A NEW VIBRATION CRITERIA FOR HIGH SPEED LARGE CAPACITY TURBOMACHINERY

by

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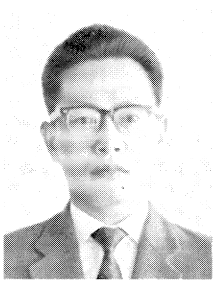

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Hiroshi Kanki is a research engineer of rotor dynamics in the Vibration Laboratory of the Takasago Technical Institute, Mitsubishi Heavy Industries, Ltd. He has been engaged in the analysis, design, and vibration diagnosis of large rotating machinery, such as steam turbines gas turbines, compressors, and pumps, and he developed the useful field balancing technique based on the polar diagram of unbalanced response. He is responsible for engineering on the optimum rotor design, balancing, and diagnostic system for rotating machinery.

Dr. Kanki received his M.S. degree in Mechanical Engineering from Kobe University in 1970, and received his Ph.D. degree from Osaka University in 1977.

ABSTRACT

This paper presents the development and analysis of new critical speed criteria. The critical speed has been defined as the resonant speed of the rotor bearing system at which large response amplitudes can occur. However, recent observations of actual rotor vibrations show that the sensitivity of the critical speed is not so high for many types of turbomachinery.

This new critical speed criteria, considering damping, Q-factor criteria, and modal mass consideration, is developed based on operating experience, analysis and physical considerations. The procedure that is followed in the analysis is shown and the approximate modal analysis used is introduced. Typical low sensitive rotors are used as examples to illustrate the application of the analysis.

INTRODUCTION

Recently there have been many rotor dynamics computer programs developed to be applied in the design of turbomachinery. Simultaneously, the vibration measuring techniques using electronic instruments have also progressed and much field data has been acquired. These capabilities are being improved to develop more cost effective, smooth running turbomachinery designs for all application areas.

In this paper, the critical speed, which is the fundamental parameter of rotor dynamics, is reviewed and a new criteria for its evaluation is described.

"Critical speed" has been defined for many years as "the resonant speed of the rotor bearing system at which large response amplitudes can occur."

However, recent observation of actual rotor vibration shows that the sensitivity of the critical speed is not so high for many types of turbomachinery.

The margin of separation between critical and operating speeds, which is specified in the API standard [1], is not always supported by operating experience.

The extensive rotor dynamics analysis and physical consideration of the vibration data for many actual machines show the following points of critical speed evaluation [2]:

1. In addition to the critical speed margin, the vibration sensitivity for the disturbance has a significant effect on the smoothness of operation.
2. The sensitivity is evaluated by direct response calculation or test, modal consideration (Q-factor and modal mass) for the mode.

The new critical speed criteria considering damping effects are developed based on operating experience, analysis, and physical considerations. In the recently proposed API standard [3] similar concepts seemed to be introduced.

ROTOR DYNAMICS ANALYSIS

To increase the reliability of rotating machinery, rotor dynamics is the most important item to consider. Extensive work in this area has been continued for more than ten years. The work is the result of the feed-back between analysis and experimentation for models and actual machines.

The vibration characteristics of large rotor systems are heavily dependent not only on the rotor itself, but also on the support conditions such as bearing oil film, pedestal and foundation. Efforts to increase the accuracy of rotor dynamics analysis are concentrated on this point.

Figure 1 outlines the specific areas of our rotor dynamics analysis studies.

1. The accuracy of rotor section data for numerical calculation is confirmed by "free-free exciting test" as shown in Figure 2. This method allows direct comparison of the shaft model actual rotor characteristics, without error induced by the rotor support conditions, and the test results as directly simulated by the calculation. Comparisons of test and calculated results for a large gas turbine rotor are shown in Figure 3.
2. The stiffness and damping coefficients of the oil film bearing have a very important affect on the rotor dynamic characteristics. The study on this item is done by model test and numerical calculation.

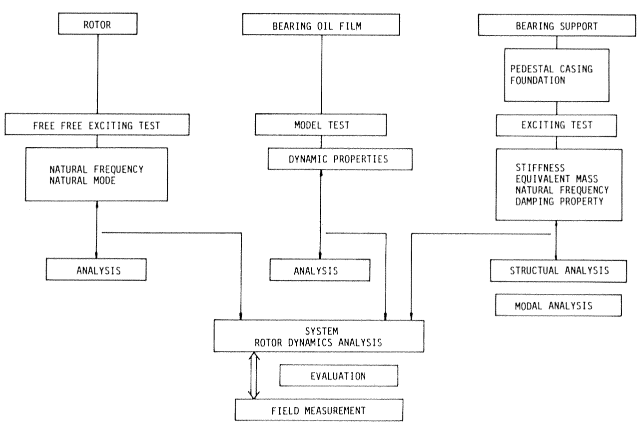

Figure 1. Points of Rotor Dynamics Study.

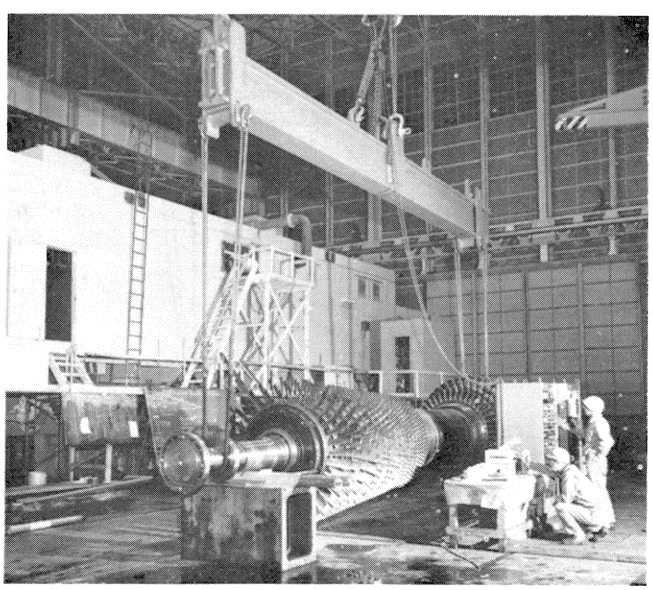

Figure 2. Free-Free Exciting Test of a Large Gas Turbine Rotor.

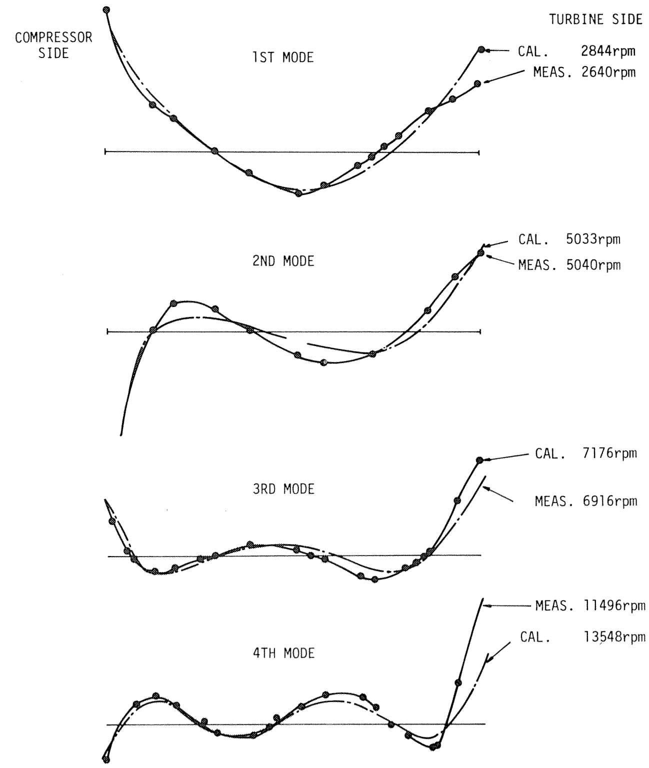

Figure 3. Comparison of Measured and Calculated Free-Free Modes for a Large Gas Turbine Rotor.

3. For large rotating machinery, the dynamic stiffness of the bearing support has a large effect on the operating characteristics. Because the actual bearing support is very complex, an excitation test is necessary to establish an accurate dynamic stiffness.

   Two methods are used for the excitation test:

   a. the impact method using the FFT system (Figure 4) for small machines.
   b. the hydraulic exciter (Figure 5) and single or multichannel mechanical impedance measuring (Figure 6) method for large machines.

   For the simple pedestal, the structural analysis method is also effective.

Figure 7 shows an example of dynamic pedestal stiffness measurements for a large turbine.

The measured data are used either as direct input data to the rotor dynamics analysis or in a simplified manner as shown in Table 1. Modal analysis techniques are applied to understand the complex data characteristics shown in Figure 7.

In parallel with this work, a rotor dynamics computer program system was developed and modified for effective application. The program system, shown in Figure 8, contains the basic calculation program and the detailed analysis program. The system is easy to use, cost effective, and provides an excellent plotter display.

With the accumulated test data and the computer analysis system, a reasonable degree of accuracy is obtained even for a very complex rotor system. For example, Figure 9 shows the results of the critical speed analysis and Figure 10 shows the response analysis for a large turboset.

In the utilization of the rotor dynamics program system, various machinery manufacturers use different techniques.

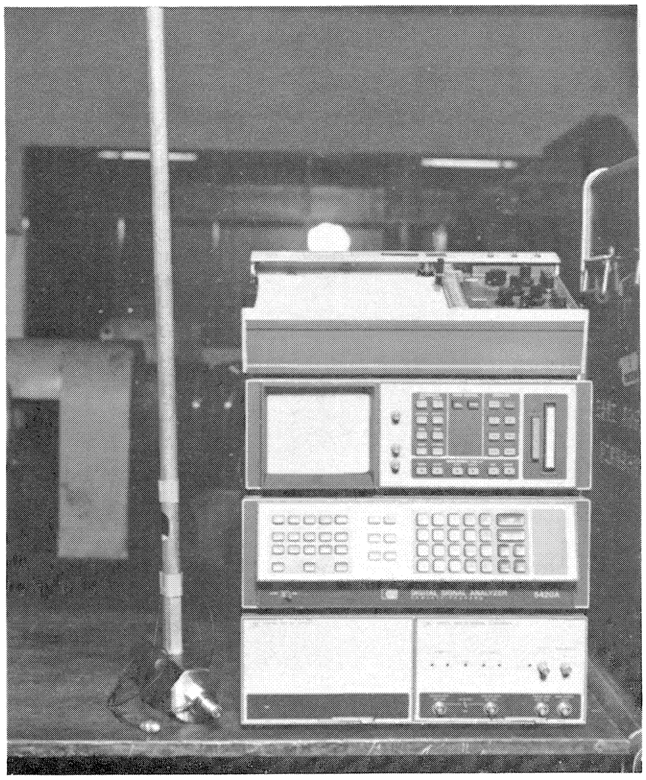

Figure 4. Impact Hammer and FFT System.

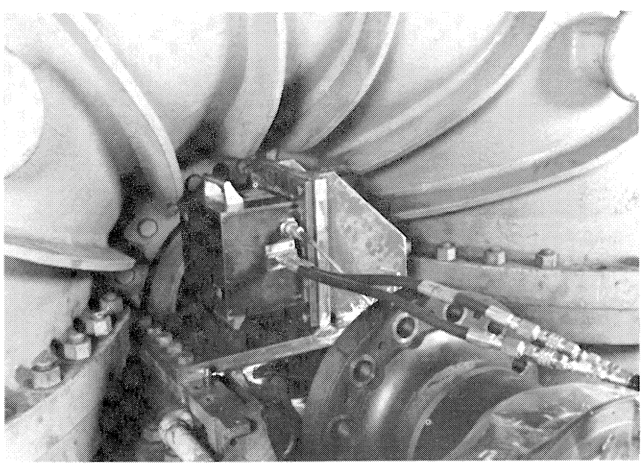

Figure 5. Hydraulic Exciter on the Pedestal.

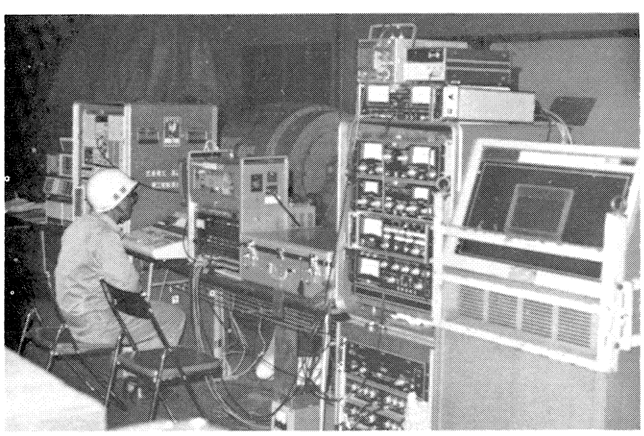

Figure 6. Mechanical Impedance Measuring System.

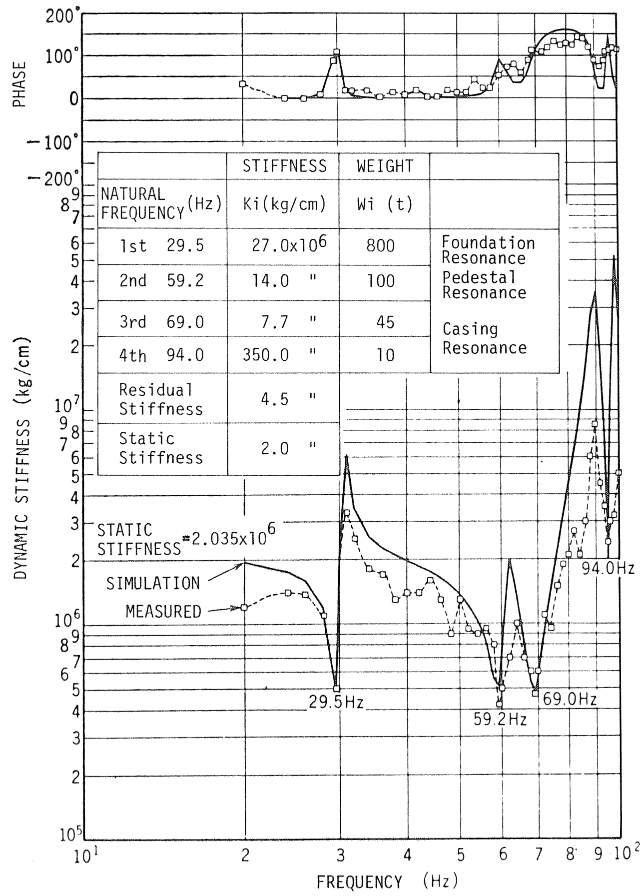

Figure 7. Measured Dynamic Stiffness Data for the Pedestal of a Large Turbine.

Regardless of the technique, it is most important that the analyzed results have reasonable reality and utility for judgement. Overly detailed analysis is not effective in producing good machines without the application of good engineering judgement.

Figure 11 is an example of a procedure used to analyze and optimize the rotor dynamic characteristics concerning lateral vibration. This procedure is composed of the case studies in the planning stage and of the final detailed analysis.

Modal Circle (Polar Plot) Balancing Method [4]

We developed the "Modal Circle Balancing Method" in 1972, and the method has been used successfully for many machines. This method is a practical application of modal balancing. A field balancing technique based on this method is very effective in reducing the number of trial weight runs and in diagnosing various rotor dynamics problems for larger machines.

The method is based on the simple points which are characterized by exciting point response and transfer point response. The balancing is the typical searching problem of unknown axial and circumferential unbalance distribution. The most important point of field balancing is the good estimation of the main unbalance in the early stage of balancing. The polar plot aids the physical understanding of the unbalance response characteristics and provides a rough estimation of the residual unbalance distribution. Typical examples of the polar plot are shown in Figures 12, 13 and 14 for a simplified rotor bearing system.

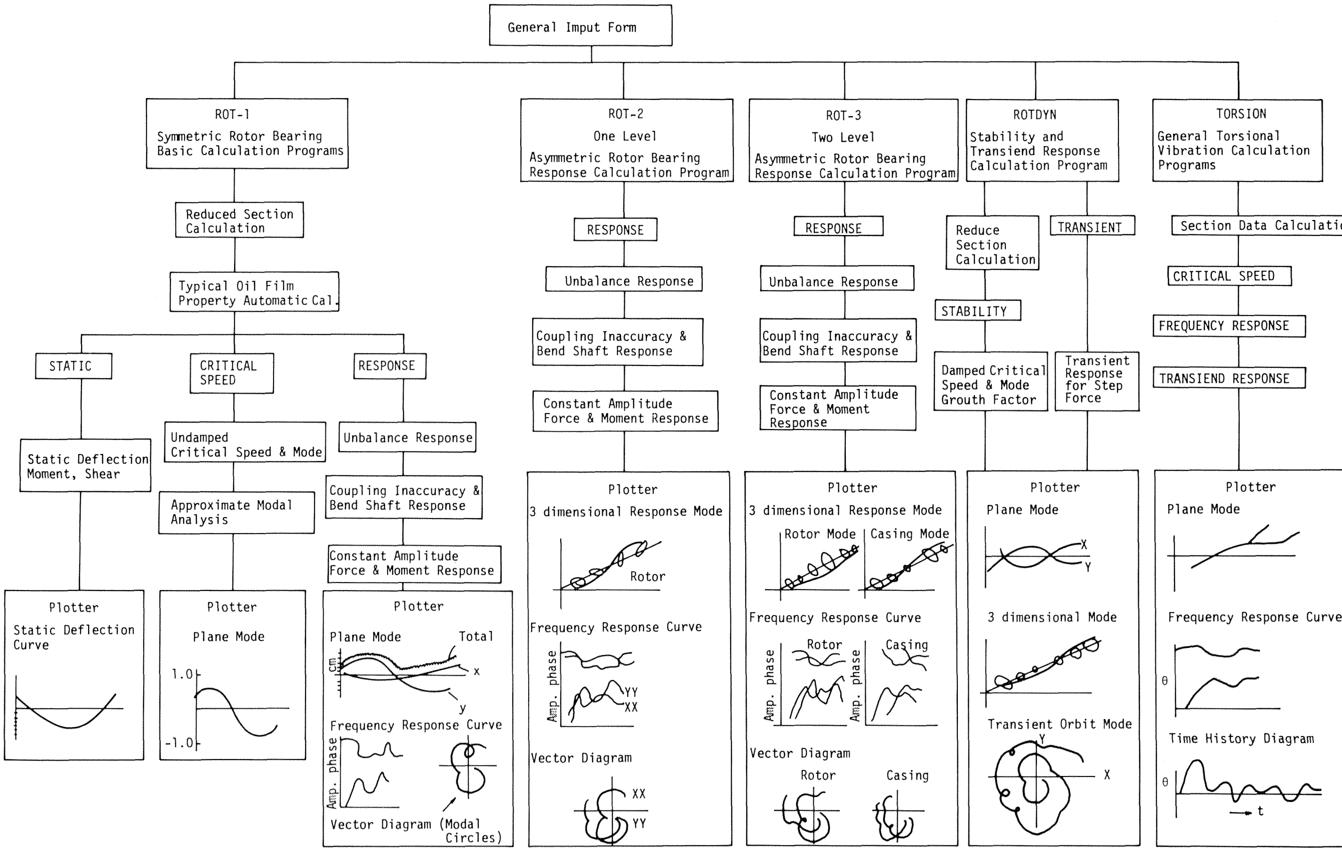

Figure 8. Rotor Dynamic Computer Programs System.

VERTICAL CRITICAL SPEED			
CAL/MEAS.	MEASURED	CALCULATED MODE SHAPE	
0.95	1100 rpm	1042 rpm	
0.99	1550	1532	
1.05	1750	1786	
1.00	1800	1806	
1.00	2040	2055	
0.97	2800	2721	
1.00	3100	3109	
1.00	3300	3316	

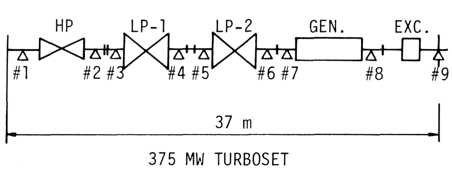

Figure 9. Example of Critical Speed Analysis.

TABLE 1. SIMPLIFIED MODELS OF THE BEARING SUPPORT.

MODEL			CONDITION	EXAMPLES
(A)	$K_2=\infty$ / K_2	Pin Joint or Spring	Hard Pedestal Light Pedestal Low Damping Pedestal	Spring Supported Pedestal Small Machine
(B)	K_2, $C_2\omega$	Spring Damper	Natural frequency is very high.	Hard Pedestal on Hard Foundation
(C)	Mp, K_2, $C_2\omega$	Mass Spring Damper	One natural frequency is low.	Usual Large Machine
(D)	Mp(ω), $K_2(\omega)$, $C_2\omega(\omega)$	Frequency Dependent Mass Spring Damper	Complex Support	Complex Support of Large Machine
(E)		Common Support Structure	Interaction between pedestals must be considered.	Gas Turbine, Heavy Rotor on Steel Bed

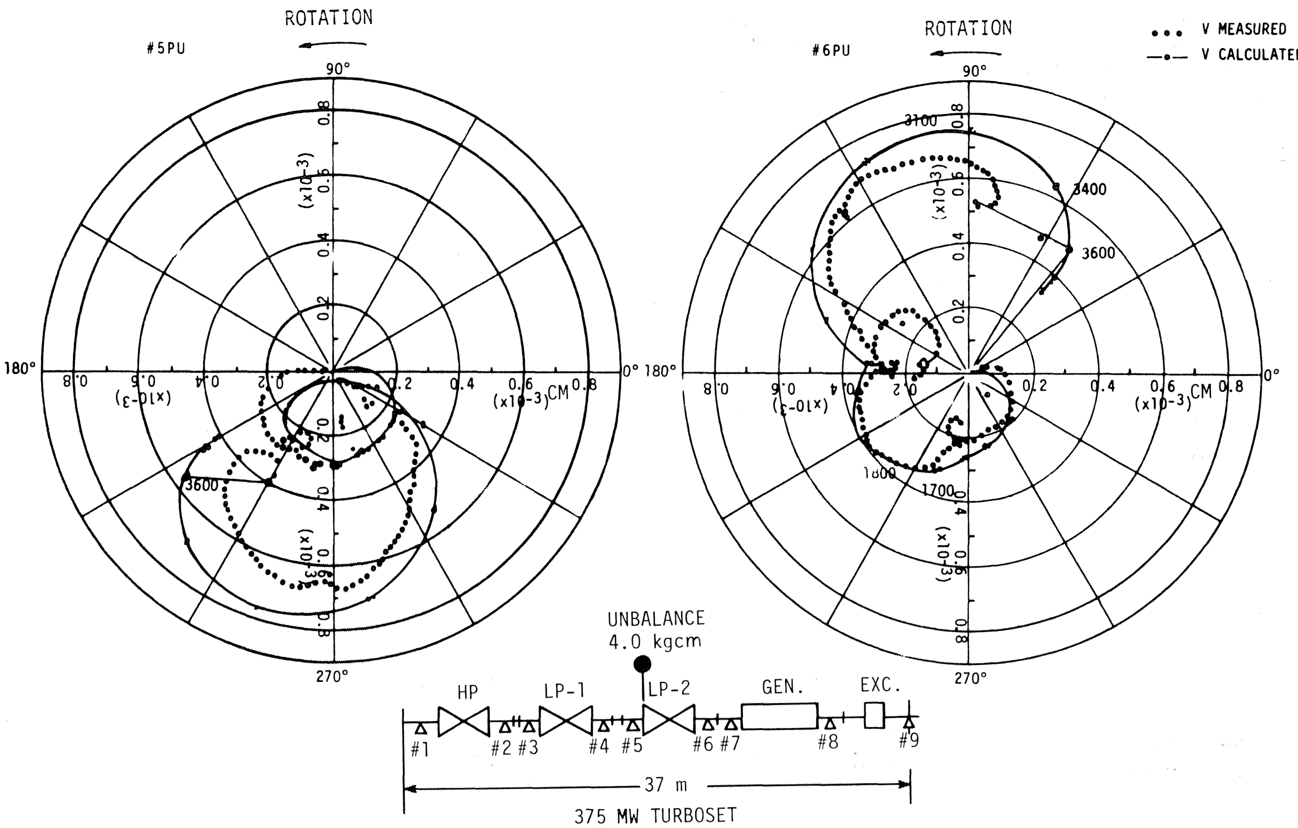

Figure 10. Example of Unbalance Response Analysis.

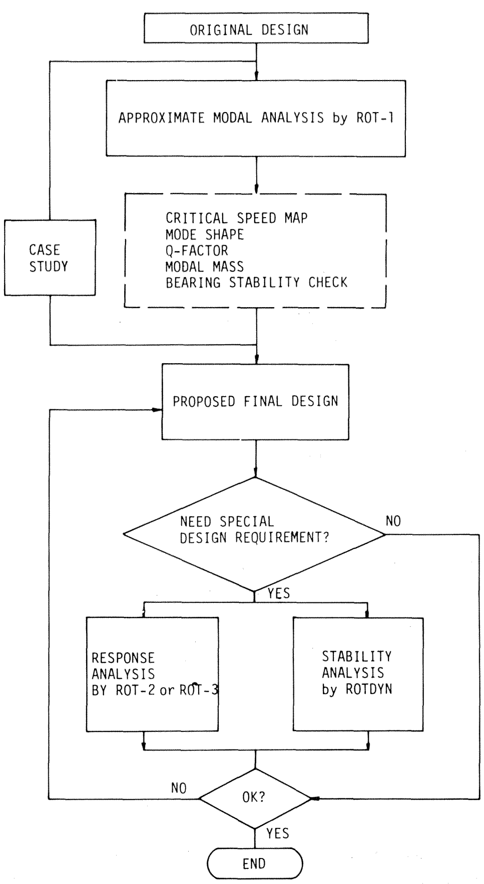

Figure 11. Critical Speed Analysis Flow Chart.

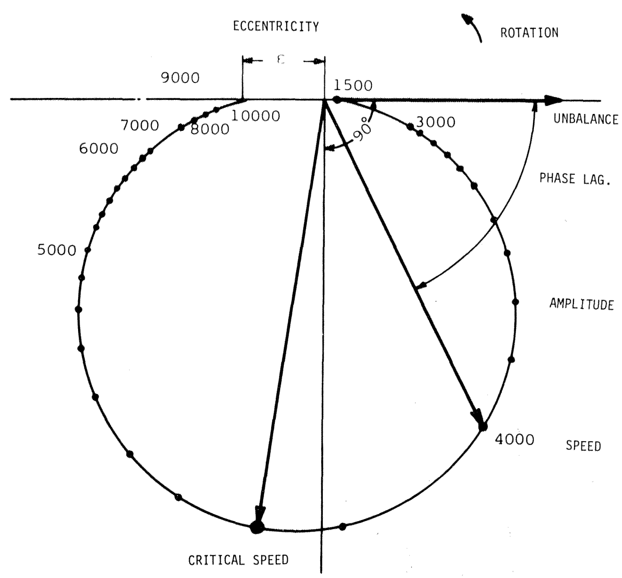

Figure 12. Polar Plot of Unbalance Response (Calculated).

Figures 13 and 14 show the symmetric rotor unbalance response characteristics of the #1 and #2 bearing journal points, respectively, for the unbalance on the side of the #1 bearing. The response of bearing #1 shows the typical exciting point response in which the first and second modes have the same circle orientation. Bearing #2 is the transfer point with the first and the second mode having opposite orientation. The circumferential direction of the unbalance is estimated by the directions of circles for both modes. Figure 15 shows the typical response polar plot of a single rotor with an overhung mass.

The detection of the unbalance location along the bearing span or in the overhung mass is the most important problem for effective balancing. The biggest difficulty will be in locating this point; however, the polar plot efficiently assists in assuming the unbalance location. The merits of this method, summarized in Figure 16, are as follows:

1. The number of trial runs will be minimal to obtain reasonable residual vibration because the trial weight will always be the balancing weight.
2. Accurate measurement of synchronous vibration by this method gives ample information which is useful for the diagnosis of the machine condition and for design feedback data in the form of:
   a. Critical speeds
   b. Q-factors of each critical speed
   c. Mode shapes.

NEW CRITICAL SPEED CRITERIA [2]

Extensive rotor dynamics analysis and vibration data acquired during start-up and field balancing (using the modal circle balancing method) of many machines have made it possible to establish the following points:

1. For the smooth operation of rotating machinery, the Q-factor ($Q \cong \frac{1}{2}\zeta$ (where ζ is the modal damping ratio for the mode) must be considered in addition to the proximity of the operating speed to the critical speed.

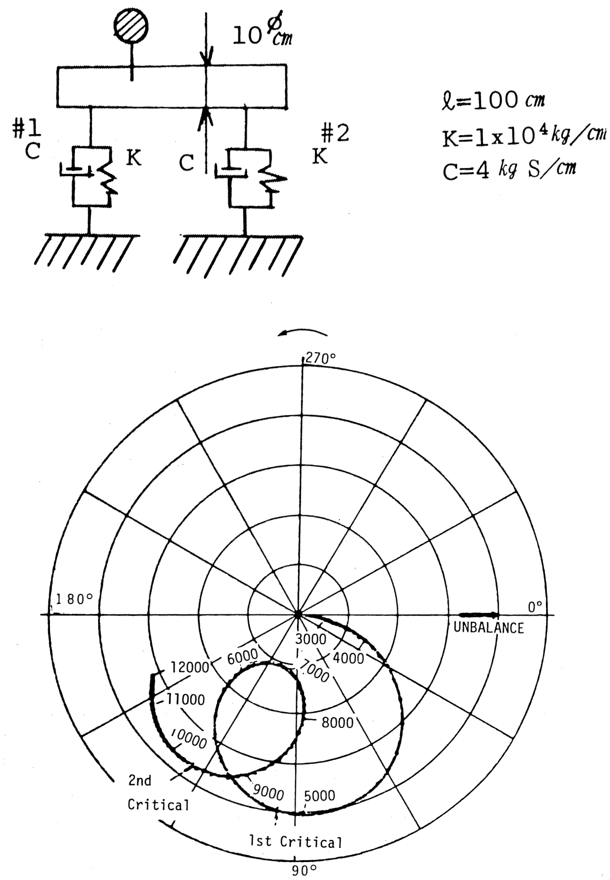

Figure 13. Response of Bearing #1 for a Typical Symmetric Rotor.

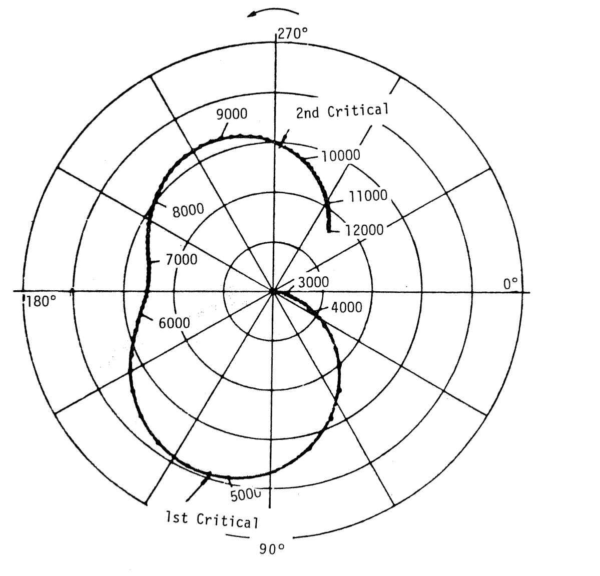

Figure 14. Response of Bearing #2 for a Typical Symmetric Rotor. (Same condition as in Figure 13).

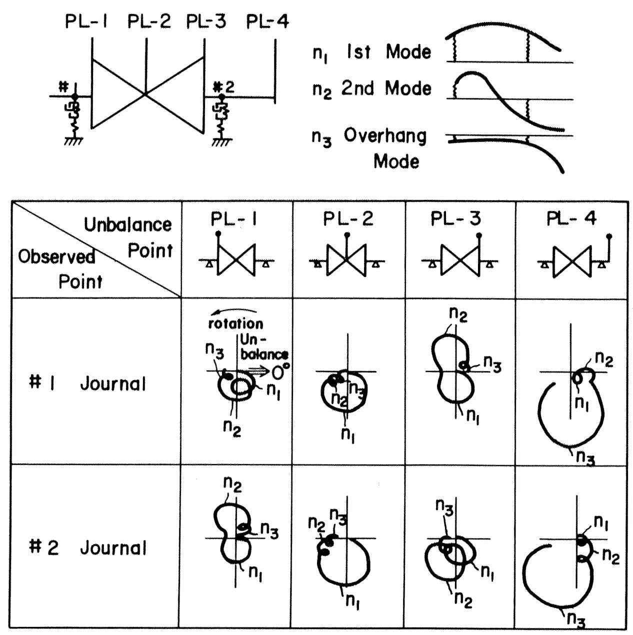

Figure 15. Modal Response Circles of Single Rotor with Overhang.

The operating experience and Q-factor are shown in Figure 17 for large turbomachinery.

2. The modal mass must be very carefully evaluated. Even a slight imbalance can cause large vibration amplitudes for a small modal mass (for example, the coupling in an overhung mode).

Basic Consideration

The vibration characteristics of the rotor bearing system for actual turbomachinery is very complex. However, the modal analysis technique enables the consideration that the actual system vibration is the sum of the individual mode components. With this assumption, it is then sufficient to consider

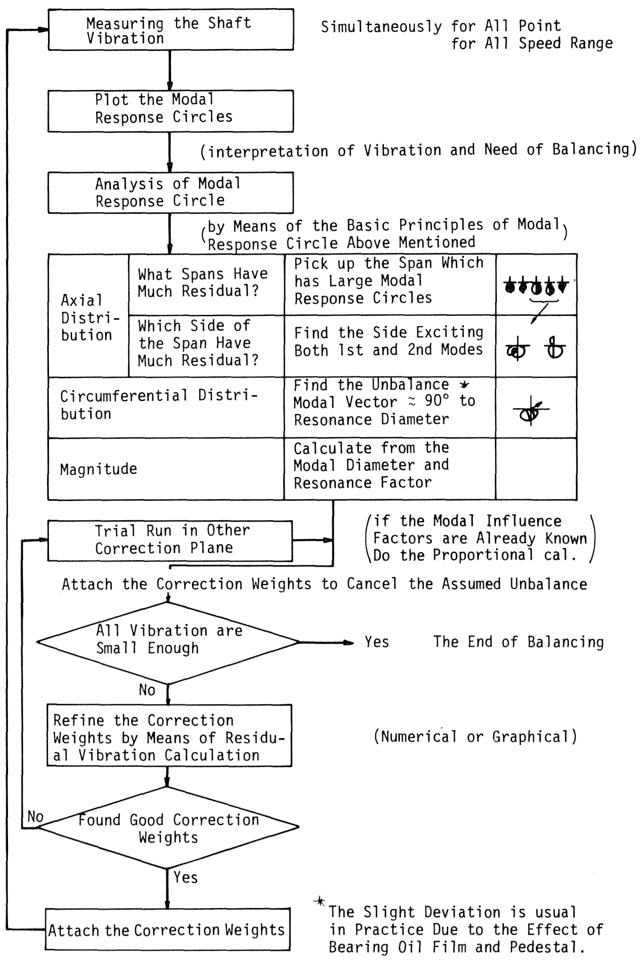

Figure 16. Method of Modal Response Circle Balancing.

the single degree systems independently. The response of rotor on point x is approximately shown in the following equation.

$$Z(X) = \sum_{r=1}^{\infty} \frac{\sum_{k=1}^{\eta} U_k \, \phi_{rk}^{\,2}}{M_r} \cdot \frac{\Omega^2 \phi_r(X)}{(\omega_r^{\,2} - \Omega^2) + 2i\,\zeta_r \omega_r \Omega} \tag{1}$$

U_k : unbalance at station k
ϕ_{rk} : normalized rth mode at station k
Ω : rotating speed
ω_r : rth critical speed
M_r : modal mass of rth mode
ζ_r : damping ratio of rth mode
η : speed/critical speed

The vibration magnification factor M (Ω) near the critical speed is represented by the following equation.

$$M(\Omega) = \frac{\left(\frac{\Omega}{\omega}\right)^2}{\left\{1 - \left(\frac{\Omega}{\omega}\right)^2\right\}^2 + 4\zeta^2\left(\frac{\Omega}{\omega}\right)^2} \quad \text{for unbalanced mass} \tag{2}$$

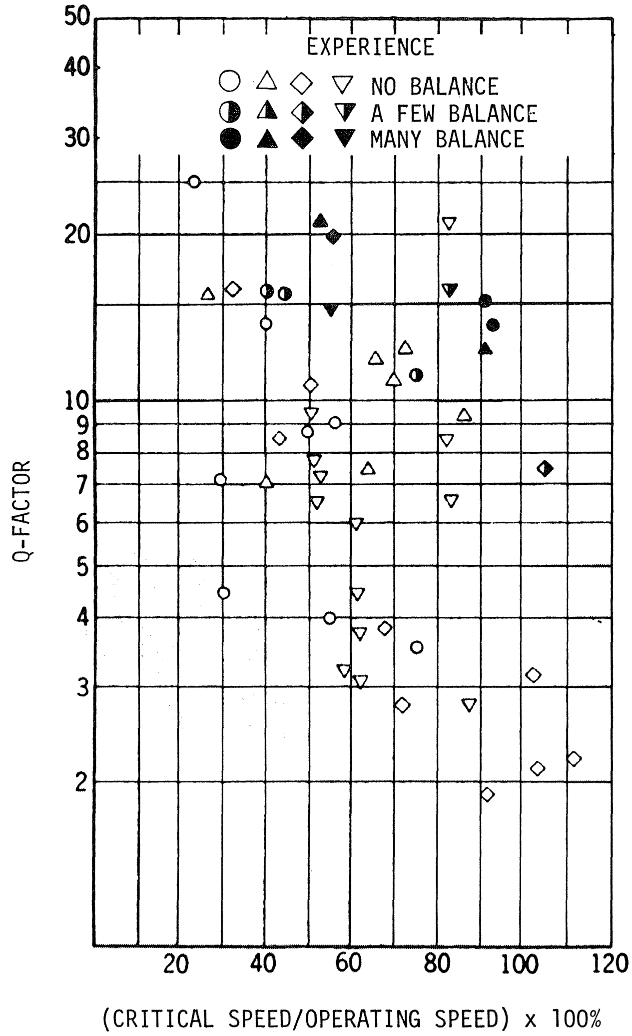

Figure 17. Q-Factor Experience for a Large Turbine.

$$M'(\Omega) = \frac{1}{\left\{1 - \left(\frac{\Omega}{\omega}\right)^2\right\}^2 + 4\zeta^2\left(\frac{\Omega}{\omega}\right)^2} \quad \text{for constant amplitude excitation} \tag{3}$$

Figure 18 shows values of magnification factors based on equation (2). The classical critical speed criteria [1] seems to be based on the undamped magnification factor ($\zeta=0$). However, actual turbomachinery has fairly large damping from the oil film bearings or other machine components such as seals. Therefore, the actual response characteristics do not always have sharp peaks. The response magnification factors are checked in the undamped condition as shown in Table 2. The magnification factor value M (Ω) on the critical speed boundary is 2.3 to 3.6, which will be interpreted as the response factor which allows continuous operation. This value may be valid for small or moderate size machines. However, it has been found that large machines on which field balancing is available have smaller unbalances so that larger response factors will be allowed. The above consideration is for the operating speed range. However, the response at the critical speed ($\eta=1$) is also important (particularly for large machines which are gradually accelerated or decelerated) to avoid large resonant response. Then the Q-factor must be limited within some allowable range.

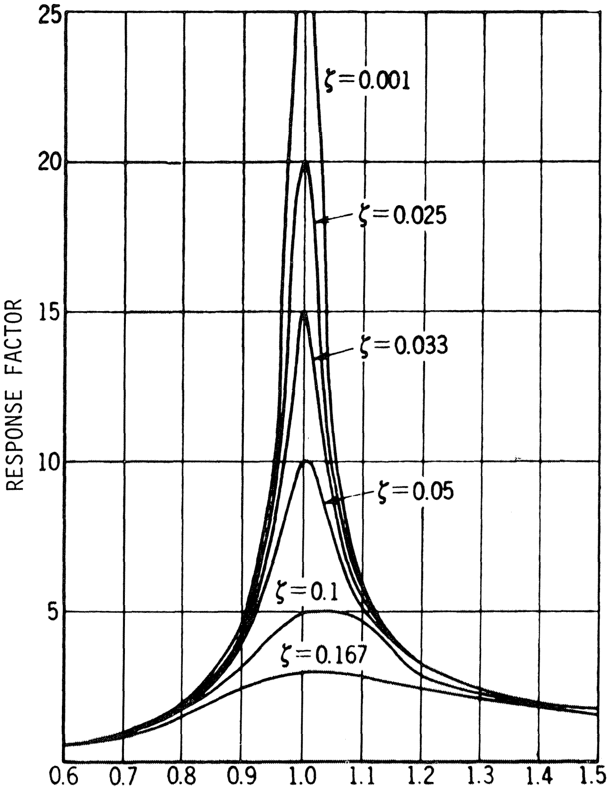

Figure 18. Response Factor in Critical Speed Region.

TABLE 2. RESPONSE FACTOR ON THE CRITICAL SPEED BOUNDARY.

	EXPRESSION WITHOUT DAMPING	-15%(η=1.176)	+20%(η=0.833)
UNBALANCE RESPONSE	$\eta^2/(1-\eta^2)$	3.6	2.3
CONST. FORCE RESPONSE	$1/(1-\eta^2)$	2.6	3.3

$$\eta = \frac{\text{SPEED}}{\text{CRITICAL SPEED}}$$

Based on the above two considerations, the Q-factor criteria shown in Figure 19 is proposed from operating experience. The typical characteristics of each region are summarized in Table 3. Recommended values are the Q=15 line for a large machine on which field balancing is available and the Q=10 line for the machine on which field balancing is not available.

The following comments will be necessary to use Figure 19.

1. The values on this chart will normally apply to large turbomachinery on which field balancing is available.
2. For the moderate size machine on which field balancing is not available, a value (one rank lower) should be used.
3. On the other hand, for the motor driven turbomachine, twice the value will be allowed except in the neighborhood of the operating speed.
4. For the first mode which has large modal mass, twice the value will be allowed.

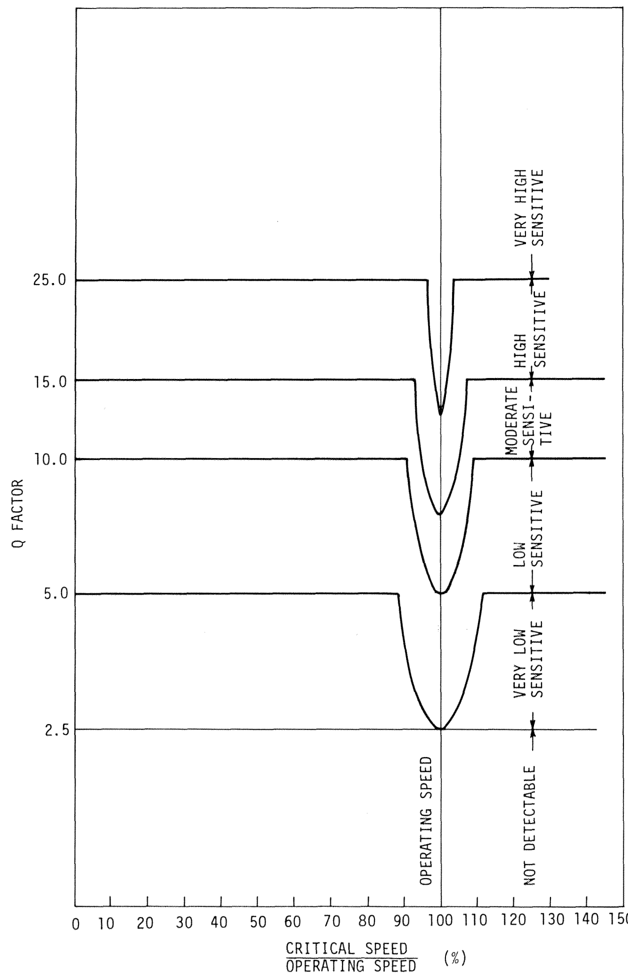

Figure 19. Proposed Q-Factor Criteria.

TABLE 3. TYPICAL CHARACTERISTICS OF EACH Q-FACTOR REGION.

Q-FACTOR	TYPICAL CHARACTERISTICS
$Q \leq 2.5$	Peak response of the critical speed is not detectable for normally balanced rotor and there is no problem in continuous operation.
$2.5 < Q \leq 5$	The critical speed is detectable, however the response is very low and continuous operation is possible for fairly balanced rotor.
$5 < Q \leq 10$	The critical speed is clearly detectable, the response is low, however continuous operation requires high speed balancing.
$10 < Q \leq 15$	The critical speed is moderately sensitive, the field balancing is required for some rotors.
$15 < Q \leq 25$	The critical speed is highly sensitive and the field balancing is required.
$Q > 25$	The critical speed is very much sensitive, so the special balancing technique will be required. If possible, it is better to try to reduce Q-factor.

The Q-factor which is used to evaluate this criteria is the actual value in principle. According to the increase of accuracy of rotor dynamics analysis, the calculated Q-factor will also be useful.

On the other hand, the simplified method which is based on the "Balda chart," Figure 20, is useful for simple rotors. From the critical speed ratio (ω/ω_{rigid}) and the bearing damping factor ($C\omega/K_B$), the Q-factor will be found. This chart considers the optimum stiffness and damping of the bearings. (The optimum line is shown by the solid line.)

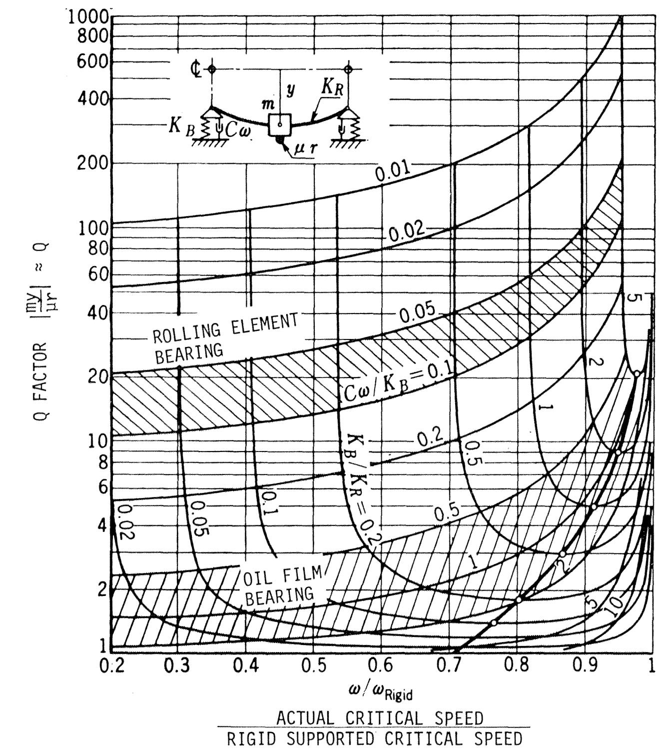

Figure 20. Q-Factor Calculation Chart [5].

The Evaluation of Modal Mass

Modal mass is defined as the equivalent mass for the mode in normalized mode. Equation (1) shows the response of the actual rotor which will be determined by the Q-factor and "modal mass." The response will be large for the mode which has a small modal mass for the same Q-factor. The modal mass is different according to the mass and stiffness distribution for each rotor. The modal mass for the typical rotor with both ends supported is 1/2 to 3/4 of the total mass for the first mode and 1/3 to 1/5 of the total mass or less for the overhung part mode. Then the modal mass for the overhung part, such as a gear or coupling, must be considered.

Critical Speed Criteria with Damping Effect

The classical critical speed separation margin criteria is based on the critical speed itself. However, the damping effect of the bearing system decreases the resonant effect significantly.

Then the new critical speed criteria are established by the following assumptions. Half of the Q-factor is defined to the equivalent response factor for the operating speed, based on the consideration of the operating experiences. The criteria is shown in Figure 21. In this chart the operating speed is assumed as fixed. For a variable speed machine, the chart must be enlarged in the 100% line.

The critical speed separation margin is easily evaluated by plotting the speed ratio and damping ratio (or Q-factor) on this chart. This chart is based on the unbalanced vibration. If there is another type excitation, for example, constant amplitude force, similar criteria will be available.

Examples

Two typical examples will be given.

1. High Speed Compressor Drive Turbine

Table 4 lists the results of an approximate modal analysis [6]. To check the actual response characteristics, the direct response calculations are done for two different unbalance distributions. In Figure 22, response plot (a) is based on the uniform eccentricity, and response plot (b) is based on the asymmetric distribution of eccentricity ($\epsilon = 1\mu\eta$). The journal vibration is 2 to 3 times higher than the eccentricity at the first and second critical speeds and 0.5 to 2 times higher at the operating speed (approximately 10000 rpm).

As shown by Table 4, the second critical speed is not sensitive, which indicates that any speed above the first critical speed is suited for continuous operation.

2. Large Axial Compressor

A second typical low sensitivity rotor system is the large axial compressor. The Q-factor diagram for this machine is shown in Figure 23. The measured vibration data are plotted in Figure 24. In this example, smooth operation is fully established in spite of critical speed existence in the operating speed (3600 rpm), because of low sensitivity.

CONCLUSION

The progress of the rotor dynamics analysis technique and accumulated field data allow the establishment of new critical speed criteria, considering the damping effect. The criteria has much merit for design flexibility and smooth operation of rotating machinery.

The procedure of analysis was shown, and the approximate modal analysis was introduced. Typical low sensitive rotors were shown as examples.

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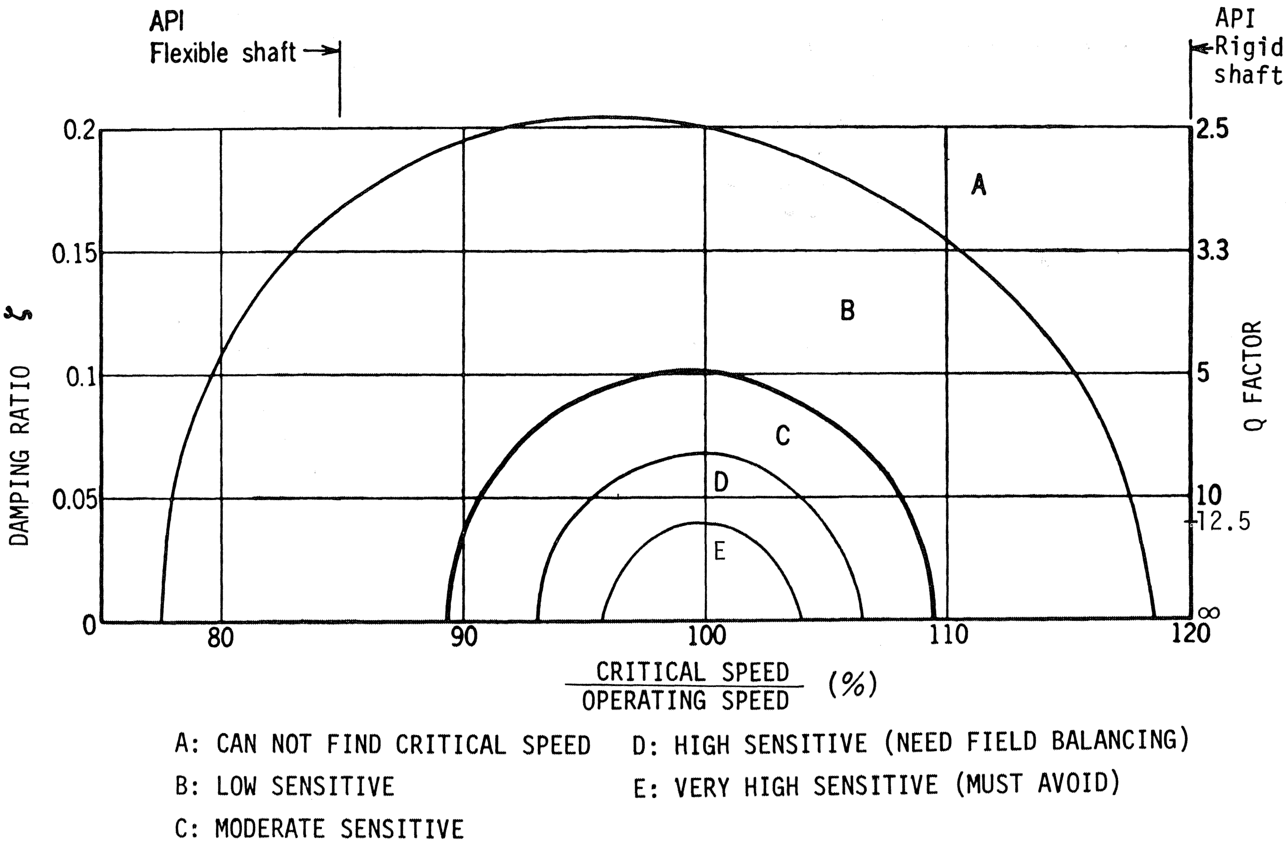

Figure 21. Critical Speed Criteria with Damping for Unbalance Response.

TABLE 4. RESULTS OF APPROXIMATE MODAL ANALYSIS FOR A SINGLE ROTOR.

	DIREC-TION	MODE NO.	UNDAMPED CRITICAL SPEED	REASONANCE SPEED	Q-FACTOR	MODAL MASS (MODAL WEIGHT)	RESIDUAL RESPONSE AT OPERATING SPEED	Q/M
COMPRESSOR DRIVE TURBINE WEIGHT = 1352kg	V	1	3501	3545	4.539	1.005 (984.9kg)	1.121	4.516
		2	7905	8695	1.698	0.318 (311.6kg)	1.613	5.340
		3	14573	15043	2.852	0.101 (99kg)	0.956	28.24
	H	1	2273	2350	2.786	1.185 (1161.3kg)	1.046	2.351
		2	4816	5089	2.186	0.314 (307.7kg)	1.224	6.962
		3	13298	13523	3.896	0.116 (113.7kg)	1.458	33.596

APPENDIX

APPROXIMATE MODAL ANALYSIS

Various methods are available to evaluate the sensitivity of a rotor bearing system. The following methods are usually used:

1. Damped natural frequency analysis.
2. Direct response analysis.
3. Approximate modal analysis.

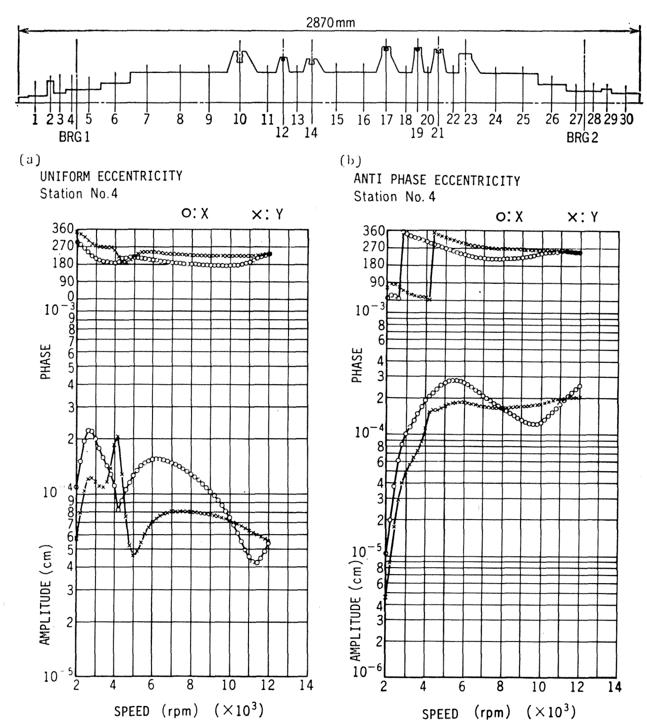

Figure 22. Unbalance Response for a High Speed Turbine.

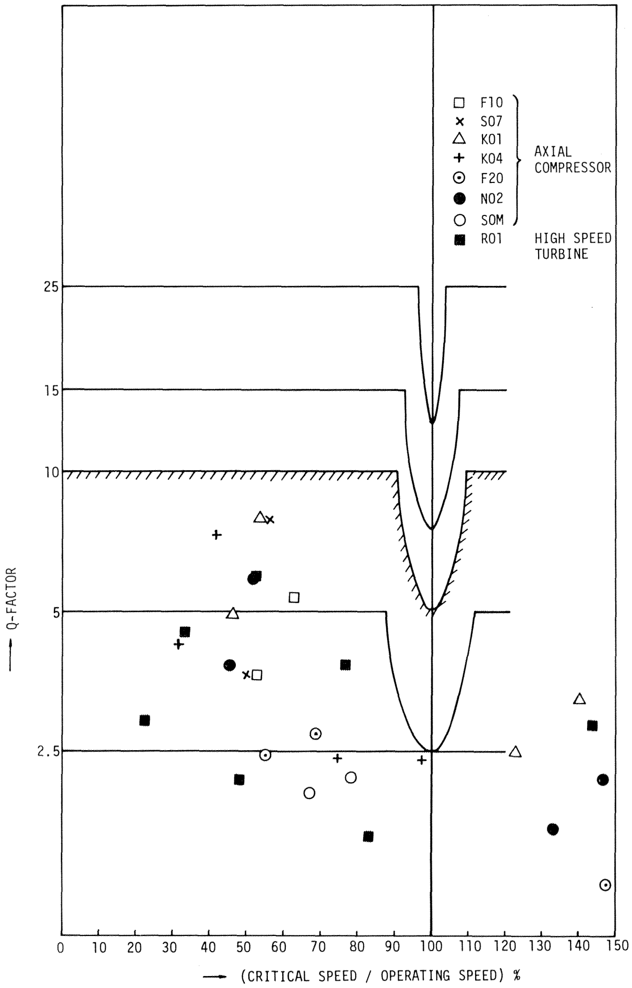

Figure 23. Q-Factor of Large Axial Compressor and High Speed Turbine.

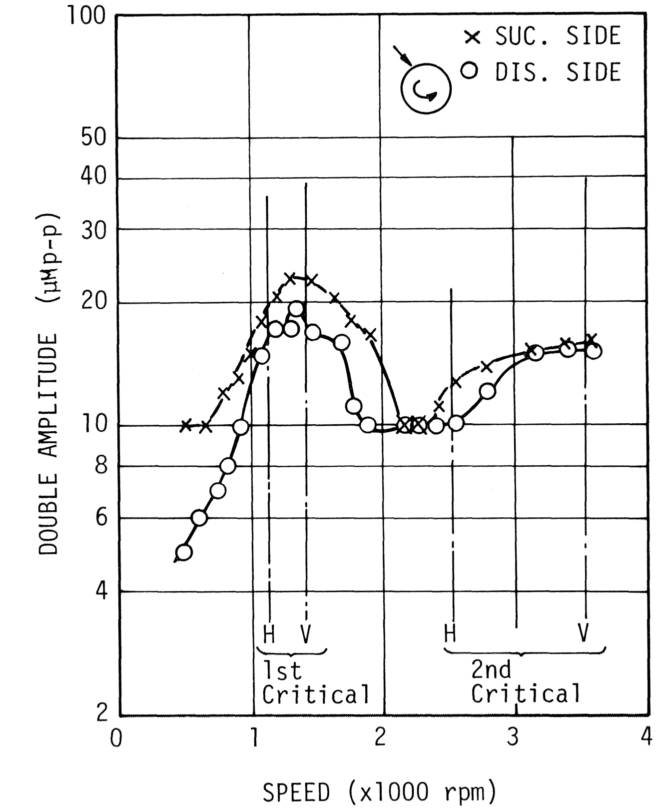

Figure 24. Measured Vibration of a Large Axial Compressor.

Damped natural frequency analysis is suitable for the simple rotor bearing system to evaluate sensitivity to synchronous excitation and stability characteristics. However, this method will be expensive for the large multibearing rotor system.

Direct response analysis is also an effective method to evaluate the sensitivity for external excitation such as unbalance. However, the assumption of disturbance is very troublesome for a complex rotor bearing system.

The above two methods have a high accuracy and will be suitable for a final check of the system.

Approximate modal analysis is the method to evaluate the modal characteristics by undamped calculation. The assumptions of this method are as follows:

1. The modal analysis is approximately used.
2. Damped modes are approximately equal to undamped modes.
3. Damping is concentrated at the bearings.

The calculations are done by the following steps:

STEP 1. Equivalent stiffness and damping calculation of the bearing part shown in Figure A-1 is given by

$$K_e = \frac{(K_1 + K_2')(K_1 \cdot K_2' - C_1\omega \cdot C_2\omega) + (K_1 \cdot C_2\omega + K_2' \cdot C_1\omega)(C_1\omega + C_2\omega)}{(K_1 + K_2')^2 + (C_1\omega + C_2\omega)^2} \quad \text{(A-1)}$$

$$C_e\omega = \frac{(K_1 + K_2')(K_1 \cdot C_2\omega + K_2' \cdot C_1\omega) - (C_1\omega + C_2\omega)(K_1 \cdot K_2' - C_1\omega \cdot C_2\omega)}{(K_1 + K_2')^2 + (C_1\omega + C_2\omega)^2} \quad \text{(A-2)}$$

$$K_2' = K_2 - M_p\omega^2 \quad \text{(A-3)}$$

STEP 2. Undamped critical speed and modes calculation

The undamped critical speed and modes are calculated by conventional undamped critical speed calculation.

ω_1, ω_2 . . . rth critical speed

ϕ_{11} . . . normalized natural modes

STEP 3. Calculation of modal parameters

1. Modal mass

$$M_r = \sum_{i=1}^{n} W_i \, \phi_{ri}^2/980 \quad \text{(A-4)}$$

where W_i : Weight of ith point

ϕ_{ri} : Normalized rth mode on ith point

2. Modal damping

$$\zeta_r = \sum_{j=1}^{m} C_e\omega\phi_{rj}^2/2M_r\omega_r^2 \quad \text{(A-5)}$$

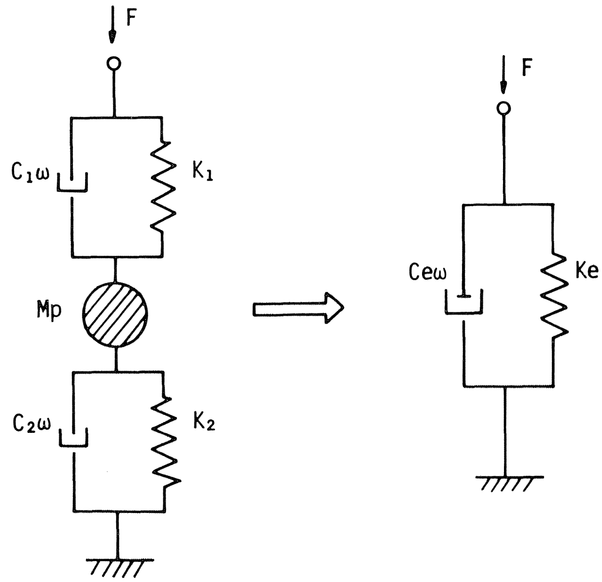

Figure A-1. Equivalent Stiffness and Damping of Bearing.

where $C_e\omega$: Equivalent damping of equation (A-2)

ϕ_{rj} : Normalized rth mode on jth point

m : Bearing number

3. Q-factor

$$Q_r = \frac{1}{2\zeta_r\sqrt{1-\zeta_r^2}} \cong \frac{1}{2\zeta_r} \qquad \text{(A-6)}$$

4. Resonant speed for unbalance

$$\Omega_r = \frac{\omega_r}{\sqrt{1-2\zeta_r^2}} \qquad \text{(A-7)}$$

5. Residual effect on the operating speed

$$\alpha_r = \frac{(\omega_o/\omega_r)^2}{\sqrt{\{1-(\omega_o/\omega_r)^2\}^2 + 4\zeta_r^2(\omega_o/\omega_r)^2}} \qquad \text{(A-8)}$$

ω_o : Operating speed