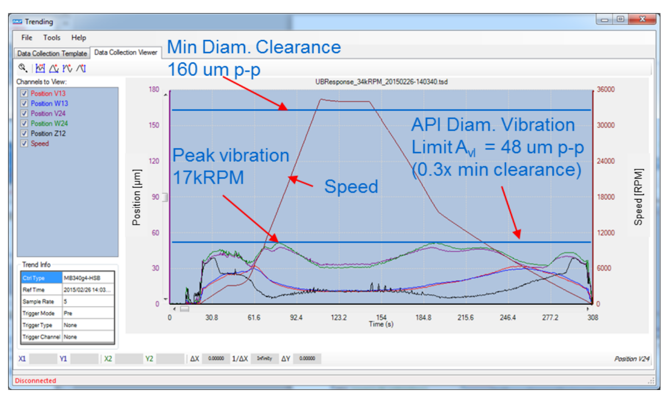

Figure 19 Unbalance Response

Another technique is known as a “touch and go” (see Figure 20), in which the rotor, while rotating at an agreed-upon speed, is delevitated (by inhibiting the amplifiers, for example), and then re-levitated. In this test, the position response and speed can be monitored, and based on previous tests and jointly-agreed criteria, the status of the auxiliary bearings can be estimated. This test also validates the ability of the control system and bearing tuning to recover from the upset condition of rotating on the auxiliary bearings, and validates that the auxiliary bearing damping system prevents destructive whirl from becoming established.

A full coastdown test is sometimes specified, after which the auxiliary bearings are normally replaced to ensure a system with the maximum life possible remains in operation.

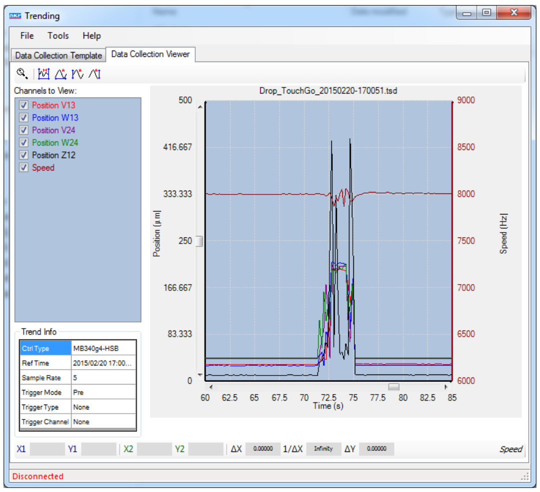

Figure 20 Position (peak-peak) and Speed Response of a "Touch and Go" Test

A Note on Model Validation

Model validation is useful for identifying major mechanical errors, such as a wrong or improperly attached impeller wheel. Gross discrepancies should be investigated and understood prior to beginning bearing tuning efforts.

However, spending effort (and money) to accurately match the modeled and measured transfer functions can in many cases be of limited commercial or technical value. If the AMB vendor uses model-based tuning techniques (the H-infinity method for example) then the model must be very accurate, but if traditional, manual techniques are used (PID, augmented with first and second order filters), updating the analytical model is not normally necessary or useful. Further documentation updates of the rotor dynamics report are of limited utility. This model matching does not contribute to the two key AMB-related performance criteria – 1) that the bearing system is sufficiently stable, and 2) that the bearing system is sufficiently stiff to accommodate process conditions.

For example, it is common for two machines, with (by design) identical housings and impellers, supplied with identical bearing tuning, to have transfer function differences exceeding the API617 model/measurement matching criteria, but for both to meet the stability and performance. Many expander-compressor End Users will purchase a Spare Rotating Assembly (SRA). It is also common for the primary rotor and SRA, due to unavoidable manufacturing tolerances, to have differences exceeding the API matching criteria, when installed in the same machine, even though they are built to be identical. If the bearing tuning must be modified, normally very slightly, then significant time is required to update the rotor dynamics report to have two rotor models, two bearing tunings, and all of the new modeled transfer functions, etc.

Housing natural frequencies can also cause departures between modeled and measured transfer functions. These can vary slightly (but exceeding the API limits) from machine to machine, or even between installations of an individual machine. They can, with difficulty and expense, be modeled post-installation, but accurately predicting them is not feasible, and not necessary, especially for expander-compressors. Expander-compressor manufacturers generally ensure mounting is sufficiently rigid to cause housing modes to be manageable. Engineers skilled in tuning AMB systems will be able to identify unacceptable housing modes, and make a request for improved machine mounting. This occurs during OEM testing more commonly, when the machine may be only temporarily mounted.

Unless specified by the purchaser, or unless model-based tuning is employed, the utility of spending large amounts of time on model validation and harmonizing design and field measurement reports should be carefully considered.

CONCLUSION

Annex 1E of the Eighth Edition of API617 is a welcome and valuable addition to the standard, and will support the expanded use of AMB expander-compressors. Some of its strong points are:

- It develops common terminology
- It leverages the ISO 14839-3 stability criteria rather than establishing new norms
- It reaches a good balance between mandatory requirements and "if specified" elements
- It appropriately expands the vibration limits to be applied to AMB machines
- It contains the accumulated experience of AMB developers, machine builders and End Users
- It builds further confidence in potential purchasers of AMB expander-compressors

In the experience of the presenters of this tutorial, the following are some areas where the new Annex should be carefully interpreted:

- The strong requirement for model validation and re-calculation of analytical results using field tuning should be assessed by the user to understand the benefit they would realize from a potentially time consuming exercise of model validation.
- The requirement for all analytical modes to maintain a log dec > 0.1 is not always feasible for the first backward bending mode.
- Increased guidance or criteria regarding bearing cooling requirements would be welcome and useful

In this presentation and the associated tutorial session, we have attempted to illustrate and demonstrate some of the major topics of Annex 1E.

APPENDIX A: COMPONENTS OF AN AMB SYSTEM

Active Magnetic Bearing Expander-Compressors

The principles and details of Active Magnetic Bearings technology have been well described in many sources (Swanson et al, Schweitzer et al, etc.) and will not be covered in detail here. To provide context for the audience member not familiar with AMB technology, the following brief description is provided.

Magnetic Bearing Controller (MBC)

This cabinet contains all of the electronic elements required to achieve levitation in the mechanical system. These include bearing amplifiers, sensor drive electronics, and a computation device, typically a high-speed digital signal processor (DSP), which executes the feedback control loop at the heart of magnetic levitation. The MBC also contains necessary and optional support equipment such AC to DC power converters, condition monitoring or data logging devices, BNC break-out boards for signal access, back-up power (batteries or a UPS), ground-fault monitors, communications links, interlock inputs and outputs, temperature monitoring (RTD) inputs, etc.

AMB vendors classify their MBCs in terms of their most important feature – the voltage and current delivered by their power amplifiers to the magnetic bearings. As amplifier power increases, the capability of an MBC to accommodate higher speeds, heavier rotors, and greater dynamic forces increases. For AMB expander-compressors, typical amplifier voltages range from 150 to 300 Volts, while currents range from 15 to 60 Amps.

Radial Magnetic Bearings (two, four actuators each)

These (quantity two) center the shaft and maintain no-contact levitation radially, working against gravity and all radial process loads. Each bearing, referred to as a stator, consists of four separate actuators spaced at 90 degrees around the circular bearing. The actuators opposite each other form a bearing axis.

Radial AMB stators are constructed of laminated silicon-steel, similar to a transformer core, with coils of magnet wire around each pole. The laminations reduce the formation of eddy currents in the stator magnetic core in the presence of changing flux density, and greatly increase the dynamic performance of the radial bearing.

In some AMB designs, permanent magnets are used to create bias flux in the stators and reduce the demand on the bearing coils, but full electromagnetic levitation is by far the technology most commonly used in expander-compressors.

The radial bearings considered herein are "heteropolar" bearings, meaning any point on the rotor interacting with the bearing stator will see multiple magnetic polarity reversals (from North to South and back) as it completes one revolution.

Being electromagnetic in nature, the radial and axial bearings can only "pull" on the shaft. Repulsive (passive) magnetic bearings are not used in modern high-speed industrial machines, and are not discussed in API617.

Axial Magnetic Bearings (two)

The axial magnetic bearings center the shaft and maintain no-contact levitation axially. They act on a flange which is normally an integral part of the common expander-compressor shaft, but can be affixed to the shaft as a separate component. The axial bearing actuators are made of solid steel cores – a fully laminated core would increase dynamic performance, but is difficult (expensive) to achieve, and not normally necessary. Using partially laminated axial cores and careful design practice is generally sufficient for expander-compressor applications.

API617 requires that each radial and axial bearing stator contain two temperature measurement devices – one primary, and one installed spare.

Shaft Position and Speed Sensors

These provide shaft radial and axial position feedback, to allow the closed-loop feedback position control of the shaft. Without this position feedback, it would be impossible to have stable rotor levitation. Even so-called "sensorless" active magnetic bearings have position feedback – they achieve this without dedicated sensor hardware in the machine and with compromised sensor performance. Normally, two axial sensors are included, one at either end of the machine, to provide rotor to stator differential expansion information.

Sensor assemblies also include a primary and spare shaft rotation speed sensor.

Figure A1 below is an illustration of radial and axial bearing stators, and a position sensor element and their associated rotor elements.

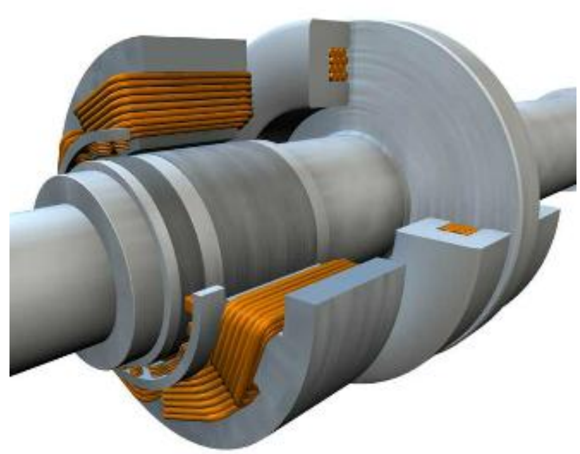

Figure A1 Radial and Axial AMB and Sensor Elements

Auxiliary Bearings

These are rolling element bearings, normally angular-contact ball bearings (ACBB), which support the shaft when it is not levitated by the AMBs, during an upset event (eg MBC failure, bearing overload condition, etc.) or when the machine is not in operation. These bearings are preloaded to prevent drag-induced rotation, which can quickly destroy the bearing, and to remove backlash caused by internal clearance. They can accept both radial and axial loads. They are always cageless, and typically contain steel bearing races and a full complement of steel or ceramic balls. They can be greased, or be coated with dry lubrication (for example MoS_2), or both, and can have seals to retain the grease if present. The radial and axial clearance between the rotor and the auxiliary bearings is typically half or less than the clearance to the magnetic bearings and sensors.

API617 specifies that the bearings must be supported in their housings by a compliant damping material. This material allows the outer race of the auxiliary bearing to move slightly in its housing, providing damping to the otherwise very stiff bearing characteristics.

This damping material is necessary to prevent rotor whirl from developing during a delevitated coastdown from high speed. Ideally, while running on the auxiliary bearings, a horizontal rotor will remain in the bottom section of the auxiliary bearing, rocking slightly back and forth, only climbing up the inner race by a few degrees. Rotor whirl (in this context) is when the rotor climbs up and around the inner race, and drives around and around within the auxiliary bearing clearance. The damping material transforms lateral motion of the outer bearing race (kinetic energy) into small amounts of heat. This energy removal from the rotor translational motion dynamic system reduces the likelihood and severity of this type of rotor whirl.

Housings

Housings are required to attach and combine all of the magnetic bearing components together and to the expander and compressor volutes. The housings are mentioned here because OEMs and AMB vendors work together to carefully select the housing materials used in the machine frame sizes. Both non-magnetic (austenitic) and magnetic (martensitic) stainless steel is used to manage leakage flux from and around bearings and sensors.

As required by API617, a radial and axial bearing, sensor and auxiliary bearing are incorporated into a removable housing referred to as a cartridge. A machine will have separate expander-side and compressor-side cartridges.

Rotor

The rotating assembly consists of the expander and compressor impellers, or "wheels". Between the wheels are all of the rotating components of the bearing, sensor and seal systems. The rotor is constructed of a shaft with a central flange (integral or attached) for the axial bearing to pull upon, and laminated sections beneath the radial bearing stators. While laminations in the radial stator core increase dynamic performance, they are necessary on the rotor to not only increase performance but to prevent the bearing rotor from quickly melting from the rapid magnetic flux reversals seen on the shaft.

The rotor also supports sensor targets (laminated or solid) and removable, hardened landing sleeves, which interact with the auxiliary bearings.

Typical clearances between the rotor and the position sensors/magnetic bearings (the "magnetic gap") are 0.4 mm – 1.0 mm (0.016" to 0.040"). Between the rotor and the auxiliary bearings (the "mechanical gap") the clearance is normally half that of the magnetic gap or less. See Figure A2.

AMB expander-compressors are normally sub-critical machines, meaning they operate below the rotor first flexible mode frequency.

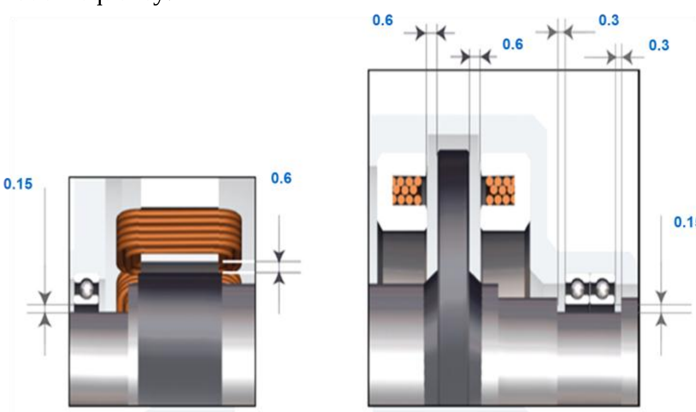

Figure A2 Typical AMB E-C Radial and Axial Air Gaps (mm)

Cables

The connection between the MBC and expander-compressor is achieved by two custom industrial cables per bearing cartridge, one carrying amplifier currents and the other carrying sensor signals, including RTD temperature signals. As length increases, these cables can be a significant portion of the overall system cost, and can reach up to 1000 m (3281 ft) in length depending on the MBC and cable technology employed.

It is common for a machine to be tested at the OEM factory with a standard set of short test cables, and commissioned at the End User site with long, custom cables.

Magnetic Bearing Axis Conventions

AMB expander-compressors are "five-axis" magnetic levitation machines, with active levitation control along one axial axis, and four radial axes. The sixth degree-of-freedom is rotation about the axial axis, controlled by the operation of the compressor and expander.

The axial axis is referred to as the Z axis. Instead of an X-Y horizontal-vertical configuration, as noted (but not specified) in API617 the radial axes almost always have a diagonal (±45°) configuration, referred to as V and W following the right-hand rule (V rotates into W with the right hand grasping Z, thumb is positive Z). This allows both radial axes to work against gravity, rather than one axis levitating vertically and one horizontally. The typical naming convention for the radial and axial axes is:

Expander Radial Bearing:	V13 (said V-one-three)
	W13 (said W-one-three)
Compressor Radial Bearing:	V24 (said V-two-four)
	W24 (said W-two-four)
Axial Bearing:	Z12 (said Z-one-two)

The "top" bearings are the actuators that pull in the positive direction along their respective bearing axes (Z1, V1, W1, V2, W2), while the bottom bearings pull in the negative direction (Z2, V3, W3, V4, W4). See the illustration in Figure A3. If one wishes to blend in with AMB professionals, one should be sure to use, for example, the term "V-one-three" and not "V-thirteen".

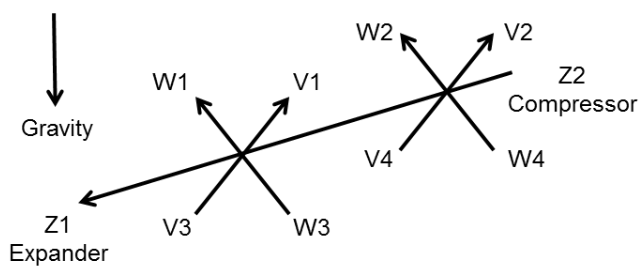

Figure A3 AMB Axis Naming Convention

Part 4 of API617 contains design specifications relating to these machines generally and specifies that if magnetic bearings are used, they shall be supplied in accordance with Annex E of Part 1.

APPENDIX B: ADVANTAGES OF AMBs IN EXPANDER-COMPRESSORS

The advantages of AMB-equipped machines are summarized below:

Maintenance/Reliability: With their zero-contact support, AMBs are accepted as the lowest maintenance, highest reliability bearing solution available, when used within their operating limits. In terms of the machine, only auxiliary bearings are considered a consumable part, and they require no attention on a properly operating machine.

For the MBC, cooling fan filters, which can require periodic cleaning depending on site air quality, and backup batteries, which can degrade over several years, are the only regular maintenance items. Electronic component degradation and obsolescence can limit the MBC service lifetime to a few decades; however these issues can normally be managed with a simple spares management strategy or through an MBC upgrade schedule. The MBC can be replaced or serviced during a brief planned machine outage.

Removing oil bearings eliminates the maintenance required on the entire oil system and all of the associated sensors (level, temperature, and pressure), valves, piping and filters.

Zero Contamination: Particularly in the case of cryogenic systems used in ethylene plants, the absence of lubrication oil eliminates the risk of frozen oil fouling the heat-exchanger (cold box). It is well known that once fouled, it is almost impossible to completely decontaminate modern heat exchangers.

Reduced Footprint, Mass: While magnetic bearings typically require a slightly larger envelope in the expander-compressor itself, the overall machine footprint and mass are much smaller due to the absence of an oil skid. This is illustrated in Figure B1 below, which is an aerial view comparison of the footprints of comparable LOB and AMB high-speed motor-compressor units. While not an expander-compressor, this example is still illustrative. This foot print reduction is especially critical off-shore. The reduced mass also decreases the crane and transport requirements, and in some cases allows AMB units to be installed on an elevated mezzanine instead of directly on the ground.

Better Condition Monitoring, Diagnostics: AMB expander-compressors have built in vibration, bearing load, speed and temperature sensing that can provide valuable reliability information when the MBC is connected to online condition monitoring software.

Reduced Case Vibration: Using a control technique known as Unbalance Force Rejection Control (also known as auto-balancing or vibration control) the synchronous vibration induced on the machine by the unbalance forces of rotation can be eliminated. This is achieved by allowing the rotor to rotate about its mass center, rather than its geometric center. See Figure 12.

Environment: The elimination of oil reduces the chance of oil spills or leaks. It importantly reduces the fire risk, often reducing the cost of insurance and code-required fire suppression systems. AMB bearings are highly efficient, producing roughly one-tenth of the bearing losses of LOB systems.

Reduction of Seals: AMB expander-compressors typically have a labyrinth seal behind each impeller, to prevent very cold or very hot gases entering the bearing cavities. They do not, however, normally require dry gas seals, due to the submersion of the bearings in the process gas, saving cost and complexity.

Automated Verification of API617 Criteria: MBCs are capable of automating the collection of data and reporting against API617 requirements. This improves the speed of acceptance testing and simplifies the performance evaluation.

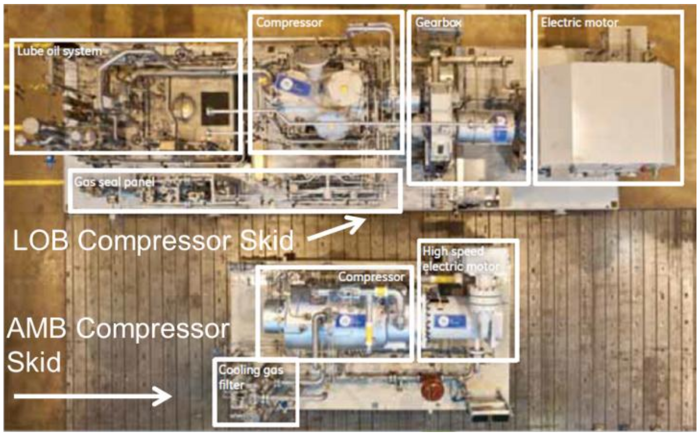

Figure B1 LOB vs AMB Compressor Installation Size
Photo courtesy of GE.

NOMENCLATURE

AMB	Active Magnetic Bearing
LOB	Lube Oil Bearing
API	American Petroleum Institute
E-C	Expander-Compressor
OEM	Original Equipment Manufacturer
EPC	Engineering, Procurement and Construction (contracting corporation)
MBC	Magnetic Bearing Controller
UPS	Uninterruptible Power Supply
FEM	Finite Element Model
DSP	Digital Signal Processor
PID	Proportional, Integral, Derivative (control)
N_{mc}	Maximum continuous speed, RPM
A_{vl}	Mechanical test vibration limit, μm (mil)
F_{max}	Max achievable AMB axis force
dF/dt	Time rate of change of force (N/s)
UCSM	Undamped critical speed map
FAT	Factory acceptance test
SAT	Site acceptance test
URRVT	Unbalanced rotor response verification test
TFBP	Transfer function based procedure
ACBB	Angular contact ball bearing
RTD	Resistance temperature detector

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