Unbalance Response Assessment of a Subsea Compressor

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Abstract

Robustness to unbalance degradation is examined for a subsea compression application involving a hermetically sealed, integrated motor-compressor supported on active magnetic bearings. The unique rotordynamic analysis methodology employs induced infinity norm principles to determine whether various limits, such as probe vibration, seal clearances and AMB force capacities, are exceeded for worst case unbalance distributions. The compressor's acceptability with respect to API unbalance response design criteria is examined. Additional results indicate that, by introducing speed-scheduled synchronous filter control, the machine should be able to handle significant balance degradation (approximately 7.5× the API allowable residual) before reaching the ISO recommended shutdown vibration levels.

For this particular design/application, it was determined that, with the nominal feedback controller, the unbalance capacity is dictated by the AMB dynamic capacity, not by probe vibration or seal clearance limits. Such knowledge can be vital to the end-user and manufacturer where, if unbalance degradation capacity is less than expected for the particular service, design changes can be made to help ensure that the machine's desired robustness and service life are achieved.

1 Introduction

Existing industrial standards, such as those from API and ISO, are designed to achieve high reliability levels in turbomachinery. To address API's unbalance response requirements for centrifugal compressors [1] and other capacity limits, three fundamental questions must be answered for a machine supported by active magnetic bearings (AMBs):

1. Are adequate separation margins maintained between the operating speed range and any well-amplified critical speeds and other natural frequencies?

2. Can the machine design handle twice the API allowable residual unbalance and not exceed the probe vibration limit and the AMBs' dynamic capacity?

3. When the rotor's balance state degrades to the probe vibration limit, are vibration levels along the rotor sufficiently low to avoid rubbing close clearances?

There is considerable field experience with critical, unspared machinery installed in process plants to support confidence in these design criteria. However, the remoteness and harsh environment associated with sea-floor operating environments dictates that subsea machinery must achieve reliability levels comparable to space applications. As a result, more stringent standards must be enforced for these subsea machines.

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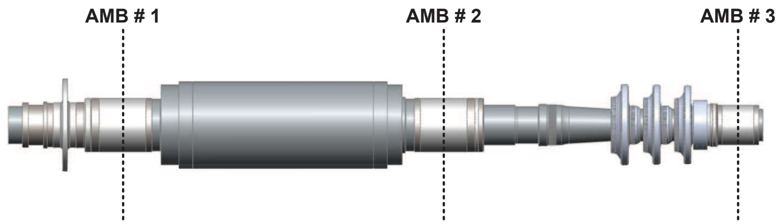

Figure 1: Schematic of integrated motor-compressor's rotor assembly

One area for additional assessment is the machine's robustness to unbalance degradation. A reality in any machine, unbalance degradation in a subsea application could be significant over the course of a typical five year desired service life and must be tolerated by the machine in order to avoid extremely expensive shutdown / repair cycles.

Given these reliability demands, several additional questions deserve scrutiny:

4. How much unbalance degradation can the machine handle before exceeding either the probe vibration or AMB capacity limits?

5. Using a combination of controls with and without synchronous filtering, how much unbalance can the machine handle that will allow it to freely operate across the entire speed range *and* maintain the probe vibration below ISO's [2] recommended shutdown limits?

This paper investigates these unbalance response issues for a specific subsea compressor application. The machine consists of a 10 MW hermetically sealed, integrated motor-compressor supported on three radial active magnetic bearings. Figure 1 presents the rotor assembly which includes three centrifugal impeller stages.

Separation margin aspects (question 1 above) are not examined in this paper, since this is generally a straightforward process. This investigation will focus on the remaining unbalance response issues that all deal with the machine's robustness to potential distributions of unbalance. To help ensure the worst case unbalance distributions are considered, a unique analysis methodology is employed and described in the next section.

2 Analysis Methodology

The objective is to determine whether or not the machine response exceeds some known limits. One such limit is a typical "probe vibration limit" (specified limit to vibration detected by the probes) but it is also possible to directly check seal or touch–down bearing clearance limits. In addition, because the machine employs magnetic bearings, it is important to consider the bearing forces and the bearing force slew rates (maximum rate at which bearing force can be changed) to be additional output signals of the system and to check them against known bearing capacity (static and dynamic) limits.

The potential distributions of mass unbalance are described in terms of bounds on the level of mass eccentricity (g-mm) at each of several planes along the rotor. As the actual distribution of mass unbalance is unknown, both in terms of actual magnitude at each plane and also in terms of relative phase, it is most important to check these machine response limits under the *worst case* unbalance distribution[1]. To answer this worst case unbalance response question, the formulation employed here closely follows that described in [3], which is based on singular value decomposition (SVD).

Conceptually, the machine's response to unbalance may be written as

$$\underline{y} = C(\Omega)\underline{u} \tag{1}$$

in which the response vector $\underline{y}$ lists the semi-major axes of the response at each critical clearance point, probe signal, and bearing force or slew rate; the unbalance vector $\underline{u}$ lists the phased unbalances at each unbalance plane. $C(\Omega)$ is then the machine's unbalance compliance function[2], which depends on the rotor speed, Ω. $C(\Omega)$ incorporates models of the rotor, casing, pedestal, any seals or other aerodynamic effects, and all of the components of the AMB system.

The unbalance bounds specification estimates that

$$|u_i| \leq U_i \tag{2}$$

in which the U_i are the unbalance bounds for each unbalance plane. In the same way, the clearance rub checks, probe vibration limits, and bearing capacity checks take the form

$$|y_i| \leq Y_i \tag{3}$$

in which the Y_i are the limit values. Thus, if for the worst case combination of u_i subject to $|u_i| \leq U_i$, we can determine that all of the y_i satisfy $|y_i| \leq Y_i$, then the machine's performance is satisfactory.

Equivalently, the machine's performance is acceptable if $|y_i/Y_i| \leq 1$ for any possible $\underline{u}$ subject to $|u_i/U_i| \leq 1$. The leads to the *normalized* response and unbalance vectors: $\hat{y}_i = y_i/Y_i$ and $\hat{u}_i = u_i/U_i$ which produces the associated notion of *weighting* matrices, W_y and W_u: $W_y\underline{\hat{y}} = \underline{y}$ and $W_u\underline{\hat{u}} = \underline{u}$ (which implies that $W_y = \text{diag}[Y_i]$ and $W_u = \text{diag}[U_i]$). Combining these weighting matrices with the machine's unbalance compliance function produces

$$\underline{\hat{y}} = W_y^{-1}C(\Omega)W_u\underline{\hat{u}} \tag{4}$$

and we can now define a *normalized* machine unbalance compliance function, $\hat{C}(\Omega) \doteq W_y^{-1}C(\Omega)W_u$.

Satisfactory unbalance performance of the rotor means that, for a worst case unbalance distribution, none of the elements of the response exceed the threshold levels. Through the normalization process, this means that, for any $\underline{\hat{u}}$ subject to the limitation that $|\hat{u}_i| \leq 1$, the machine response computed as $\underline{\hat{y}} = \hat{C}(\Omega)\underline{\hat{u}}$ is acceptable if $|\hat{y}_i| \leq 1$.

We can pose the problem in a standard mathematical formulation to take advantage of well defined solution techniques. Namely, the *infinity*[3] *norm* of a vector is defined as $|\underline{x}|_\infty \doteq \max_i |x_i|$. This means that the unbalance performance requirement can be written concisely as: if $|\underline{\hat{y}}|_\infty \leq 1$ for all possible $\underline{\hat{u}}$ subject to $|\underline{\hat{u}}|_\infty \leq 1$ then the machine response is acceptable. In standard mathematical notation, the requirement for satisfactory response is that

$$\sup_{|\underline{\hat{u}}|_\infty \leq 1} \left|\hat{C}(\Omega)\underline{\hat{u}}\right|_\infty \leq 1.0 \tag{5}$$

[1] In general, it is not possible to exactly bound the unbalance distribution so there are many approaches to establishing these bounds. One is to look at manufacturing tolerances to estimate potential levels of unbalance. Another is to look at component or full rotor balancing processes to determine acceptance levels. Either approach establishes initial unbalance bounds. For in-service bounds, experience with aging rotors may give expected bounds. Alternatively, tolerated levels of unbalance may be chosen as part of the acceptable service conditions.

[2] The unbalance compliance function computes the runout response to given levels of mass unbalance.

[3] In this context, *infinity* has a somewhat arcane mathematical significance but the exact meaning of the infinity norm is the largest magnitude of any element of the vector. Literally, the $n-$norm is $|x|_n = \sqrt[n]{\sum_i |x_i|^n}$ so $|x|_\infty = \lim_{n\to\infty} \sqrt[n]{\sum_i |x_i|^n}$

When a matrix is measured by examining the norm of the output vector for a given norm bound on the input vector, the measure is called an *induced norm*. In this case, since the two vectors are measured using an infinity norm, the measure of $\hat{C}(\Omega)$ is its *induced infinity norm* and we require, quite simply, that

$$||\hat{C}(\Omega)||_\infty \leq 1 \tag{6}$$

That is, we require that the induced infinity norm of the machine's normalized unbalance compliance function is less than 1.0. This gives a comprehensive simultaneous assessment of all of the outputs of interest from the machine due to the worst case unbalance distribution. Conveniently, the induced norm sought in (6) is even easier to compute than the SVD employed in [3]:

$$\text{if } G = [G_{i,j}], \text{ then } ||G||_\infty = \max_i \sum_j |G_{i,j}| \tag{7}$$

In all of the analyses of this investigation, various weighted unbalance compliance functions for the machine are constructed considering different norm requirements either for the unbalance or for the response and, in each case, it is sufficient to plot the induced infinity norm of the resulting normalized unbalance compliance function *versus* rotor speed, Ω. As long as this single number is less than 1.0, the machine meets the target objective. Unlike common practice, it is not necessary to look at multiple different response functions for multiple unbalance distributions and to then try to tease out the worst case and evaluate it: all of this is done at once by the induced infinity norm of the normalized unbalance compliance function. Although the mathematical name may seem obscure, it provides precisely the evaluation called for by the engineering objective. In order to avoid repeated use of the cumbersome phrase *induced infinity norm of the machine's normalized unbalance compliance function*, we will refer to this number in the remainder of this investigation as the *response norm*.

The analysis presented here considers response to rotor mass unbalance but other effects can be important to consider. Obviously, rotor bow is another boundable synchronous excitation and can be easily handled simultaneously with unbalance. More importantly, static load can substantially affect AMB dynamic capacity. The rotor studied here is vertical so gravity load is not an important source of static load. However, aerodynamic loading of the impellers could be significant, especially at off-design flow conditions where volute pressure distributions are not circumferentially uniform. In this case, a reliable bound on the static load must be computed and the resulting deflected rotor / casing shape determined. Given that the bearings implement integrators in their control, it is reasonable to assume no static offset at the *position sensor* locations but static offsets at other locations (AMB centerlines, touch–down bearings, seal and impeller clearances) will decrease the effective clearance available to accommodate unbalance excitation. In the same manner, the static load requires bearing reaction forces and these forces must be deducted from the nominal bearing capacity in determining remaining capacity for dynamic load management. So this static analysis needs to be done prior to the unbalance analysis and the clearances and AMB capacities adjusted accordingly. Finally, as required by API, clearances should be at the worst case expected condition of the rotor and casing: they will typically be significantly smaller than the cold clearances.

3 Results

When using the above analysis methodology, engineering decisions must be made to establish the response design limits as well the locations and limits for potential unbalance. For the subject compressor, the four unbalance planes selected are shown in Figure 2. These planes were selected because of their effectiveness in exciting the rotor assembly's lowest four free-free modes. The

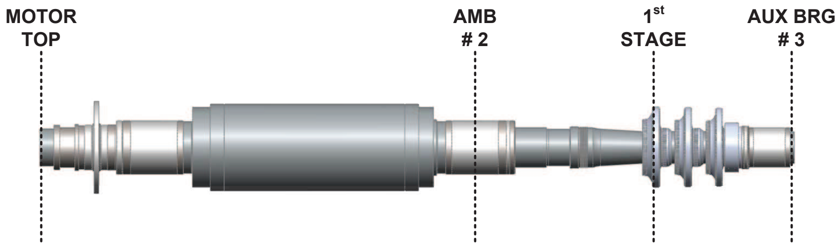

Figure 2: Unbalance planes selected

allowable unbalance at each plane is established based on the specific question being addressed. Of primary interest is their combined level relative to API 617's allowable residual unbalance U_{API} for the entire machine assembly ($\sum |u_i| \leq U_{API}$), which equates to an ISO 1940 [4] balance quality grade of G = 0.67.

The various response design limits are imposed depending on the question being addressed. These limits included vibration amplitudes at the probes, AMB dynamic capacities, and rub limits at critical clearance locations. Since a flexible casing and substructure models were incorporated in the analysis, all vibration responses were calculated based on the relative displacement between the rotor and the casing. When AMB dynamic capacities limits were imposed, they were established using each bearing's static force and slew rate limits.

With the exception of the separation margin considerations, each of the questions listed in the introduction will now be addressed for the subject subsea compressor. Those established by API 617 [1] will be addressed first.

3.1 API Questions

Question 2: Can the machine design handle twice the API allowable residual unbalance and not exceed the probe vibration limit and the AMBs' dynamic capacity?

At first glance, this question appears to consider realistic growth in unbalance due to field operating conditions and length of service. However, shop acceptance is the primary focus of this API design criterion, specifically: meeting the vibration requirement during the machine's mechanical run test. The 200% safety margin applied to U_{API} is meant to accommodate the additional unbalances encountered post-balancing and during the shop test. These additional unbalance sources can be from components that are not typically mounted during a rotor assembly's balance process (couplings, dry gas seals, etc.), as well as shrink fit adjustments that occur under centrifugal loading during the shop test.

For this application, the probe vibration limit in the operating speed range is governed by the ISO Zone A limit for new machines [2], rather than the API limit. Figure 3 presents the bounds established for the four selected unbalance planes.

The resulting response is shown in Figure 4. A broad peak in the response norm near 75% speed is associated with the rotor's first flexible mode which is well damped.[4] However, within the operating speed range, the response norm never exceeds approximately 35% of the limits, so this design criterion is passed with a substantial margin.

[4]This is a three bearing machine so the bearing system has an unusually high level of authority over the first bending mode, enabling a high level of damping and also a very substantial shift in the bending mode frequency. The first free–free bending mode is at about 42% speed but this frequency is moved up to 74% speed by the bearings.

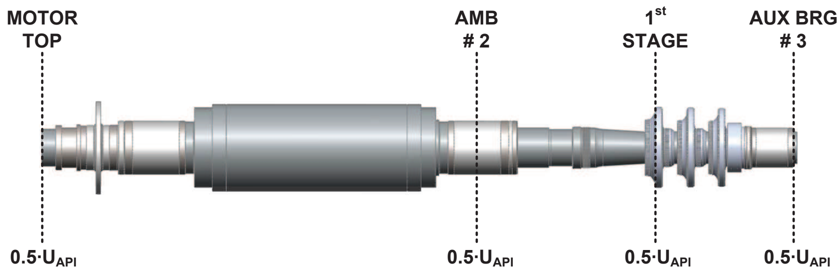

Figure 3: Unbalance plane bounds for $2 \times U_{API}$

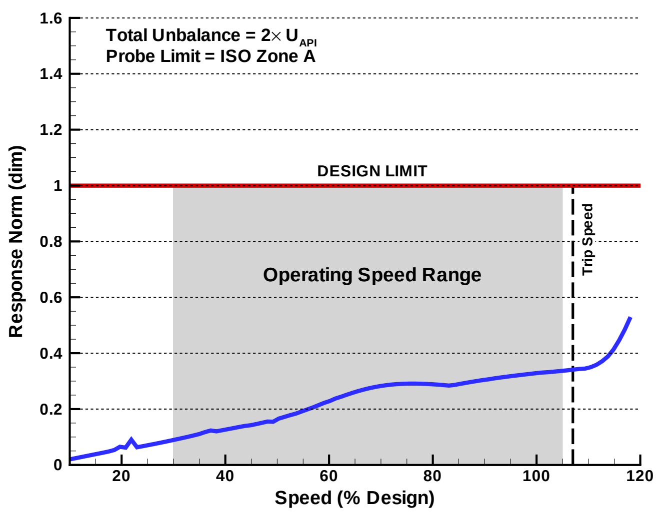

Figure 4: Unbalance response with $2 \times U_{API}$

Question 3: When the rotor's balance state degrades to the probe vibration limit, are vibration levels along the rotor sufficiently low to avoid rubbing close clearances?

Unlike the preceding question, unbalance degradation in the field is the primary consideration. Perhaps the most misunderstood API requirement, this criterion is meant to ensure that the probe vibration limit is truly a safe one that protects the machine. In other words, is it possible for unbalance degradation to lead to a rub somewhere in the machine without reaching a probe vibration limit?

Since unbalance degradation will most likely take place within the compressor section, the unbalance levels at the first stage impeller and touch–down bearing #3 were increased until the probe vibration limit (ISO Zone A) was reached at one of the probe locations. During this analysis, the other two planes' unbalance levels were left at their previous levels. In the end, it required 5.2 times the API allowable unbalance (equates approximately to an ISO G 3.5 quality grade) to reach the ISO Zone A probe vibration limit. Figure 5 presents the planes' bounds for this unbalance level.

With this level of unbalance defined, the design limits for this rub check analysis were revised to include only the critical clearances (touch–down bearings, AMBs, and seal labyrinths), excluding the probe limits and bearing capacities. Following API 617 requirements, 75% of the estimated minimum operating clearances were used as the response limits.

The resulting response versus speed is presented in Figure 6. From zero to trip speed, the response norm remains below the critical clearance limits imposed. As the rotor transverses its first flexible mode near 75% of the design speed, Figure 7 shows the normalized response values $|\hat{y}_i|$ for the individual critical clearance locations[5]. The greatest rubbing risk occurs at touch–down bearing #3 with the vibration displacement reaching almost half the bearing's clearance limit. However, it is clear that the rotor is in no danger of rubbing at any of the close clearances. Thus, the ISO Zone A limit for the probe vibrations is more than adequate to avoid rubbing.

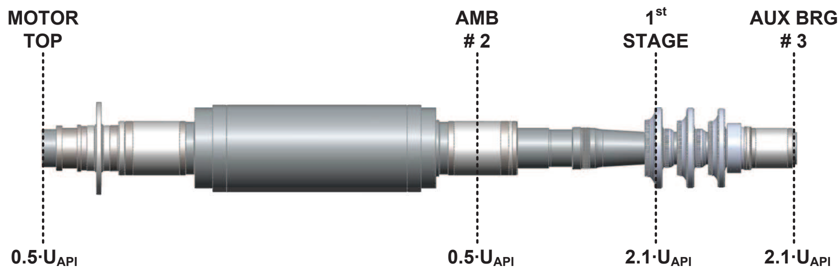

Figure 5: Unbalance plane bounds for critical clearance rub check

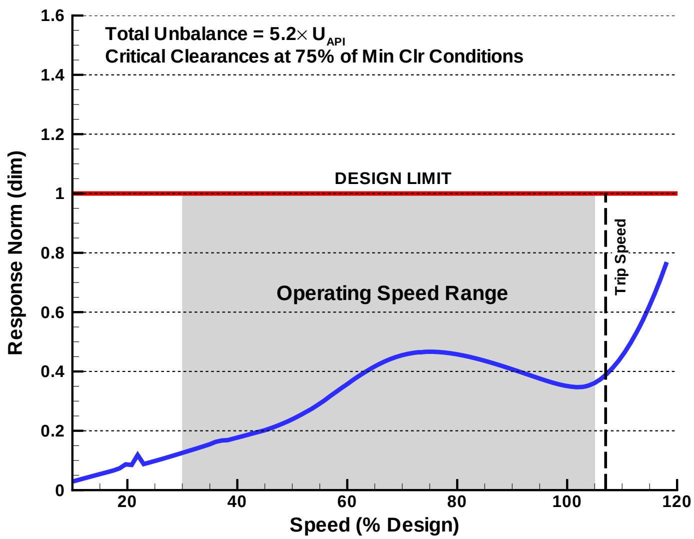

Figure 6: Unbalance response at critical clearances with unbalance high enough to reach the probe vibration limit

[5]The response norm value at AMB #3 probe location would be 1.0 because this level of unbalance was chosen to produce probe limit response at this speed. Figure 7 does not show any levels as high as 1.0 because no rub occurs and the figure does not include the probe signal.

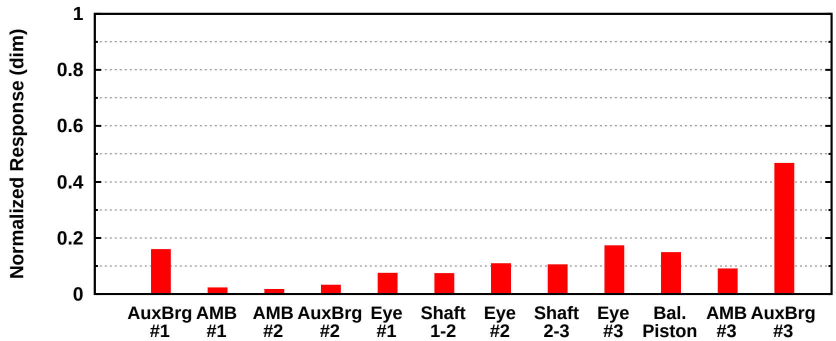

Figure 7: Critical clearances' normalized response values at 74% design speed

3.2 Additional Questions

Question 4: How much unbalance degradation can the machine handle before exceeding either the probe vibration or AMB limits?

As done for the last question's rub check, unbalance degradation was allowed at the first stage impeller and touch–down bearing #3 while keeping the other two planes' bounds at their previous levels. ISO Zone A probe vibration limits and AMB capacities were imposed for the response design limits.

To reach one of the design limits in the operating speed range, the amount of unbalance degradation necessary is 4.8 times U_{API} (equates approximately to an ISO G 3.2 quality grade). As indicated in Figure 8, this level of unbalance degradation achieves a unity response norm at approximately 100% of the design speed.

Figure 9 presents the normalized response values at this particular speed. Under this circumstance, the design limit is the slew rate capacity of AMB #3, which is reached before the probe vibration limit. The significance of this will be discussed later. However, it makes sense that AMB capacity establishes the limit since the amount of unbalance degradation ($4.8 \times U_{API}$) is less than that identified in the API rub check analysis ($5.2 \times U_{API}$) where only probe vibration limits were considered in determining limiting capacity.

Question 5: Using a combination of controls with and without synchronous filtering, how much unbalance can the machine handle that will allow it to freely operate across the entire speed range* and *maintain the probe vibration within ISO's Zone B limits?

Realistically, when the machine is installed subsea, it will be operated until the vibration levels reach the ISO recommended shutdown region (Zone B upper limit) [2]. Furthermore, the AMB control system will presumably implement synchronous filtering. The application of synchronous filtering by the AMB control system allows the rotor to spin about its principal axis of inertia, effectively eliminating the synchronous reaction forces at the bearings. Further details of this control technique can be found in [5].

In Figure 10, the response norms are plotted for the machine with the AMB control system active and also with the AMB control system suppressed, as would be the case when using a synchronous filter. In this analysis, the unbalance levels at the two sensitive planes (first stage impeller and #3 touch–down bearing) were increased until the two response norm curves intersected at a norm

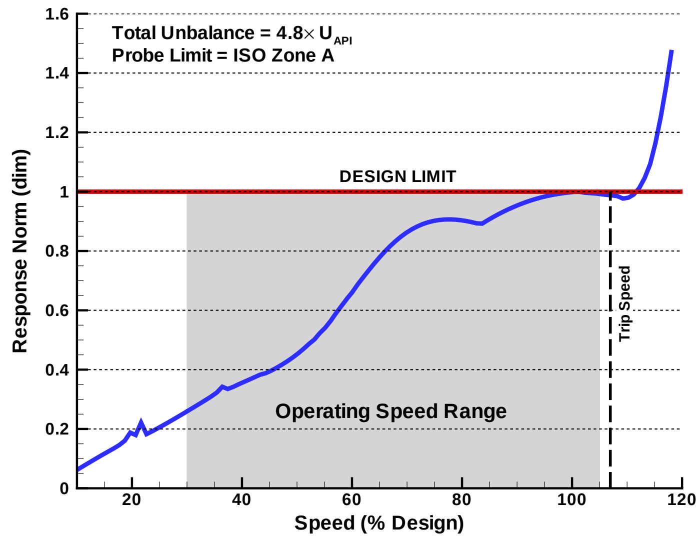

Figure 8: Unbalance response $4.8 \times U_{API}$

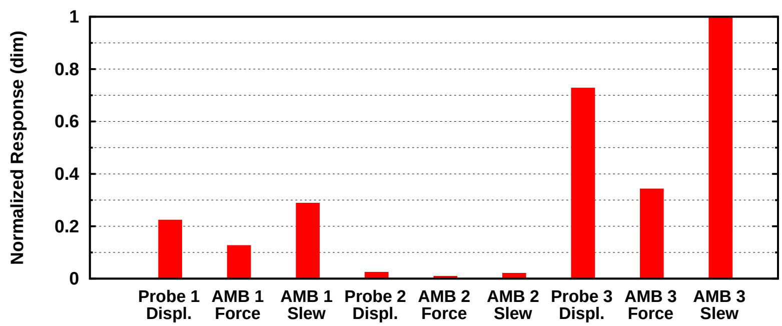

Figure 9: Normalized response values at 101% design speed

exceeding 1.0: the unbalance level at this point is 7.54× the API allowance (equates approximately to an ISO G 5 quality grade). Figure 11 shows the response norm that can be achieved by optimally switching from normal AMB feedback control to control including a synchronous filter. For the analysis depicted in Figure 11, it should be pointed out that the probe limits were the same both with and without the synchronous filter: commercial practice sometimes dictates that the probe limits are tighter with the synchronous filter engaged than when it is not engaged. This practice can be trivially accommodated in the present analysis by simply switching clearance limits when turning on/off the synchronous filter.

This result indicates that a synchronous filter can substantially increase the level of unbalance that the machine can tolerate. In particular, the limitations imposed by AMBs' dynamic capacities can be substantially mitigated using such a filter - a common motive for using such a filter.

These additional questions have assessed the machine's robustness to unbalance by quantifying how much degradation can be accommodated, and identifying which design limit governs the robustness. This enables the end-user and manufacturer to determine whether or not the machine's

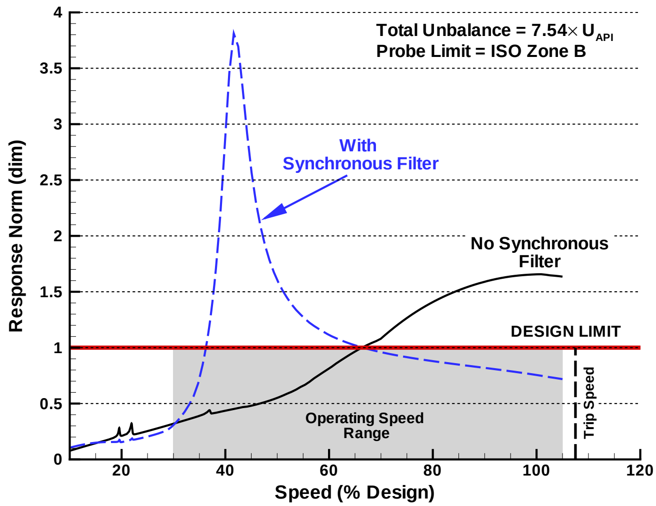

Figure 10: Unbalance response with $7.54 \times U_{API}$,
with and without a synchronous controller filter.

robustness is acceptable. The level of degradation expected in the field is determined by experience and highly dependent on the machine's design, operating conditions, and desired length of service.

For this particular subsea compressor, it was determined that, *without use of a synchronous filter*, the machine's robustness to unbalance is governed by AMB capacity, not by probe or seal clearance limits. This means that the unbalance level could grow to a point that exceeds the bearing dynamic capacity without producing a warning from the probe vibration limit. If this were to happen, then the radial vibrations might increase abruptly, suddenly causing a rub contact without warning from the probe vibration limit. Of course, the abrupt growth in radial vibration would likely produce a probe vibration limit warning during the event, but not in time to prevent it.

However, given that this problem arises at speeds well separated from any free–free flexible modes, there is no reason not to implement synchronous filtering. If this is done then, as indicated in Figure 11, there is no need to worry about rubs and, since the synchronous bearing forces are essentially zero at speeds above 67% of operating speed, there is also no need to worry about exceeding bearing dynamic capacity.

4 Conclusions and Recommendations

The main goal was to show that capacity analysis of AMB systems can be done comprehensively and efficiently using a fairly tractable analysis tool: the induced infinity norm of the weighted system response function. Using such a methodology, it is not only possible to determine whether or not the machine can manage expected synchronous loads, but it is also possible to determine how well balanced the bearing design is: what limits the performance? Ideally, bearing capacity should never be the limiting factor and, if the probe vibration limit is to be an effective safeguard against machine damage, then under no circumstances should either bearing or physical clearance limitations be exceeded before a probe vibration limit is reached.

In the example machine examined in this paper, it was determined that the nominal AMB controller would lead to bearing slew rate saturation (dynamic capacity limit) at a lower unbalance level

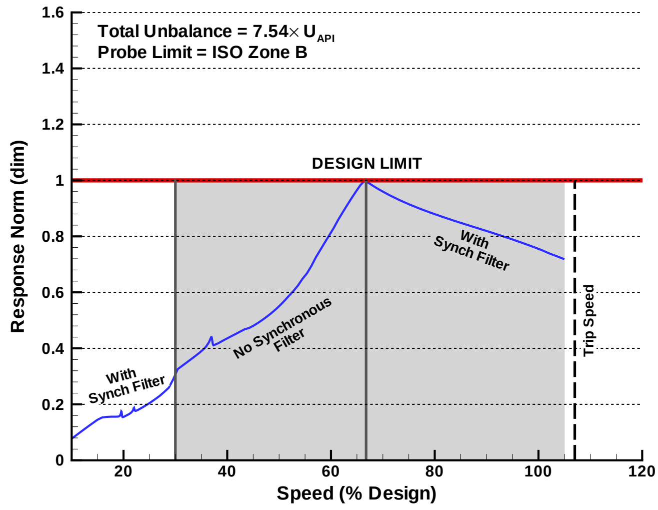

Figure 11: Unbalance response with $7.54 \times U_{API}$, with optimal controller switching between with and without a synchronous filter.

than would trip the probe limit. This would be an undesirable condition and the implication would be either that the controller should be revised or that the AMB winding count should be reduced and the associated power amplifier current capacity increased to protect bearing capacity while increasing available slew rate. However, implementing a synchronous filter accomplishes the same end with no need for engineering redesign: an acceptable outcome.

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