ROTOR RESPONSE AND SENSITIVITY

by

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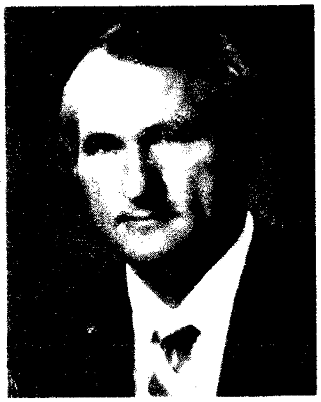

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ABSTRACT

Rotor response calculations are used to define peak response critical speeds and thus are a vital part of purchase specifications. To accurately calculate the range of critical speeds the rotordynamic analyses should consider all the variables that influence rotor response. These include bearing clearance, preload and oil temperatures and other parameters such as unbalance location and amplitude. The sensitivity of rotor response calculations to parametric changes in these variables is discussed in this paper. Procedures that are used to improve correlation between field measured data and rotor response calculations are reviewed and some general conclusions are presented.

I. Introduction

The unbalance response sensitivity of rotors is a major concern of both manufacturers and users of modern high speed equipment. Specifications such as API give recommended guidelines for the design of rotors with respect to balance, critical speed location, and vibration limits. Unbalance response analysis is an accepted and routine step in the design of safe and reliable rotating equipment due to the availability of computer codes for rotordynamic modeling of equipment.

The state of the art of rotordynamic analysis has advanced significantly in recent years; therefore, a fundamental understanding of the various parameters influencing rotor sensitivity is more important than ever. Calculation of rotor unbalance response can be useless or largely incomplete if certain variables such as unbalance location, bearing clearance, bearing preload, oil viscosity, etc. are not properly considered in the rotordynamic analysis. The existence of lateral critical speeds in the running speed range of rotors is too often unknown until field startup and operation of the unit. Detailed parametric analyses during the design stage can predict these problems and direct efforts for their solutions. The purpose of this paper is to illustrate how various design parameters and analysis techniques are important in the prediction and understanding of rotor response and sensitivity.

II. Rotordynamic Calculations

Using the definition as given in the API Dynamics section [1], "critical speeds correspond to resonant frequencies of the rotor-bearing support system. The basic identification of critical speeds is made from the natural frequencies of the system and of the forcing phenomenon. If the frequency of any harmonic component of a periodic forcing phenomenon is equal to, or approximates, the frequency of any mode of rotor vibration, a condition of resonance may exist; if resonance exists at a finite speed, that speed is called a critical speed".

To define these natural frequencies the following rotordynamic analyses need to be performed. First the critical speed map is generated. The bearing stiffness and damping characteristics are calculated and the stiffness values are plotted on the critical speed map. Finally the rotor response to unbalance is calculated. A stability analysis is normally performed for flexible shaft units that run above their first critical speed.

A. Critical Speed Map

The critical speed map is used to determine approximate locations of the natural frequencies of the system (undamped critical speeds). Figure 1 shows the critical speed map for a centrifugal compressor.

The specific bearing geometry (L/D, clearance, preload, oil viscosity, etc.) is used to calculate the bearing stiffness and damping coefficients as a function of rotor speed. The direct horizontal and vertical stiffness bearing curves are often superimposed on the critical speed map; the intersection of these bearing curves with the natural frequency curves defines the undamped critical speeds. There will be two intersections of the bearing curves with each critical speed curve corresponding to the horizontal and vertical undamped critical speeds. The vertical and horizontal intersections are sometimes called the major and minor critical speeds. Both vertical and horizontal peak responses can show up in an unbalanced rotor, especially for rotors installed with maximum bearing clearances and low damping.

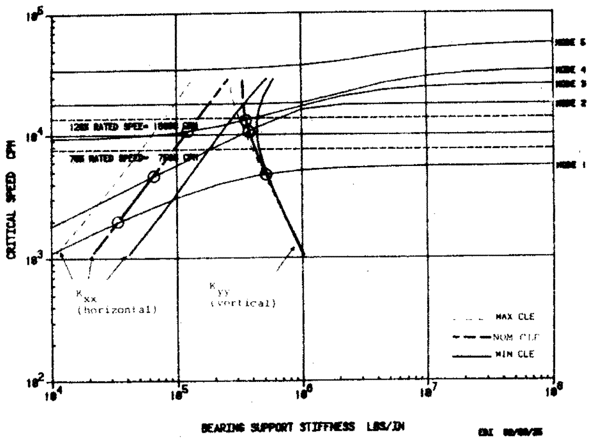

Figure 1 - Critical Speed Map for Centrifugal Compressor

Figures 2 and 3 give the critical speed mode shapes for the first two critical speeds for the compressor whose critical speed map is given in Figure 1 for a bearing support stiffness of 0.464 x 10^6 lb/in. This stiffness is approximately the vertical bearing stiffness. The first and second critical speeds for this stiffness are 4690 cpm and 11700 cpm. The first mode shape is a one-loop resonance with the nodes near the bearings. The second mode shape is a conical whirl mode shape with the node at the shaft center span.

The first three vertical undamped critical speeds for the nominal bearing clearances are 4550, 10600, and 13500 rpm. The horizontal undamped critical speeds for nominal bearing clearances are 2000, 4500, and 10500 rpm. These speeds are identified by the intersections circled in Figure 1.

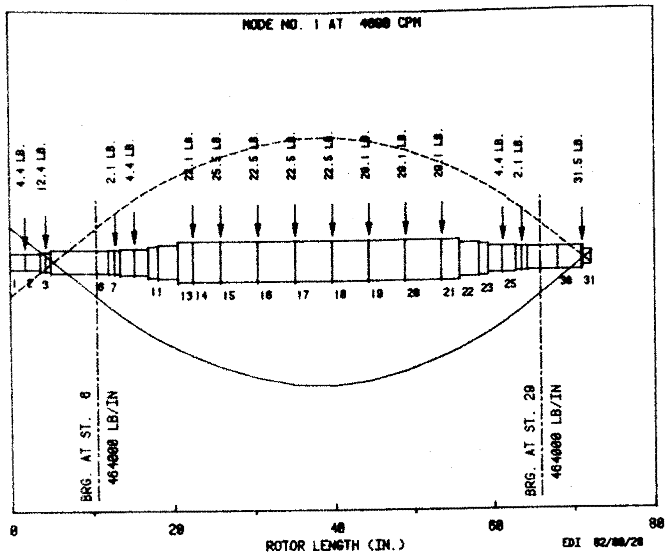

Figure 2 - First Undamped Lateral Mode Shape

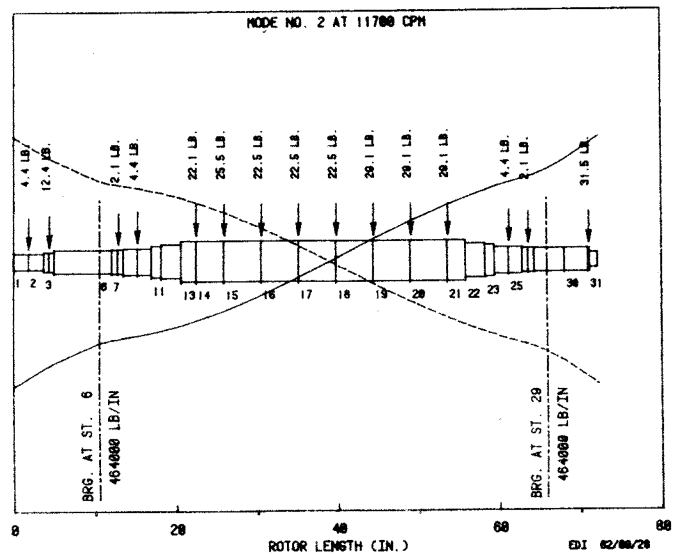

Figure 3 - Second Undamped Lateral Mode Shape

The closest correlation between the undamped critical speeds as determined from the critical speed map and those determined from the unbalance response analysis will occur on rotors with tilting pad bearings. For fixed geometry bearings such as cylindrical and pressure dam there can be significant differences due to the additional effects from the bearing cross-coupling terms.

It can be shown that damping increases the frequency of the peak response (N_{PR}) for the first lateral critical speed of symmetrical rotors [2]. In stability analyses, damping generally lowers the damped natural frequencies (instability frequencies, N_D). The equations that relate the peak response frequency and the damped natural frequency as a function of damping for a single degree-of-freedom system with rotating unbalance are given in Figure 4. The amplification factor

(A.F.) which is equal to $1/2\zeta$ is also presented on the horizontal scale where ζ is the damping ratio.

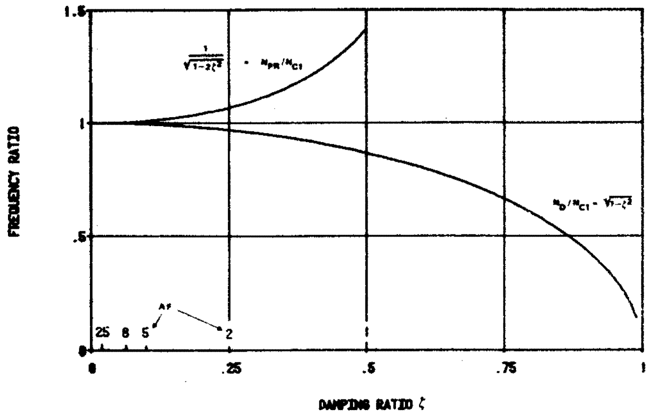

Figure 4 - Frequency Ratio versus Damping Ratio

It can be seen that for amplification factors higher than 5 the increase in the peak response natural frequency over the undamped natural frequency is less than 1 percent. For an amplification factor of 1, the peak response natural frequency for a single degree of freedom system is 41.4 percent higher than the undamped natural frequency (N_{C1}).

Since the amplification factor for the first critical speed in rotating machinery will be typically higher than 2, the peak response critical speed will normally be within 10 percent of the value read on the undamped critical speed map for a system with tilted pad bearings. Therefore, the critical speed map is a valuable tool in evaluating the locations of critical speeds of rotors.

B. Rotor Response to Unbalance

Computer programs are readily available today which allow calculation of the shaft orbit at any number of locations along the length of a rotor for various types of bearings, oil viscosity, pedestal stiffness, unbalance combinations, etc. [3]. A major difficulty is in determining or estimating the actual values of the input parameters to the program which will simulate the installed rotor's response and accurately predict critical speeds.

The API 617 specification for centrifugal compressors indicates that the "actual" peak response criticals are of concern rather than various calculated values. "Actual critical speeds are not calculated undamped natural frequencies but are critical speeds confirmed by test-stand data". In the design stage it is therefore necessary to know exactly the range of the peak response criticals. To do this, a computer program must be used which can accurately predict the peak response critical speeds. However, the prediction of the peak responses is dependent upon the assumptions made.

To predict the rotor response and actual critical speeds and to satisfy the Dynamics Section of the API specifications, it is necessary to calculate the vibration response of rotor due to anticipated unbalances. In the Vibration and Balancing Section 2.8.2, it states that the maximum allowable unbalanced force at any journal at maximum continuous speed shall not exceed 10 percent of the static loading of that journal. For symmetrical shafts the journal loading for each bearing would be one-half the rotor weight. For midspan unbalance the API residual limit is appropriate.

$$U_B \text{ in ounce-inches} = \frac{56347W_r}{N_{mc}^2}$$

where: N_{mc} = maximum continuous speed, rpm
W_r = rotor weight, lbs.

This unbalance results in a force equal to 10 percent of the rotor weight. Often one-half of this unbalance is applied at the coupling to excite the rotor. A moment unbalance can also be used to examine the sensitivity of the rotor to out-of-phase unbalances. For this case one-half the API residual unbalance is used at the coupling and another equal unbalance is used out-of-phase on the impeller or wheel furthest from the coupling or on the other coupling.

Since the most frequent problem resulting from unbalance is the excitation of high rotor vibrations, the predicted vibrations must be compared to applicable criteria. For example the API Dynamic Specifications allow a vibration amplitude of

Allowable test level (unfiltered) = Double amplitude including runout (mils)

= Vibration + Runout

$$= \sqrt{\frac{12000}{N_{mc}}} + 0.25\sqrt{\frac{12000}{N_{mc}}}$$

or 2 mils whichever is less.

API defines shaft runout as "the total indicator reading in a radial direction when the shaft is rotated in its bearings. If the vendor can demonstrate that "electrical runout" due to shaft material anomalies is present, the combined total mechanical and electrical runout shall not exceed 25 percent of the specified test level or 0.25 mil (6 micrometers) minimum, whichever is greater. Electrical runout can be deduced by slow-rolling the rotor in bearings or vee-blocks while measuring runout with a proximity probe and a dial indicator at the same shaft location".

It is difficult to define the absolute maximum vibration level that can be tolerated. The API allowable level is simply a guideline based on experience. The author has used the following guidelines for defining maximum acceptable vibration levels when the API allowables are exceeded. If the peak-peak vibration amplitudes are less than one-fourth of the bearing assembled diametrical clearance, the vibration amplitudes are acceptable. Vibration amplitudes greater than one-half the diametrical clearance are unacceptable and steps should be taken to reduce them.

$A < C/4$ Acceptable

$C/4 < A < C/2$ Marginal

$A > C/2$ Unacceptable

where: A = vibration amplitude, mils peak-peak
C = diametrical clearance, mils

As with most experience-based criteria, these allowable amplitudes are based upon the synchronous vibration component only. Whenever there are nonsynchronous vibration components present, this criteria may not be applicable.

In addition to the variation of the unbalance location, it is necessary to investigate the effect of changes in the bearing parameters on rotor response and amplitudes. The bearing stiffness and damping coefficients are calculated as a function of the Sommerfeld number.

$$S = \frac{\mu LDN\ D^2}{W_j\ C_d^2}$$

S = Sommerfeld Number
μ = Viscosity, reyns
D = Journal Diameter, in.
L = Bearing Length, in.
N = Speed, cps
W_j = Journal Load, lb.
C_d = Diametrical Clearance, in.

The bearing properties are strongly influenced by other geometrical considerations such as preload (m), offset, dimensions of the pressure dam, etc.

The normal procedure in a design audit is to calculate the bearing characteristics for the range of clearances, preload and oil temperatures. The maximum clearance, minimum preload and highest oil temperature usually define the minimum stiffness. The other extreme is to use the minimum clearance, maximum preload and the coldest oil temperature. This will typically define the range of expected stiffness and damping coefficients for the bearings. Figure 5 gives a summary of the bearing clearances and preloads that can be obtained by considering the range of dimensions on the shaft and bearing. When each of these are used with the combinations of unbalance, the anticipated range of rotor respose can be calculated. This is important since one part of the API specification (2.8.1.9) states: "If the lateral critical speed as calculated or revealed during mechanical testing falls within the specified operating speed range or fails to meet the separation margin requirements after practical design efforts are exhausted, the unit vendor shall demonstrate an insensitive rotor design. This insensitivity must be proven by operation on the test stand at the critical speed in question with the rotor unbalanced."

5 SLOP BEARINGS FOR CENTRIFUGAL COMPRESSOR
CLEARANCE VARIATIONS WITH TOLERANCES

DIMENSION	MINIMUM	MAXIMUM
BEARING BORE, INCHES	3.0040	3.0070
JOURNAL DIAMETER, INCHES	2.9990	3.0000
PAD CURVATURE (DIAMETRICAL), INCHES	3.0070	3.0090

CB MILS	CP MILS	PRELOAD (DIM)	BRG BORE INCHES	PAD BORE INCHES	JOURNAL OD INCHES
2.50	4.00	0.3750	3.0040	3.0070	2.9990
2.50	5.00	0.5000	3.0040	3.0090	2.9990
4.00	4.00	0.0000	3.0070	3.0070	2.9990
4.00	5.00	0.2000	3.0070	3.0090	2.9990
2.00	3.50	0.4286	3.0040	3.0070	3.0000
2.00	4.50	0.5556	3.0040	3.0090	3.0000
3.50	3.50	0.0000	3.0070	3.0070	3.0000
3.50	4.50	0.2222	3.0070	3.0090	3.0000

CB IS THE ASSEMBLED RADIAL CLEARANCE
CP IS THE MACHINED RADIAL CLEARANCE
M IS THE BEARING PRELOAD FACTOR

Figure 5 - Effect of Clearance Tolerances on Preload

In the design stage, it is not possible to know the exact installed configuration with regard to bearings (clearance, preload) and balance (location of unbalance). Usually the mechanical test will be limited to one configuration (clearance, preload, unbalance) which may not show any problem. Changes introduced later by maintenance during turnarounds may change sensitive dimensions which may result in a higher response. This is the reason some satisfactorily operating machines change vibration characteristics after an overhaul.

It is possible that the test stand data may not show all the peak response critical speeds for the following reasons:

(a) If the rotor is well balanced such that very little excitation is present, the vibrations will be low and the resonances so well damped that the peak responses cannot be identified.

(b) The rotor unbalance can be distributed such that the critical speed near the running speed range will not be excited. This occurs quite often for symmetrical rotors that are running near the second critical. If the unbalances are near midspan, the conical whirl mode will not be excited.

(c) The bearing selection on the test stand may result in a bearing having high stiffness and damping coefficients and thus the critical speed occurs at a higher speed and the high damping reduces the peak responses.

In many rotor response analyses, the second critical speed is well damped due to the conical whirl mode shape. Many times the second major peak response predicted is actually the third critical speed as determined from the critical speed map. The fact that a peak response is not predicted does not mean that the second critical speed has disappeared; it just means that the damping for the selected bearing parameters is sufficiently high that the peak responses are subdued. This again underscores the necessity of performing the rotordynamic analysis for the complete range of bearing clearances, preloads, oil temperatures and unbalance locations. If peak responses are predicted for the possible maximum clearance and minimum preload (minimum damping), a sensitivity test should be performed on the test stand as defined in API 2.8.1.9.

As presently written, the unbalance specified for the API sensitivity test is equal to 5 times the API residual unbalance limit. This would result in a transmitted bearing force of fifty percent of the rotor weight, or twenty-five percent on each journal for midspan unbalance; however, when it is placed at the coupling, only one-half of the 5 times amount can be used. For moment unbalances, one-half of the 5 times API residual unbalance is used at the coupling and at the other end of the rotor, out-of-phase. For unevenly loaded journals the journal loading nearest the unbalance locations should be used in determining the magnitude of the unbalance. The allowable vibration amplitude for the API sensitivity test is twice the API specified limit.

III. Sensitivity of Rotor Response

The critical speed map is used to determine approximate locations of the natural frequencies of the system (undamped critical speeds). The actual critical speed locations as determined from response peaks caused by unbalance are strongly influenced by the following factors:

(1) bearing direct stiffness and damping values
(2) bearing cross-coupled stiffness and damping values
(3) location of the unbalance
(4) location of measurement point
(5) pad load distortions
(6) pad thermal distortions
(7) bearing support flexibility
(8) oil film temperature and viscosity variations

There has been an increased awareness of the factors which affect rotor response characteristics. The recent paper by Caruso, et al [4] reviews the effect of several important parameters on the bearing stiffness and damping values. One important factor discussed was the temperature distribution effects on pad clearance and preload. Figure 6 shows a typical operating temperature profile for a tilting pad bearing. This temperature variation can cause an increase in the bearing preload as a function of speed. When the combined effects of support flexibility and pad distortion from load and temperature were considered for their machinery, the results shown in Figures 7 and 8 were obtained. These results indicate reductions in stiffness terms of nearly 40 percent and a reduction in damping by as much as 5 to 1.

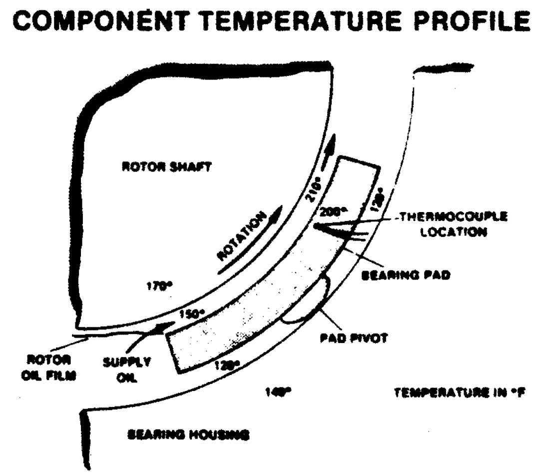

Figure 6 - Operating Temperature Profile for Tilt-Pad Bearing (Ref. 4)

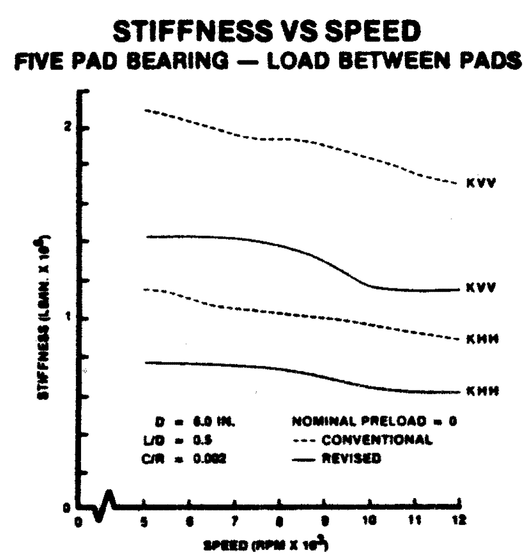

Figure 7 - Effects of Pad Distortion on Stiffness (Ref. 4)

Other investigators have studied this problem [5-11] and some conclude that the calculated damping and stiffness terms for bearings are usually higher than actually is realized in practice. Others [10] show good correlation between calculated and measured bearing stiffness and damping values.

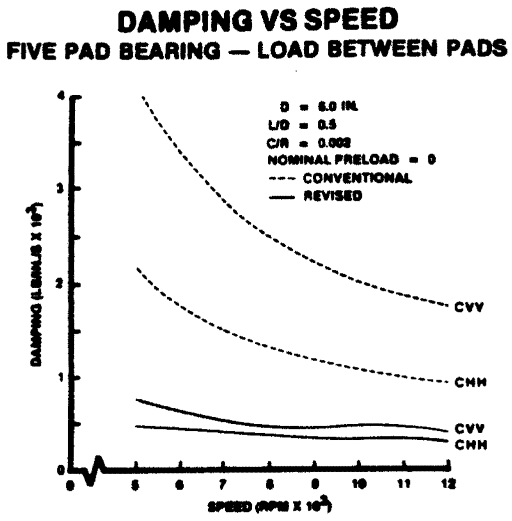

Figure 8 - Effects of Pad Distortion on Damping (Ref. 4)

In trying to solve rotordynamic problems it is necessary to model a system's critical speeds, amplification factor, and vibration amplitudes for a given unbalance. Once a machine has been modeled properly, possible changes can be studied to determine more accurately the effect of system changes.

It is quite a challenge to match the computer calculations with measured field vibration data. The following discussions of actual rotordynamic analyses point out the influence of the various parameters on rotor response and sensitivity.

A. Sensitivity to Bearing Clearances

To determine the peak response critical speeds for the compressor whose critical speed map is given in Figure 1, the responses due to coupling unbalance and midspan impeller unbalance were calculated. The API specified residual unbalance limit was calculated to be 0.25 in-oz. This unbalance was used at the coupling and midspan. The allowable API vibration amplitude for this compressor was 1.03 mils peak-to-peak (excluding runout) since its maximum continuous speed was 11300 rpm.

The computer response analyses for coupling unbalance are given in Figures 9a-c. For this compressor the vibrations were predicted at the two 45-degree probe locations for comparison with test stand data. The vibrations in the horizontal and vertical directions or any other direction can also be predicted.

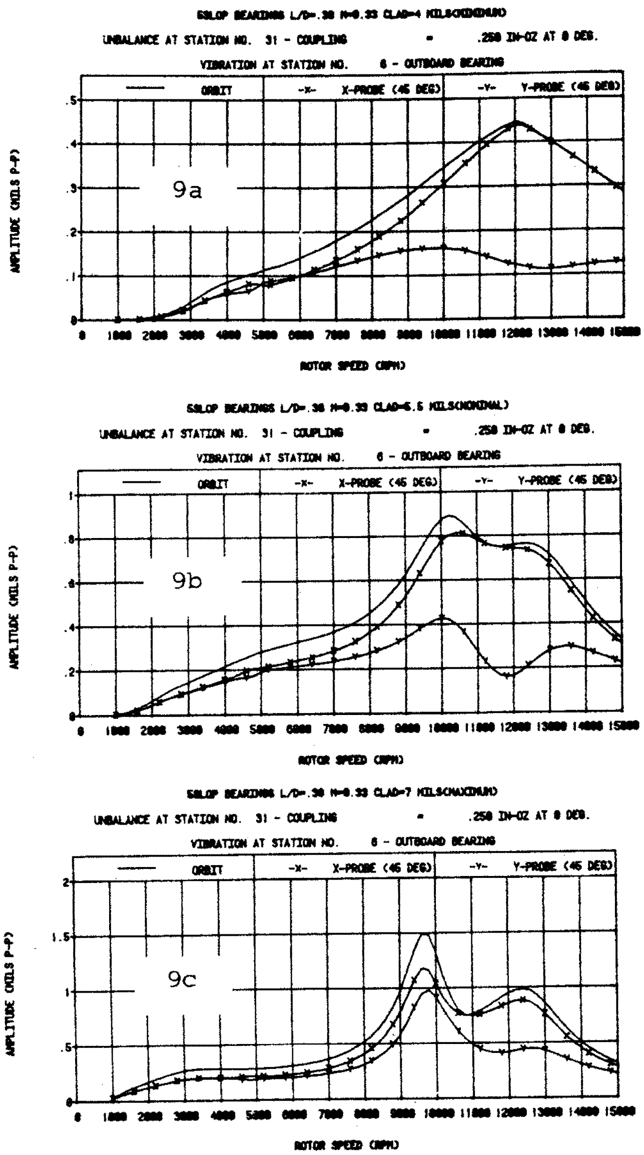

Figure 9 - Predicted Response at Outboard Bearing for Coupling Unbalance

The predicted response of the outboard bearing for coupling unbalance for minimum, nominal and maximum clearances are given Figures 9a, 9b, and 9c. The responses predicted in Figure 9a were the first critical speed at 4800 cpm (well damped) and another critical at 12000 cpm. Note that the Y-probe on the outboard has a peak response near 10000 rpm.

As the clearances increase, assuming that the preload remains constant at 0.33, the predicted peak response at 12000 rpm is lowered to 9700 rpm at maximum clearance (Figure 9c). The predicted amplitudes were less than the API limit for the nominal clearance (Figure 9b); however, when the maximum bearing clearance was used (Figure 9c), the responses became more pronounced and the predicted amplitudes exceeded the API limits.

Figure 10 shows the predicted unbalance

response for midspan unbalance for minimum and maximum clearances. For minimum clearance (Figure 10a), response is noted at the first critical speed near 4800 rpm with very little response at the second critical speed near 12000 rpm. However, for the maximum clearance (Figure 10b), the second critical speed at 9700 rpm becomes the predominant response. This again shows the sensitivity of a rotor to bearing clearance changes.

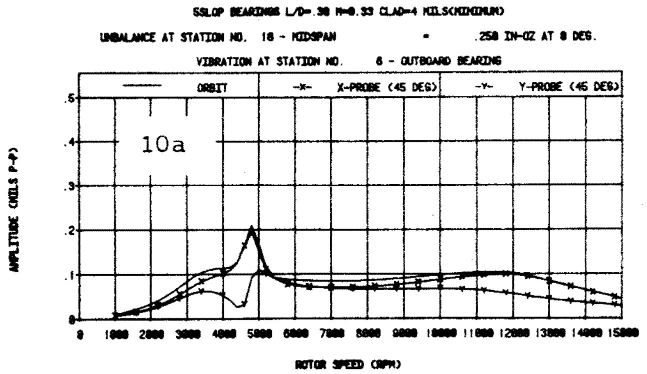
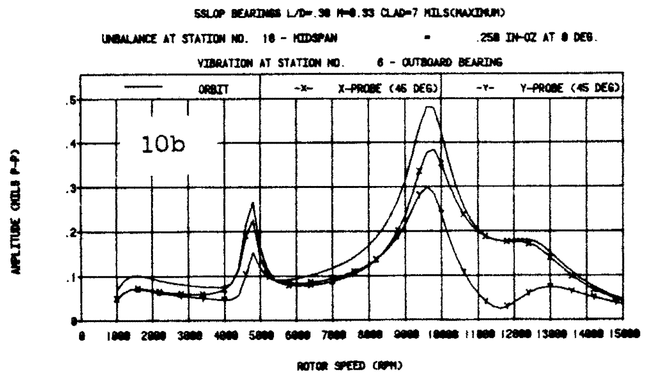

Figure 10 - Predicted Response at Outboard Bearing for Midspan Unbalance

Many people have the mistaken belief that the undamped horizontal critical speeds predicted by the intersection of the horizontal bearing curves with the natural frequency lines on a critical speed map will not occur in the unbalanced rotor response analysis or in the field. Note that in Figure 10a a critical speed response is predicted at 3300 rpm on the Y-45 degree probe. This frequency corresponds to the intersection of the horizontal bearing stiffness for the minimum clearance in Figure 1. The predicted undamped critical speed is 3100 rpm. Also in Figure 10b a response is predicted at 1500 rpm. This matches the intersection of the horizontal bearing stiffness for maximum clearance in Figure 1 (intersection at 1200 rpm). If the response calculations had been plotted for the horizontal and vertical directions rather than the 45-degree probes, these modes would be even better defined.

B. Sensitivity to Unbalance Location

A response analysis was performed on a turbine which had a speed range of approximately 2500-5000 rpm. Figure 11a shows the results of the response calculations for unbalance at the coupling. Two peak response critical speeds were excited at 3000 and 5000 rpm. For this unit, very little difference in predicted responses was noted as the bearing parameters were changed. However, when the unbalance was moved to the disc (Figure 11b), there were considerable differences in the predicted responses. Note that peak responses at 800, 2500, and 5000 rpm are predicted. The two low frequency critical speeds are considerably different from the 3000 rpm mode excited by coupling unbalance.

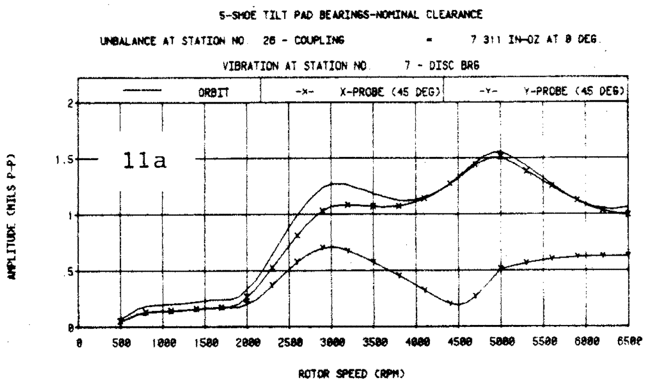
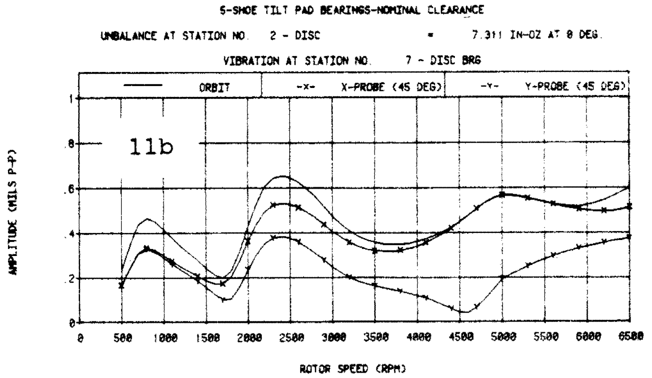

Figure 11 - Predicted Response at Disc-End Bearing for Nominal Clearance

C. Sensitivity to Bearing Damping

Rotor system damping is primarily the result of the damping contribution of the oil film. Since unbalance response levels at critical speeds are sensitive to the amount of damping in the system, a great deal of effort is spent in designing bearings which maximize damping. Rotor designers must depend on the accuracy of computed values of damping in the design stage in order to accurately predict peak response critical speeds.

As previously discussed, experimental and analytical investigations [4-8] indicate that damping values predicted using generally accepted design principles may be overly optimistic, that is, the calculated damping is higher than actually exists in the bearing. The reduced damping capacity of bearings may be caused by the pad or pivot distortion from loads or temperature, variations in oil film properties due to temperature distribution, and bearing support flexibility. Inclusion of all these effects in rotor analyses greatly increases the complexity of the computer codes required, the required computational time, and the expense of the analysis. Increasing the complexity of the analysis does not necessarily improve the accuracy, especially if there are uncertainties about the input assumptions.

Comparison of measured shaft vibration and predicted unbalance response often gives poor correlation with the rotor response calculations based upon the nominal bearing clearance and the API residual unbalance at the coupling or midspan. In trying to match the field measured data, one variable that is sometimes adjusted to obtain better correlation is the damping in the bearings. Good correlation is judged by the (1) agreement of the predicted peak response critical speed with the measured critical speed, (2) agreement of the amplification factor between the field and predicted, (3) agreement of the predicted and measured amplitudes.

An example of the effect of damping on the sensitivity of the first lateral critical speed of a six-stage centrifugal compressor is illustrated in Figure 12. The runout corrected field data is presented with predicted response curves for the outboard and inboard bearings for an unbalance of 1.05 oz-in at the midspan. Response calculations were made for 100%, 75%, 50% and 25% of computed bearing principal damping values. The bearing assembled clearances were measured in the field and used for the calculations of bearing properties. For this case the curve for 50% damping gives the best correlation with the measured data at frequencies near the first critical speed. In order to study the effect of other bearing parameters such as clearance and preload on this same data, other parametric variations were made.

Figure 13 illustrates the effect of bearing clearance on rotor unbalance response. The bearing preload was held constant at 0.3 and the machined clearances of the bearings were varied from 3 to 5 mils diametrical, which corresponds to the minimum and maximum clearances based on the tolerance ranges. As the clearances were increased, the effective bearing stiffness decreased, lowering the critical speed. The amplitudes of vibration also increased. The cases with variation of the clearances did not correlate as closely to the measured data on the inboard bearing as did the damping variation of Figure 12.

Figure 14 shows calculated unbalance response of the same rotor assuming a constant "ground-in clearance" (difference of radii of curvature of journal and pads) and a variation in preload. This change in preload results in a change in the actual assembled or installed

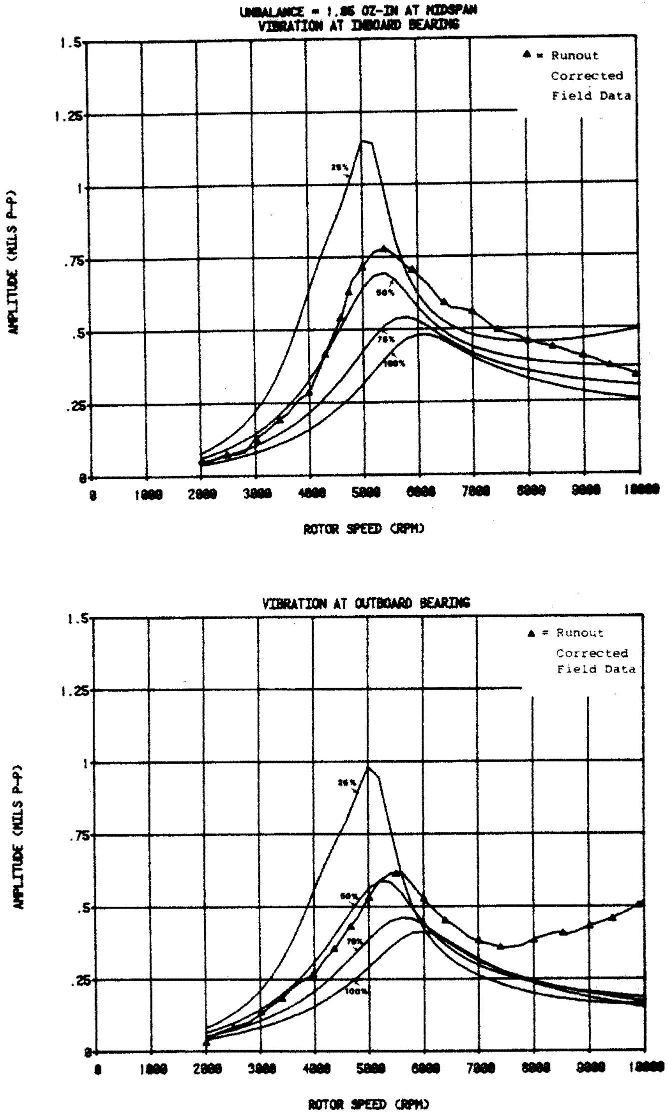

Figure 12 – Effect of Damping on Rotor Response

bearing clearance. As the preload (m) is increased, the assembled bearing clearance is reduced as described by

$$C_b = C_p(1-m) \qquad m = 1 - \frac{C_b}{C_p}$$

where C_b is the assembled diametrical clearance between the pad and the journal and C_p is the ground-in clearance. As illustrated in Figure 14, a preload change from 0.0 to 0.3 has little effect. The amplitudes are somewhat reduced at lower speeds indicating a stiffening effect. From 0.3 to 0.6 preload, a large change is seen. The correlation between these results and the measured data is still not as good as that obtained from the variation in damping since agreement was not as good on the inboard bearing for preloads of 0 to 0.3.

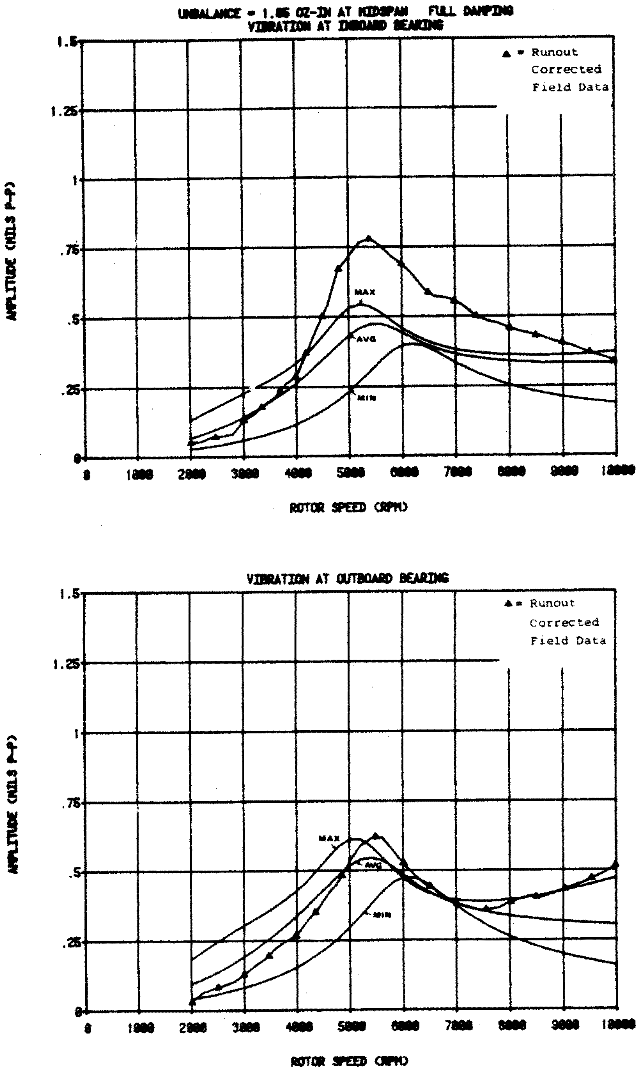

Figure 13 - Effect of Bearing Clearance on Rotor Response

Figure 15 illustrates the effect of holding the assembled clearance constant and varying the preload. This study was made to simulate the effects of temperature and load distortion of the pads that can cause an increase in the bearing preload without a significant change in the assembled clearance. It can be seen that the closest agreement was for the preload equal to 0.6. Again, the correlation is not as good as that obtained by simply reducing the damping.

The bearing pivot and support flexibility can affect the response analyses; however, for many light compressor rotors, the bearing stiffnesses are the controlling stiffness since they are less than one million lbs/in. Hertzian stress deflections on the pad pivots usually give stiffnesses on the order of 5 million lbs/in. Typical bearing housing stiffnesses are in the 1-5 million lbs/in range. Therefore these effects are small for many units.

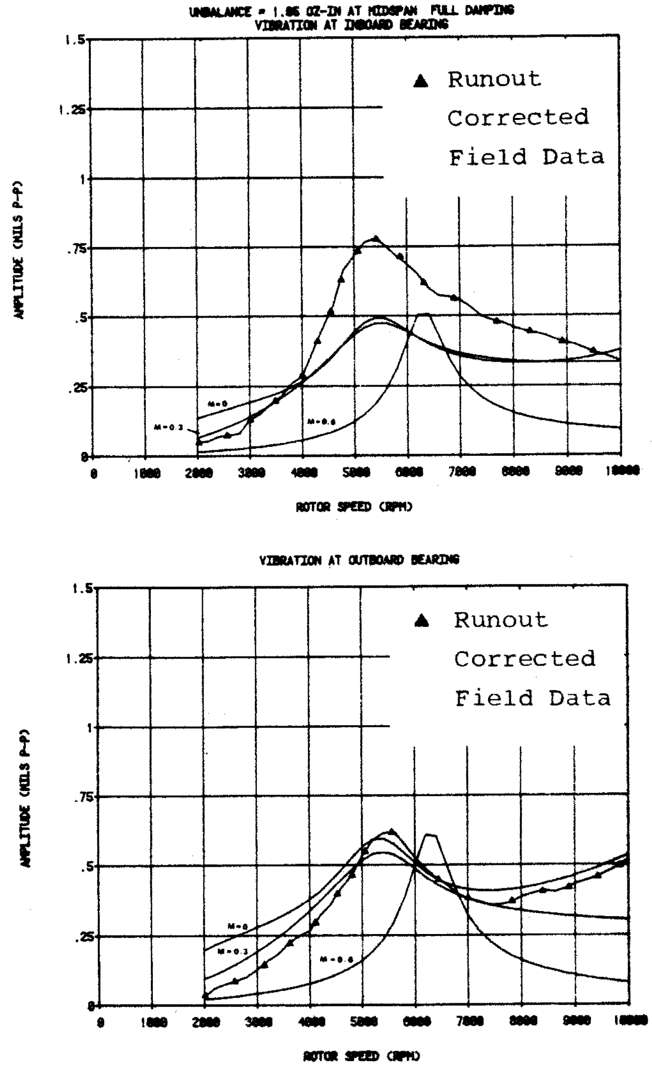

Figure 14 - Effect of Bearing Preload on Rotor Response - Constant Machined Clearance

D. Sensitivity to Unbalance Magnitude

One problem that repeatedly occurs is the sudden occurrence of a critical speed problem on a machine that has successfully run for several years. Many times the problem is caused by the fact that rotor response to unbalance is not linear with unbalance magnitude. In performing a computer analysis of unbalance response, the magnitude of the unbalance determines the response amplitude sensitivity of the rotor. The calculated response to the unbalance is linear; if the unbalance is doubled, the predicted vibrations will be doubled. However, in the analysis of field data, it has been found that the amplification factor which is a function of system damping (AF = $1/2\zeta$), sometimes depends upon the level of the unbalance. This nonlinear response can be illustrated by the data given in Figure 16 which gives the measured vibration on a rotor with the rotor balanced and with the API sensitivity unbalance weight added to the coupling. In addition to this field data, the predicted rotor response is plotted for 25%, 50%, 75% and 100% assumed bearing damping with unbalance at the coupling.

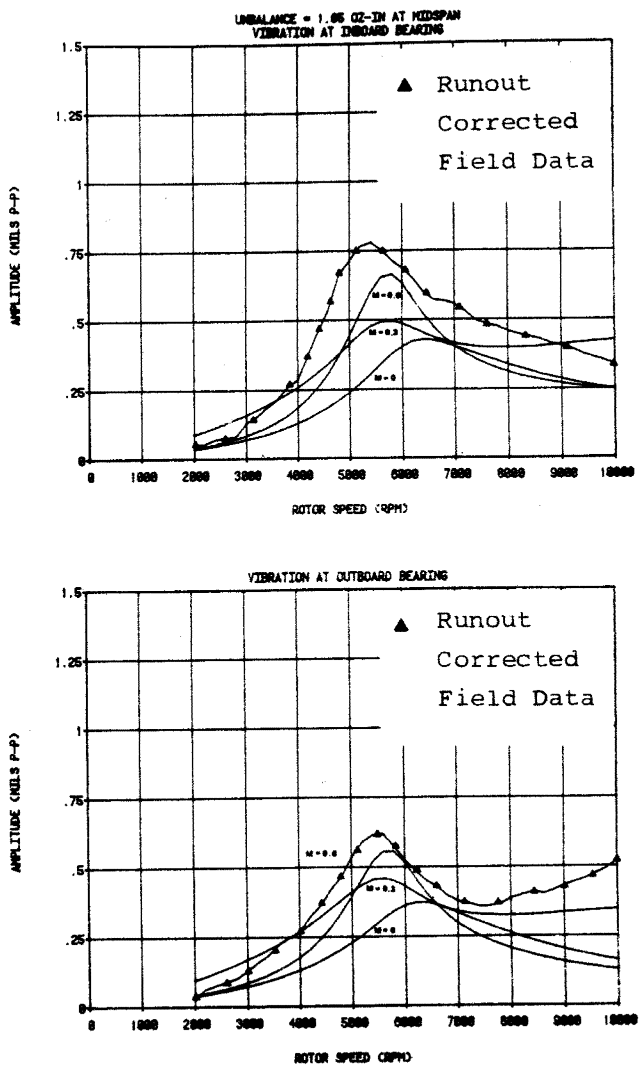

Figure 15 - Effect of Bearing Preload on Rotor Response - Constant Assembled Clearance

Comparison of the field and calculated data indicates that the balanced rotor field data has low amplification factors and in general correlates well with the 50%-75% damping cases with regard to overall shape and location of critical speeds. The API sensitivity test data shows much higher amplification factors and agrees more closely with the calculations using 25% damping. This response data illustrates the desirability of performing the API sensitivity tests whenever a critical speed is "calculated" or revealed during the shop tests.

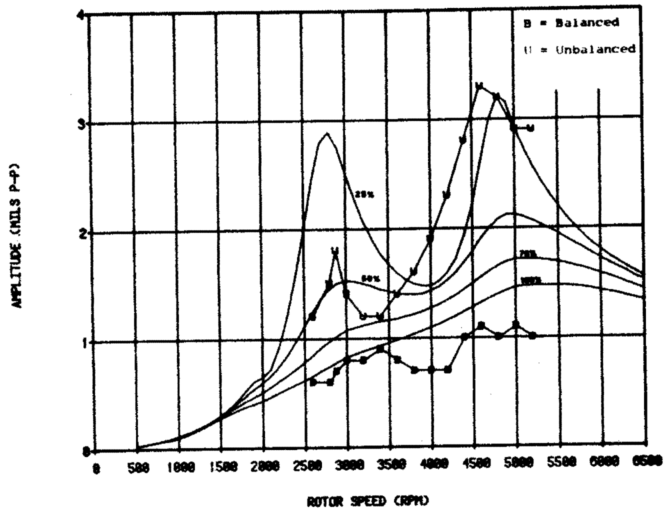

Figure 16 - Effect of Unbalance Magnitude on Rotor Response

IV. Conclusions

The evaluation of the acceptability of a compressor or turbine rotor with the applicable API Dynamics Specifications requires the calculation of the peak response critical speeds. The accuracy of the calculations and the agreement with shop or field data depends upon many factors that have been discussed in this paper.

1. The undamped critical speeds determined from the critical speed map and the bearing horizontal and vertical stiffnesses are useful in determining potential (peak response) critical speeds.

2. The prediction of peak response critical speeds should include the effects of the range of bearing clearance, preload, and oil temperature variation.

3. Midspan, coupling, and moment unbalances should be analyzed to determine the rotor peak response critical speeds.

4. Pad distortion due to load and temperature can change the effective bearing preload and adversely affect the bearing stiffness and damping properties and the rotor response characteristics.

5. Comparison of rotor response calculations to measured shaft vibration data indicates that the presently used procedures for calculating bearing damping is generally optimistic. Experimental comparisons of calculated versus measured damping coefficients by several investigators indicates that the damping may be only 20% of the calculated values for some tests.

6. The comparisons in this paper of measured rotor responses with reduced damping calculations

indicate that good correlation could be obtained, for the cases studied, by using response calculations with damping reduced to 25-50% of calculated values.

7. Vibration data measured on a balanced rotor and during its API sensitivity test showed that the increase in the unbalance caused a reduction in the system damping (increase in the amplification factor), thus indicating that the rotor response was nonlinear with the amplitude of unbalance.

8. Present computer codes assume a linear relationship between vibration amplitude and unbalance; therefore, the predicted responses can be lower than those actually measured when relatively high levels of unbalance are present in a rotor.

9. Rotor response calculations for peak response amplitudes using theoretical stiffness and damping values may agree with measured field results (correct critical speeds, correct amplification factors, correct amplitudes) for cases where the rotor balance is good and the overall amplitudes are low.

10. For those cases where the rotor balance is not as good and the vibrations become an appreciable percentage of the bearing clearance (greater than about 25 percent), the best correlation with measured field data may be obtained by derating the bearing damping. For many "trouble jobs", the bearing damping must be reduced to 25 to 50 percent of the calculated values to get good correlation as defined above.

11. The reduction in bearing damping to match field data should be done after the other effects such as pad distortion and support flexibility are considered.

12. Test stand data on a rotor may not reveal critical speeds that are calculated if the unbalance is low. However, the API specifications recognize this and call for a sensitivity test if a critical speed is calculated in the prohibited speed range.

13. Presently-used design procedures used by many manufacturers often result in machinery running on or near its second critical speed. The second critical speed is usually a conical whirl mode and for high bearing damping may not be excited; however, as illustrated in this paper, when the clearances are increased and the preloads decreased, thus causing the bearing damping to be significantly lowered, the second critical becomes excited. Thus the second critical speed becomes an "as calculated" peak response. When the second critical speed is predicted to occur in the disallowed speed range, the sensitivity test should be run to establish the rotor response to larger levels of unbalance.

14. Further research is needed to more accurately define the magnitude of bearing damping and its effect on rotor response.

15. The author's experience with instability analyses [12] indicates that stability calculations using the calculated bearing stiffness and damping values are in better agreement than stability calculations using derated damping.

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