Applicability of API 617 8^{th} ed. and ISO 14839-3 in evaluating the dynamic stability of AMB-supported compressors

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***Abstract*— Oil & Gas (O&G) original equipment manufacturers (OEMs) have long used API 617 as the international standard of choice to guide turbomachinery rotordynamics requirements. Recently, the O&G industry has experienced a growing interest in applying active magnetic bearings (AMB) in high-speed applications, hermetically-sealed applications, and compressors with a large performance envelope. As such, the API 617 standards have recently been expanded to include a section (API 617, 8^{th} ed., "Annex E") regarding evaluation of magnetic-bearing-supported compressors. New standards included in Annex E both compliment and compromise existing, well-regarded international magnetic bearing rotordynamics standards, namely ISO 14839-3. This paper reviews and compares the dynamics requirements included in API 617 8^{th} edition Annex E with those of ISO 14839-3. Notable, this paper compares the stability criteria of both standards by simulating the synchronous response, sensitivity transfer function, and closed-loop transfer function of simple 1-D dynamic systems.**

I. Paper Guidelines

A. Introduction

Several standards exist for the evaluation of the dynamic performance of turbomachinery supported by active magnetic bearings (AMB). Of these, ISO 14839 is one of the most widely-used standards [1], as it includes industry-developed criteria for total vibration and closed-loop stability. The standard limits the total permissible rotor-bearing vibration to a percentage of the auxiliary bearing clearance. The standard requires the evaluation of closed-loop stability using the *sensitivity transfer function* (SNTF). The standard does not distinguish between analytical and experimental criteria, meaning the same requirements are used for both design simulations and acceptance testing.

API 617 has historically been the choice international standard used to qualify the rotodynamic performance of compressors and expander-compressors used for oil & gas (O&G) applications. Most of these machines that have been developed to date are supported by fluid-film journal bearings. In recent years, there has been a push towards adapting AMB systems into these compressors to achieve higher speeds, expand the performance envelope, and allow for hermetically-sealed applications. Because of several differences in the dynamic characteristics of magnetic bearings and hydrodynamic bearings, as well as several practical considerations in the operation of the two bearings, API 617 (8^{th} ed., 2014) has recently been amended to include Annex E [2], which details the dynamics performance requirements for AMB-supported compressors. API 617 8^{th} ed., Annex E evaluates total vibration as a function of rotor speed, consistent with its evaluation of total vibration for hydrodynamic-bearing-supported compressors. Swanson et al. [3] provide an overview of the new AMB requirements included in API 617 8^{th} ed.

API 617 8^{th} ed. evaluates stability using the *Level I* and *Level II stability analyses*, which characterize the log decrement of the machine in the continuous operating speed range. The Level I and Level II stability analyses are not used during acceptance testing. The standard also evaluates stability of a critical speed using the calculation of an *amplification factor (AF)* and *separation margin (SM)* from the synchronous response of the rotor, as shown in Figure 1. As the figure shows, the amplification factor is calculated by evaluating the frequencies associated with the peak response and the half-power points. Generally, the AF and SM are verified during a mechanical run test in order to verify the model and to ensure stability of any critical speeds close to or within the operating speed range.

Due to the different methods by which dynamics criteria are evaluated in the two standards, it is not easily deducible what the technical differences are in the two standards. The ISO 14839 standard makes use of standard control system evaluation techniques, while API 617 leans on standard vibration evaluation techniques. This paper explores the two standards and aims to answer the questions:

1.) What are the technical differences between the two standards?
2.) Is one standard more restrictive/conservative than the other?
3.) Are there situations in which one standard is more useful than the other?

While the two standards include several other dynamics criteria, the focus of this paper is to evaluate and compare the total vibration and stability criteria described in this section. Chiefly, this paper explains the differences between the AF, log decrement, closed-loop transfer function, and sensitivity transfer function of a closed-loop pole located within the operating speed range of an AMB-supported compressor.

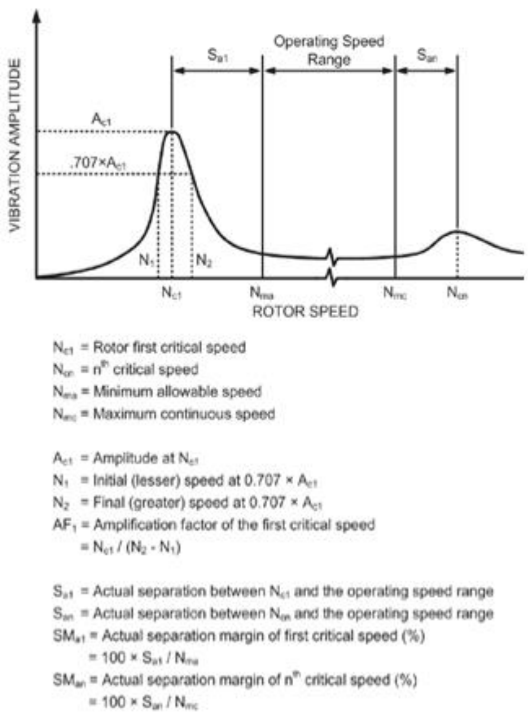

Figure 1. The iconic graphical depiction of amplification factor and separation margin, from API 617 8th ed. [2]

B. A overview of ISO 14839 and API 617 stability and total vibration requirements

API 617 includes 2 major design requirements for qualifying the stability of the rotor:

1.) The Level I and Level II stability criteria
2.) The AF and SM requirements

The Level I and Level II stability criteria are used during the design phase to quantify the effects of cross-coupled stiffness coefficients on system stability. This paper will focus on evaluating the second criteria, the AF and SM requirements.

The AF and SM requirements are designed to ensure that the rotor does not operate close to a lightly-damped critical speed, so the rotor does not impart unbalance forces to the lightly-damped mode. The definition of the AF and SM are shown in Figure 1 above. The AF is calculated by evaluating the half-power points of a critical speed, as shown in Eq. (1).

$$AF = \frac{N_{c1}}{N_2 - N_1} \tag{1}$$

$$N_{c1} = Speed\ corresponding\ to\ peak\ response$$
$$N_1 = Speed\ (< N_{c1})\ corresponding\ to\ 0.707 * peak\ resp.$$
$$N_2 = Speed\ (> N_{c1})\ corresponding\ to\ 0.707 * peak\ resp.$$

Separation margin is defined as a percentage difference between the operating speed range and the critical speed, as shown in Eq. (2)

$$SM = \frac{N_{min} - N_c}{N_{min}}\ for\ subsynchronous\ modes$$

$$SM = \frac{N_c - N_{max}}{N_{max}}\ for\ supersynchronous\ modes$$

$$N_c = Critical\ speed$$
$$N_{min} = Minimum\ operating\ speed$$
$$N_{max} = Maximum\ operating\ speed \tag{2}$$

The AF and SM requirements are summarized as follows:

1.) If the AF<2.5, the critical speed is considered to be well-damped and no separation margin is required.
2.) If the SM is greater than 17% for subsynchronous modes and/or greater than 27% or supersynchronous modes, the mode is considered to be sufficiently removed from the operating speed range, such that it will not be excited by rotor unbalance forces. Thus, there is no requirement for the amplification factor.
3.) All other modes must obey the following criteria:

$$SM_r = 17 * \left(1 - \frac{1}{AF - 1.5}\right)\ for\ subsynchronous\ modes$$

$$SM_r = 10 + 17 * \left(1 - \frac{1}{AF - 1.5}\right)\ for\ superynchronous\ modes$$

$$SM_r = required\ separation\ margin$$
$$AF = Amplification\ Factor \tag{3}$$

The AF and SM are generally measured when possible during the unbalance verification test. This measurement is performed for both model verification and to validate the robustness of the system. API 617 8th ed. Annex E allows for a closed-loop transfer function (CLTF) to be performed in lieu of the unbalance verification test for model-validation purposes. However, an unbalance verification test is still sometimes performed to attempt to validate the robustness of the system critical speeds. The closed-loop transfer function is shown in Figure 2, and it is also described in detail in ISO 14839-3. In this simple magnetic bearing closed-loop model, the sensor, actuator, and amplifier dynamics are included in the plant.

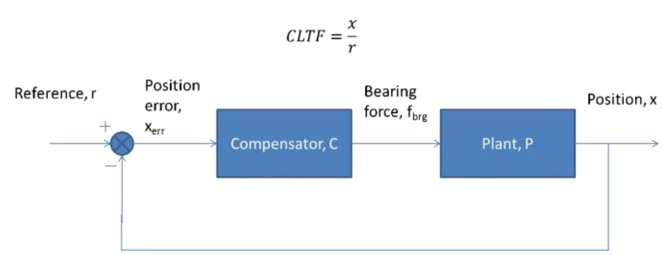

Figure 2. Model of CLTF allowed in API 617 8th ed. Annex E in lieu of unbalance verification test; measurement originally described in ISO 14839-3

ISO 14839-3 requires the evaluation of stability on the basis of the SNTF. The SNTF of a feedback control-loop system is a measure of the robustness of the system to changes in the plant. For a typical feedback control system consisting of a plant and compensator, the SNTF is described by Figure 3.

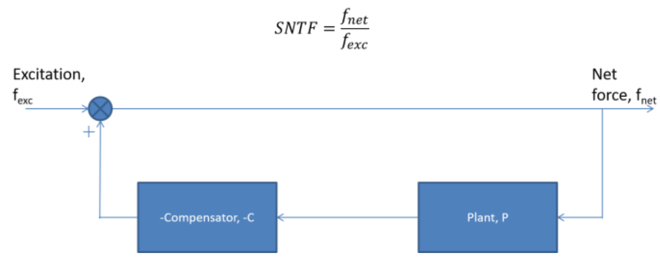

Figure 3. Definition of SNTF for a standard control system

Notably, ISO 14839-3 gives the following criteria regarding the SNTF:

1.) The SNTF gain is usually less than 3.0 (9.5 dB) at all frequencies for newly commissioned machines. This range of SNTF values is referred to as Zone A.
2.) SNTF gains between 3.0 (9.5 dB) and 4.0 (12 dB) are considered acceptable for long-term operation. This range of SNTF values is referred to as Zone B.
3.) SNTF gains above 4.0 (12 dB) and below 5.0 (14 dB) are generally considered unsatisfactory for long-term operation. This range of SNTF values is referred to as Zone C.
4.) SNTF gains above 5.0 (14 dB) are severe enough to cause damage to the machine.

ISO 14839-3 also defines the open-loop and closed-loop transfer functions but sets no hard criteria for their evaluation. Typically, the maximum frequency analyzed/measured for all transfer functions corresponds to the bandwidth limitations of the controller.

C. An overview of hydrodynamic bearing dynamics and active magnetic bearing dynamics

This section provides a general comparison of the dynamics of hydrodynamic bearings and active magnetic bearings. Hawkins et al. [4] provide a more detailed description and comparison of the dynamics of hydrodynamic bearings and active magnetic bearings.

The rotordynamic coefficients of most hydrodynamic bearings can be modeled using a spring-damper (K-C) system, wherein the force generated by the bearing is a function of the relative displacement and velocity between rotor and stator. Equation 4 shows the *bearing transfer function* (Bearing TF) of a typical hydrodynamic bearing along one lateral axis:

$$\frac{F_{brg}}{x} = K + Cs \tag{4}$$

Many studies have also shown that including an *inertia* (or *added mass*) term (force proportional to relative acceleration) can be important in characterizing the high-frequency behavior of hydrodynamic bearings. The inertia coefficient is not included in the development of this analysis.

The rotordynamic coefficients of magnetic bearings are determined by the selected control architecture and corresponding control law for the system. The simplest implementation of a magnetic bearing control architecture is a proportional-integral-derivative (PID) controller along each radial axis. The transfer function of a PID controller is shown in Equation 5.

$$Compensator\ TF = K_p + K_d s + \frac{K_i}{s} \tag{5}$$

Note that the proportional and derivative terms of the AMB PID transfer function behave dynamically similarly to the stiffness and damping terms of a hydrodynamic bearing. The integral term serves to stabilize the low-frequency behavior of the system by removing any steady-state error.

Most modern magnetic bearing control architectures will employ a ratio of higher-order polynomials in lieu of the derivative term. Specifically, AMB systems will commonly use a steady-state gain, an integral term, and a ratio of *n*th order transfer functions, as shown in Eq. (6). The orders of the numerator and denominator need not be equal; the order of the numerator must be equal to or less than the order of the denominator, such that the gain at high frequencies decreases with frequency. If the order of the numerator were larger than that of the denominator, the gain would continually increase with frequency, causing high-frequency noise to be amplified. Note that the proportional term can be added in series as shown in Eq. (5), or in parallel, as shown in Eq. (6).

$$Compensator\ TF = K_p * \frac{(b_n s^n + \cdots + b_2 s^2 + b_1 s + b_0)}{(a_n s^n + \cdots + a_2 s^2 + a_1 s + a_0)} + \frac{K_i}{s} \tag{6}$$

In the system described above, all gains and coefficients are usually chosen to allow for robust closed-loop control, low response resulting from forced vibration, low response when traversing critical speeds, and noise immunity. The coefficients of the *n*th order polynomials are generally calculated by combining several first and second order filters, such as lead, lag, lead-lag, low-pass, band-stop, and band-pass filters.

D. Bridging the gap between controls and vibrations: A comparison of SDOF dynamics for a KCM system and a PD motion control system

This analysis presents the similarities and differences between the two stability criteria applied in different situations. To gain a better appreciation for the similarities between the two criteria, the two criteria will be compared when applied to the forced response of a single-degree-of-freedom (SDOF) dynamic system. Figure 4 shows a SDOF K-C-M system subjected to an external force, F_{exc}.

The stiffness and damping are representative of a bearing/compensator and the mass is representative of the plant, usually the rotor for rotating machines. The external force is representative of a dynamic force applied to the plant in addition to the bearing force, usually an unbalance vector or external dynamic load.

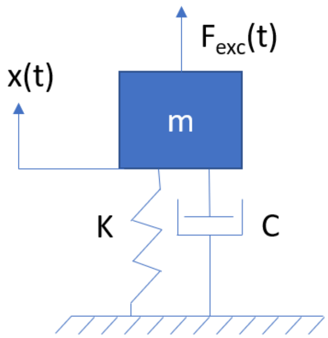

Figure 4. SDOF forced spring-mass-damper system

The ratio of the displacement of the mass, x, to the external force, F_{exc}, can be represented by:

$$\frac{X}{F_{exc}} = \frac{1}{ms^2+cs+k} \tag{7}$$

In the typical frequency response analysis of a SDOF K-C-M system, F_{exc} is a constant-amplitude sinusoidal function:

$$F_{exc} = A_f \cos(w_f t) \tag{8}$$

The steady-state response amplitude of $x(t)$ varies with the excitation frequency, w_f, and the peak response occurs at the damped natural frequency, w_d.

Though not immediately obvious at first, a dynamically-equivalent active control system model can be built to control the motion (displacement) of the mass using a standard PD controller, rather than a spring and a damper, as shown in Figure 5.

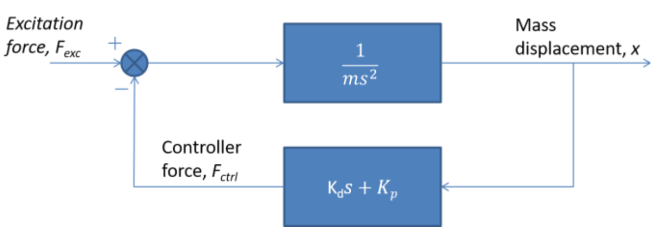

Figure 5. PD motion (displacement) control system subjected to external force

In a PD motion control system, the proportional term produces a control force proportional to displacement. The derivative term produces a control force proportional to velocity. As such, the K_p and K_d terms of a PD-based motion control system, behave dynamically similar to the K and C terms, respectively, of a SDOF K-C-M vibratory system. Because the input to the plant in Figure 4 is a force excitation, this transfer function is also known as the *load disturbance transfer function*. The load disturbance transfer function of a PD motion control system can be expressed as:

$$\frac{x}{F_{exc}} = \frac{1}{ms^2+K_d s+K_p} \tag{8}$$

Vance [5] defines a critical speed as a speed at which a local maximum of the synchronous response exists. The main external force that causes synchronous motion (and thus critical speeds) of a rotor is usually the unbalance force. It is widely known that the amplitude of the rotor unbalance force is proportional to the square of the rotor speed, w_s, as shown in Eq. (9).

$$F_{unb} = mew_s^2 * \cos(w_s t) \tag{9}$$

This represents a major deviation from standard SDOF system dynamics theory. A result of this phenomenon is that a critical speed will not directly correspond to w_d. Rather, the peak response will always occur at a speed above w_d. This phenomenon also indirectly results in other apparent deviations from standard SDOF dynamics. Notably, an increase in bearing damping will result in a decrease of w_d, but an increase in the critical speed.

It is important to understand this distinction because API 617 calculates AF based on the synchronous response of the rotor, rather than the single-axis forced response (wherein the excitation force has a constant amplitude vs. frequency). The CLTF and SNTF from ISO 14839 are calculated using a single-axis forced response with a constant-amplitude excitation force.

E. *Comparison of damping ratio, AF, CLTF, and SNTF for SDOF dynamic system*

A comparison of the calculated damping ratio, AF, peak CLTF, and peak SNTF is presented for the SDOF control system shown in Figure 4. Note that the AF is calculated in two ways in this analysis:

1.) Using a single-axis excitation with constant force amplitude
2.) Using an excitation with the amplitude of the force varying with the frequency squared, to mimic an unbalance response

A comparison is presented for damping ratios of 15.0%, 17.5%, and 20.0%. Table 1 summarizes the characteristics of the system for the various SDOF simulation cases.

Table 1. System characteristics for SDOF dynamic control system simulation cases

Case #	Damping Ratio	Mass (m), lb	Stiffness (k), lbf/in	Damping (c), lbf-s/in
1	0.150	50	100,000	45.5
2	0.175	50	100,000	39.8
3	0.200	50	100,000	34.1

Figure 6 shows the bearing transfer function for the 3 cases under consideration. As the damping decreases, the magnitude of the gain and phase plots decreases. A positive damping corresponds to a positive phase difference between the bearing force and rotor-bearing relative displacement. The magnitude of the effective stiffness corresponds to the magnitude of the bearing TF.

Figure 7 shows the load disturbance TF for the SDOF system. When using this TF to calculate the AF, case 1 produces an AF of 2.27, case 2 produces an AF of 2.72, and case 3 produces an AF of 3.26. This TF is not generally used to calculate AF but is a common TF used in the evaluation of AMB-supported systems.

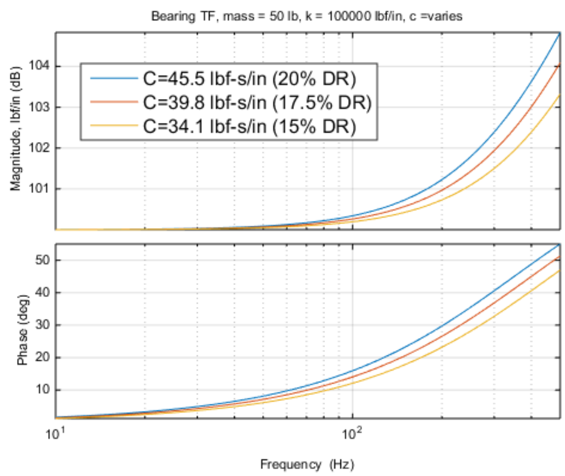

Figure 6. Bearing TF for all 3 SDOF cases

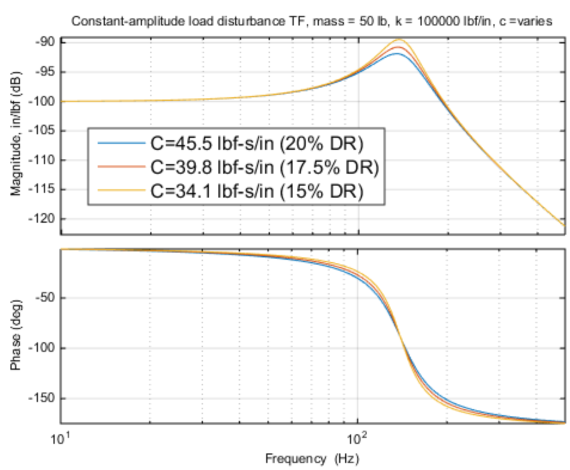

Figure 7. Load Disturbance (constant amplitude force) TF of SDOF dynamic system

Figure 8 shows the "unbalance response" transfer function for the SDOF system. This transfer function employs a force amplitude that is proportional to the square of the frequency. When using this TF to calculate the AF, case 1 produces an AF of 2.08, case 2 produces an AF of 2.48, and case 3 produces an AF of 2.98. For case 1 and 2, no separation margin to the mode would be required per API 617. If the AF and SM requirements of API 617 are applied to the unbalance response TF for case 3, there would be an 8.4% or 18.4% separation margin required between the operating speed range and the mode, depending on whether the mode is subsynchronous or supersynchronous, respectively.

Note that the AFs are always lower when using the unbalance TF compared to the load disturbance TF. As such, the load disturbance transfer function is always more conservative than the unbalance response TF. Most notably, in the case of a SDOF system mode with a 17.5% damping ratio, the unbalance response TF has an AF below 2.5 while the load disturbance TF has a AF above 2.5. When applying API 617 criteria to the unbalance response TF, the mode requires no separation to the operating speed range. If API 617 stability criteria were to be applied to the load disturbance response TF (which is not the usual case), there would be a 3.1% or 13.1% separation margin required between the operating speed range and the mode, depending on whether the mode is subsynchronous or supersynchronous, respectively.

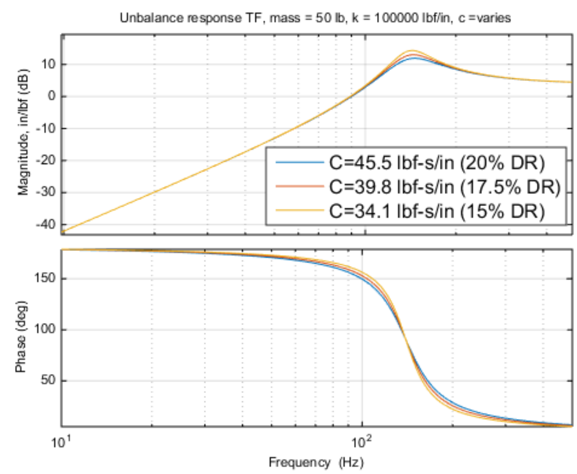

Figure 8. "Unbalance response" (force amplitude varies with square of frequency) TF

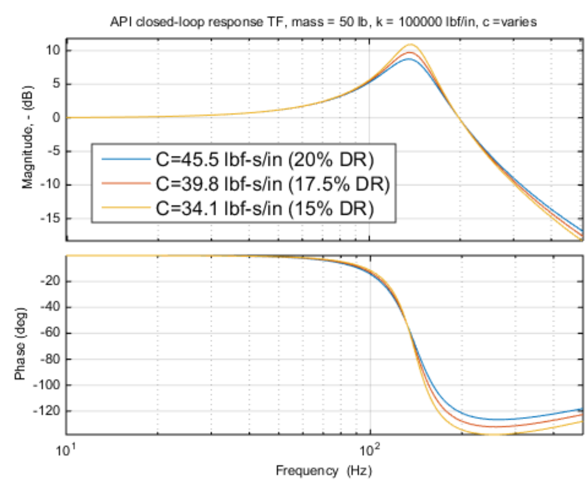

Figure 9. CLTF of SDOF dynamic system

Figure 9 shows the CLTF for the SDOF system. Neither criterion places an amplitude limit on the closed loop transfer function. API 617 gives an option to use this TF for model verification. This is generally a quicker test than the unbalance response verification test as it requires no special setup or instrumentation, beyond being able to spin the unit to full speed. Note that the CLTF has a similar shape as the load disturbance TF for a SDOF system, as both are closed-loop transfer functions employing constant-amplitude excitations.

Figure 10 shows the sensitivity TF for the SDOF system. As stated previously, this TF is described in ISO 14839-3 and is the main criteria used to evaluate stability in that standard. The results of the SDOF system show that there is a peak sensitivity of 8.12 dB for case 1, 9.25 dB for case 2, and 10.6 dB for case 3. As such, cases 1 and 2 fall under zone A of the

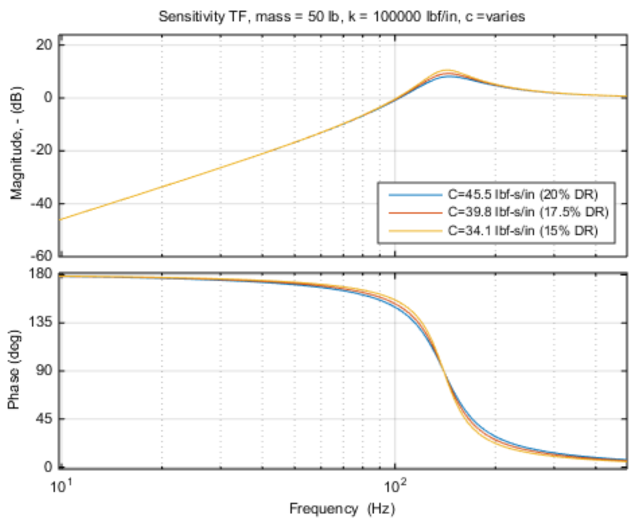

Figure 10. Sensitivity TF of SDOF dynamic system

Table 2 summarizes the results of the SDOF stability analysis for all 3 cases.

Table 2. Results of SDOF stability anlaysis

Case #	AF from Unbalance Response TF, -	Required SM (subsynch., supersynch.), %	SNTF peak gains, -	ISO 14839 SNTF zone
1	2.08	0,0	8.12	A
2	2.48	0,0	9.25	A
3	2.98	8.4,18.4	10.6	B

For cases 1 and 2, applying ISO 14839-3 to the results of the sensitivity TF yields relatively similar results to the application of API 617 to the AFs calculated from the unbalance TF. Cases 1 and 2 show acceptable stability; per ISO 14839-3, the peak sensitivity gains would be considered acceptable for newly commissioned machines (zone A).

Case 3 shows a sensitivity gain that is considered unusually high for newly commissioned machines (zone A), but acceptable damping for long-term field operation (zone B), according to ISO 14839. API 617 may be considered more restrictive for case 3 because it specifies a restriction on the separation margin, whereas ISO 14839 allows for long-term operation with no design changes required.

F. *Comparison of damping ratio, AF, CLTF, and SNTF for higher-order bearings*

Consider the case where the PD controller in Figure 5 is replaced with a higher-order transfer function. This is the typical case for most AMB systems. In this section, the damping ratio, AF, CLTF, and SNTF will be compared for a single 40 lb mass controlled by a compensator than can be represented as a ratio of higher-order polynomials, as shown earlier in Eq. (6). For this analysis, the numerator is a 24th order polynomial and the denominator is a 25^{th} order polynomial. Table 3 summarizes the coefficients used for this analysis.

Table 3. Coefficients of AMB bearing transfer function used for this analysis. Note that the transfer function is the ratio of force to displacement, in units of lbf/in

n	Numerator, b_n	Denominator, $a_{n,}$
25	N/A	1
24	6.797e09	1.578e04
23	5.312e13	1.236e08
22	3.405e17	7.184e11
21	1.533e21	3.184e15
20	5. 282e24	1.117e19
19	1.46e28	3.181e22
18	3.196e31	7.382e25
17	5.499e34	1.392e29
16	7.526e37	2.136e32
15	8.345e40	2.684e35
14	7.628e43	2.787e38
13	5.825e46	2.41e41
12	3.748e49	1.749e44
11	2.045e52	1.072e47
10	9.484e54	5.575e49
9	3.742e57	2.468e52
8	1.253e60	9.311e54
7	3.547e62	2.988e57
6	8.407e64	8.114e59
5	1.643e67	1.843e62
4	2.587e69	3.436e64
3	3.174e71	5.089e66
2	2.909e73	5.648e68
1	1.878e75	4.186e70
0	7.058e76	1.544e72
Proportional term	32,000	N/A

Figure 11 shows the magnetic bearing TF. Note that, as with the bearing TF for a standard K-C model, a positive phase difference between force and displacement represents positive damping. Note that with a magnetic bearing, there is usually always a region of negative damping (negative phase). Also note that the magnetic bearing TF can feature abrupt changes in stiffness versus frequency, as shown.

Figure 12 shows the unbalance response TF, showing two distinct critical speeds and a third critical speed composed of multiple local response peaks. The AFs of the critical speeds are shown, as calculated from by the half-power method shown earlier in Figure 1. Note that the half-power point corresponds to the point at which the peak response decreases by 3 dB (0.707 = -3 dB).

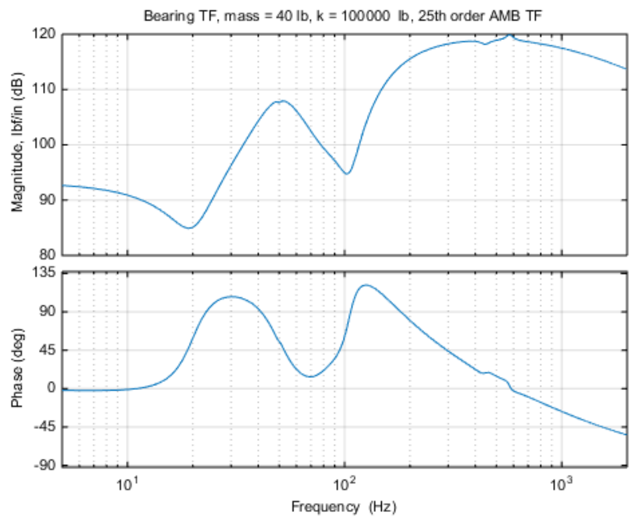

Figure 11. Magnetic bearing TF used for AMB-supported mass dynamic system analysis

The unbalance response shows a critical speed at 20.5 Hz. The AF of this critical speed is 2.27. API 617 concludes that this critical speed is well-damped and does not require any separation to the operating speed range.

The unbalance response also shows a critical speed at 99 Hz. The AF of this critical speed is 2.64. Because this AF is larger than 2.5, separation is required for this critical speed to the operating speed range. If the mode is subsynchronous, 6.6% separation margin is required, per Eq. (2). If the mode is supersynchronous, 16.6% separation margin is required.

Lastly, the unbalance response shows three closely-spaced, local peaks centered around 497 Hz. API 617 does not specify what to do in the situation of closely-spaced peak responses, as this situation is seldom encountered when hydrodynamic bearings are used. This situation is often encountered when magnetic bearings are used, due to the presence of additional bearing "control", or "support", modes; these additional modes often stem from the zeros and poles of the compensator. The AF calculated from the peak at 497 Hz is 3.41. Based on this AF, the required separation margin is either 9.9% or 19.9% depending on whether the mode is subsynchronous or supersynchronous, respectively.

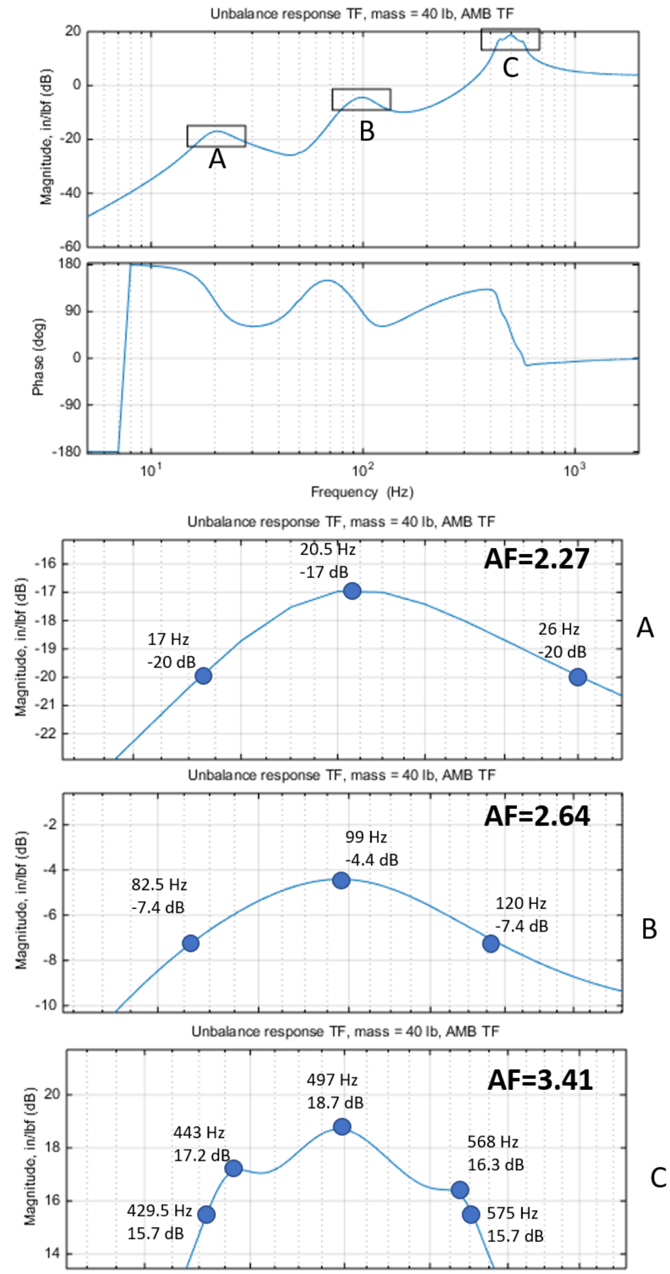

Figure 12. Unbalance response TF for AMB-supported mass dynamic system, with AFs calculated for each response peak

Figure 13 shows the SNTF for the AMB-supported mass. The results of the SNTF, as interpreted by ISO 14839-3, show that the closed-loop poles at 20.5 Hz and 98 Hz are very stable. This also confirms that the controller is quite robust to changes in the plant near the frequencies of these two poles. Both of these poles would under zone A ("newly commissioned machines") of the ISO 14839-3 criteria.

The SNTF gain in the regions near the 442 (13.3 dB) Hz and 566 Hz (12.6 dB) closed-loop poles falls into zone C ("unsatisfactory for long-term operation") of the ISO 14839-3 standard. This means that these poles are underdamped and will be sensitive to changes in the plant.

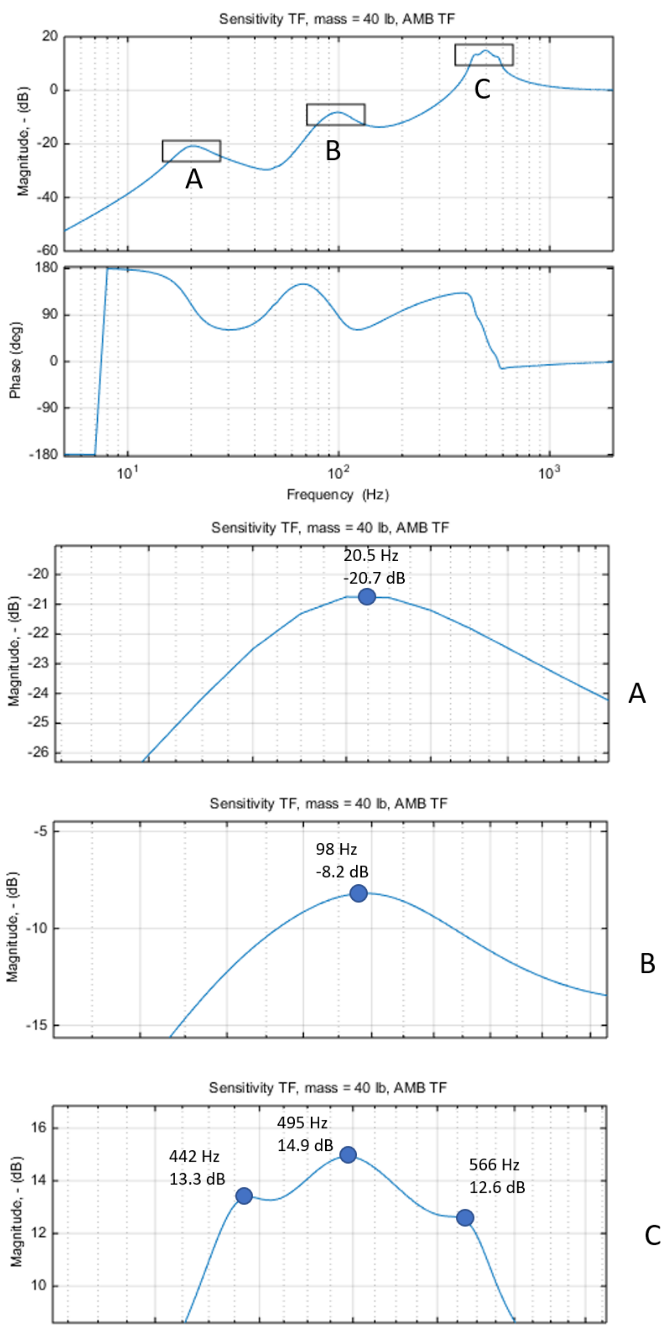

Figure 13. SNTF for AMB-supported mass dynamic system, with peak sensitivities shown

The SNTF gain in the region surrounding the pole at 495 Hz (14.9 dB) falls into zone D ("severe enough to cause damage") of ISO 14839-3. This means that this pole is marginally stable and will be very sensitive to changes in the plant.

Table 4 summarizes the stability results of this AMB-supported mass system.

Table 4. Summary of results of AMB-supported mass simulation

Local peak frequency (Hz)	AF, -	Interpretation of AF, per API 617	Peak SNTF gain, dB	Interpretation of SNTF, per ISO 14839-3
20.5	2.27	Stable	-20.7	Stable
98	2.64	Requires separation margin	-8.2	Stable
495	3.41	Requires separation margin	14.9	Not satisfactory

When comparing the two methods of evaluating stability, there are some differences in the interpretation of the stability of the various modes of this system:

1.) API 617 requires separation margin to the mode at 99 Hz, but ISO 14839 deems this mode acceptable without any change to the design.
2.) API 617 requires separation margin to the mode at 497 Hz, but ISO 14839 deems this mode to be unacceptable for long-term operation, regardless of separation margin.

Table 5 summarizes the closed-loop poles of this system associated with, or close in frequency to, the local peaks observed in Figure 11. As mentioned earlier, the closed-loop pole frequencies are not always equal to the critical speeds, because the unbalance force amplitude increases with frequency.

Table 5. Close loop poles and associated damping ratios of AMB-supported mass dynamic system

Closed-loop pole frequency, Hz	Damping Ratio, %
19.7	19.1
80.7	22.7
101	17.7
439	4.2
493	8.4
573	3.4

Figure 14 shows the load disturbance TF, using a constant amplitude excitation. As with the SDOF system, if the AF is calculated using this TF, the AF is larger than that calculated using the unbalance response TF. The load disturbance TF shows AF larger than 2.5 for all three local response peaks.

Figure 15 shows the CLTF, as defined in ISO 14839-3 and used in API 617. The closed-loop transfer function only shows a significant peak around the modes at approximately 500 Hz. The lower-frequency modes have a low closed-loop response.

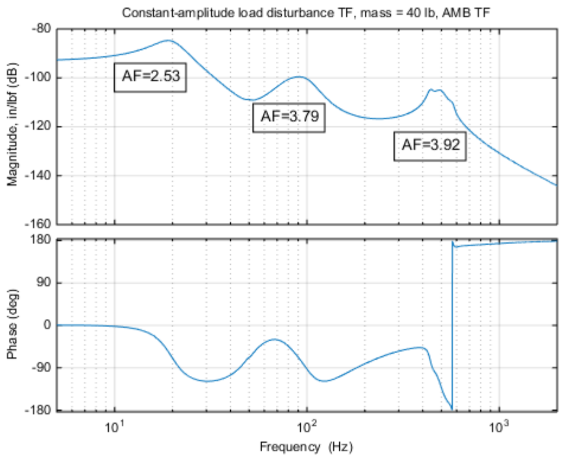

Figure 14. Load Disturbance TF for AMB-supported mass dynamic system

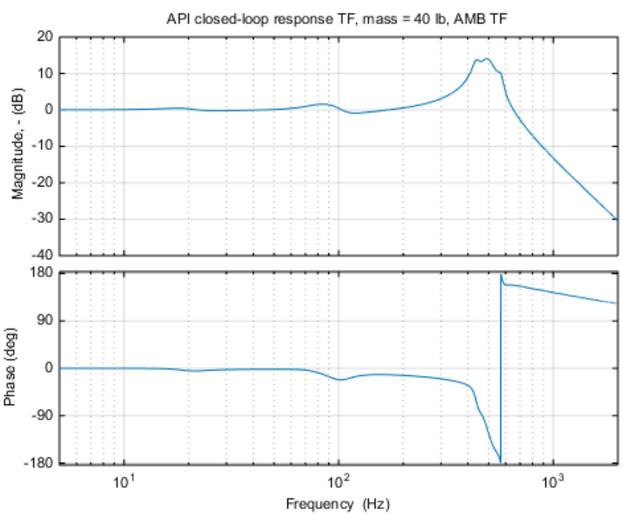

Figure 15. CLTF for AMB-supported mass dynamic system

Figure 16 shows the effect of varying the mass and the proportional gain on the unbalance response of the system. As can be seen, the modes close to 500 Hz are quite sensitive to these parameter changes. When decreasing the mass to 70% of the original value, the amplification factor of a mode close to 576 Hz increase to 19.2. When increasing the proportional gain by 30%, the amplification factor of a mode close to 568 Hz increases to 142! On the other hand, the two modes close to 20 and 100 Hz, respectively, are not significantly affected by the varied properties. Both of these modes retain their general response shape, implying that these modes are stable and robust to changes in the plant gain and have adequate gain margin.

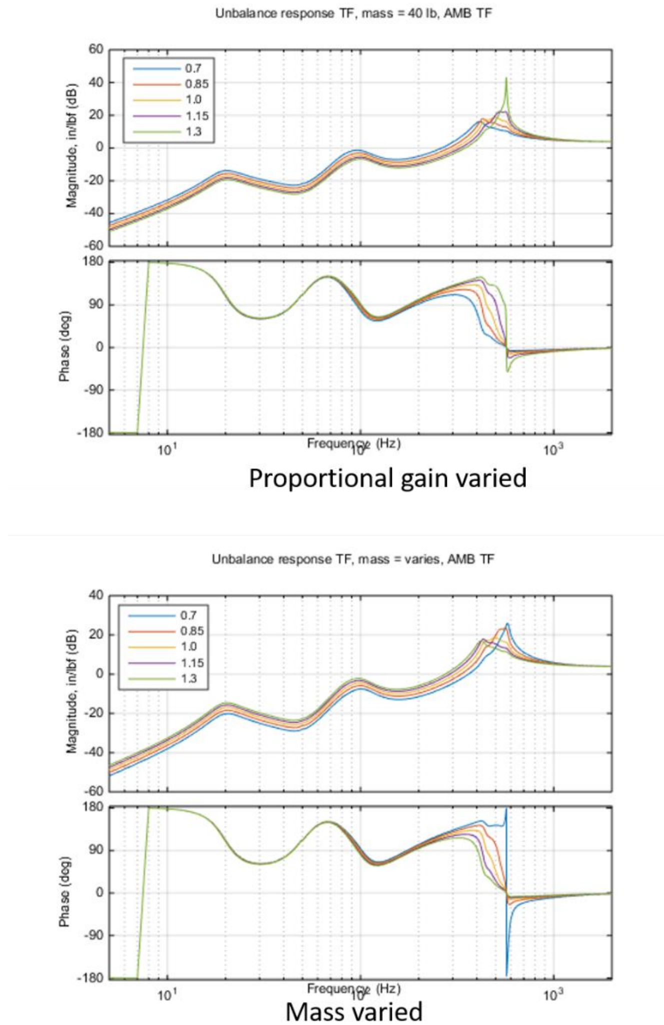

Figure 16. Unbalance response TF with proportional gain (top) and mass (bottom) varied

Based on the results in Tables 4 and 5, coupled with the results in Figure 12, 13, and 16, it can be concluded that API 617 and ISO 14839-3 both correctly evaluate the stability of the mode close to 20 Hz. However, the AF and SM requirements of API 617 are overly-restrictive in their evaluation of the mode close to 100 Hz, as Table 5 and Figure 16 both show that this mode has adequate damping, is robust to parameter changes, and has adequate gain margin. The SNTF requirement of ISO 14839 correctly assesses this mode as stable for long-term operation, and consistent with the stability of a newly-commissioned machine. The AF and SM requirements of API 617 also do not adequately assess the marginally-stable modes close to 500 Hz, as API 617 allows for operation near those modes provided there is a separation margin between the mode and operating speed range. The SNTF requirement of ISO 14839 correctly assesses that these modes are marginally-stable and not robust to parameter changes, and does not allow for long-term operation, regardless of any separation margin between the operating speed range and modes.

G. Conclusions

The results of this analysis show that the SNTF requirement, as described by ISO 14839-3 and the AF and SM requirements, as described by API 617 8th ed, may be comparable in evaluating the stability of dynamic systems supported by hydrodynamic bearings or magnetic bearings utilizing 1st-order (e.g. PID) control. Generally-speaking, the AF and SM requirements are more restrictive than the SNTF requirement for such systems.

Additionally, the results of the analyses described in this paper show that the SNTF requirement, as described by ISO 14839-3, is better-suited for evaluating the stability of an AMB-supported dynamic system utilizing high-order control. The SNTF can adequately assess the stability and robustness of closed-loop poles, regardless of their proximity to other modes or minor parameter changes in the plant. On the other hand, the AF and SM requirement, as described in API 617, was shown to be:

1.) A good evaluator of stability for some modes
2.) Overly-restrictive in its evaluation of stability for some modes
3.) A poor evaluator of stability of closely-spaced, lightly-damped modes

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