# ROTOR-BEARING SYSTEM DESIGN AUDIT

by

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## ABSTRACT

This paper concentrates on one aspect of the design audit of rotating machinery — review of the rotor-bearing system dynamics.

A systematic approach for performing this audit is presented. This approach is one which has proved effective in a broad range of process applications. Topics such as bearing capability, rotor lateral critical speeds and dynamic sensitivity, and overall rotor-bearing stability are covered.

An example problem and solution demonstrates the approach.

## INTRODUCTION

Recent attention by many users and manufacturers has centered on design audits to determine possible trouble spots prior to manufacture. In many ways the audit procedure must be aware of the design sequence and spot check to determine the adequacy of that procedure and the resulting design. This paper concentrates on one apsect of such an audit; that is, the review of the rotor-bearing system. For purposes of discussion, it will be assumed that the performance aspects (aerodynamics, motor-rotor electrical characteristics, etc.) are adequate. The relationship of the rotor-bearing system dynamics audit to the overall design procedure involving other engineering disciplines is illustrated in Fig. 1. The procedure addresses the following aspects:

1. The ability of the bearing support system to handle the loading with acceptable power losses;

2. Proximity to any lateral natural frequencies or critical speeds and anticipation of the system behavior in coming up to speed;

3. The dynamic sensitivity of the rotor in terms of its vibrational tendencies as a function of deterioration of balance;

4. The overall stability of the system to determine if the operating speed can be achieved without encountering subsynchronous vibration.

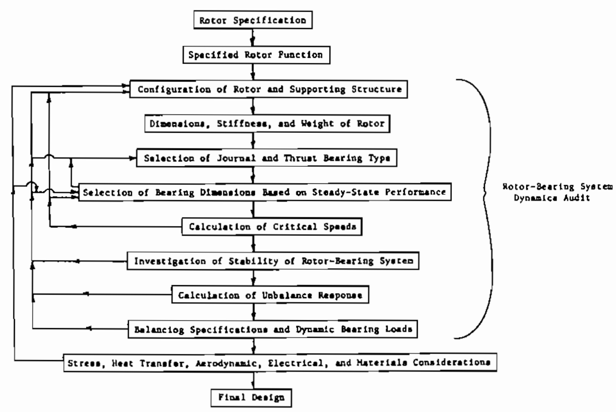


*Figure 1. Logic Diagram of Design Procedure* [1]

Unlike Refs. [1]* and [2] which are broader in scope in addressing turbomachinery design and failure prevention, it is the intent of this paper to present a systematic detailed approach towards accomplishing what we will call the rotor-bearing system dynamics audit. In addition, it will spell out certain pitfalls in terms of the adequacy of the tools used, and further, examples will be presented to demonstrate the viability of the techniques.

## DISCUSSION

Before proceeding to the presentation of a procedure, it is well to note several broad guidelines which have proved effective in performing the dynamic audit.

1. The data collection phase should be done in a consistent manner, preferably employing a checklist or a collection form. This not only assures completeness but it insures consistency.

2. Wherever possible, try to assure that the various analytical tools can accept the same rotor model to minimize the potential errors which can result when the effort is repeated.

3. The formulation of the model and the selection of stations must anticipate inputs required by the various

*Numbers refer to identically numbered References in the Reference Section.

analytical procedures. Very frequently, it will be necessary to include stations for inputs which will not necessarily be required by all the analytical tools.

4. Assure that the tools themselves are adequate and complete. Two very important requirements are that the dynamic tools consider gyroscopic effects and also include provision for an adequate number of station locations with provision to handle a multiplicity of bearings. This last requirement is extremely important in that many other system components including seals and aerodynamic elements can be represented by radial and cross-coupling stiffness and damping bearing coefficients.

The discussion which follows presents the specific procedure which will be in six parts. These are:

- Preparatory data collecting and calculation stages prior to the actual bearing and dynamic analyses.
- Lateral Support analyses.
- Critical speed determination and interpretation.
- Sensitivity or response of system to unbalance (synchronous response).
- System stability (subsynchronous response).
- System response to shock loading.

*Preparatory Data Collection and Calculations*

Generally speaking, the preliminary data collection must consider those items needed for accurately modeling the rotor and for defining the bearing characteristics. The rotor model must define the description of mass and stiffness along the shaft axis. Generally, the model is oriented to a particular computer treatment. However, several items should be common regardless of the specific calculation procedure. These are:

a. Locations such as bearings, seals, and concentrated mass locations should be identified in the model as discreet stations.

b. Station designation should be given to each change in cross-section of the rotor.

c. Transverse and polar mass moments of inertia should be identified for all major concentrated masses.

d. In addition to the above station locations, it is advisable to include stations for areas where aerodynamics excitations will be encountered. These include impellers, long labyrinth-type seals, and any long, close clearance locations. These additional stations will permit a common rotor model to be utilized throughout the dynamic analyses including the all-important evaluation of rotor system stability.

Table I presents a typical form used to gather the required information.

*Lateral Support Analysis*

As part of the preliminary calculations, the bearing steady state load reaction should be determined. These reactions include rotor weight, gear forces, hydraulic and aerodynamic forces, etc. Very frequently, the system may have more than two supports or because of the configuration, behave as though it gains support from other than the bearings. In this case, the rotor model previously described must be used to treat the shaft as an elastic beam and an

ROTOR OYNAMIC ANALYSIS – INFORMATION CHECKLIST

RDTDR DATA (Additional Sheet – )*

DRAWING ND.: ______________ DATE RECEIVED: ______________

| STA. NO. | W LBS. | $I_P$ LB.IN$^2$ | $I_T$ LB.IN$^2$ | L IN. | $D_S$ IN. | $D_W$ IN. | $D_I$ IN. | $\rho$ LB/IN.$^3$ | E PSI | G PSI |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | | |
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Balancing Specification oz. in. per plane (For Rotor Unbalance Response Cal.)

*Duplicate this sheet for more than ten stations.

W – Concentrated Weight
$I_P$ – Polar Moment of Inertia of Wt.
$I_T$ – Transverse Moment of Inertia of Wt.
L – Length between Sections
$D_S$ – Diameter of Sect. for Stiff. Cal.
$D_W$ – Diameter of Sect. for Weight Cal.
$D_I$ – Inner Diameter of Shaft
$\rho$, E, G, – Density, Young's Modulus, Shear Modulus

*Table No. I. Rotor Dynamic Analysis—Information Checklist*

analysis of the statically indeterminate system is required in order to determine bearing reactions.

The bearing characteristics can now be defined by knowing the load, operating speed and the lubricant properties. Once again, a variety of tools may exist to generate the required information. However, input information is now available to develop the following: (a) fluid film steady state characteristics, including flow, operating film thickness, attitude angle and power loss; (b) dynamic characteristics of stiffness and damping generally expressed in the form of eight coefficients.

Table II shows typical computer output sheets for the commonly used 5-pad, tilt-pad bearing. Data for a fixed geometry bearing would be similar except all eight coefficients would have nonzero values. This analysis summarizes the information noted above relative to the operating film thickness, flow, power loss, and temperature rise in the oil film. In addition, the dynamic coefficients which are utilized later in the analysis are included. These data should be reviewed carefully to determine:

1. The adequacy of the film thickness under the steady state operation loads;

2. The location of the attitude angle relative to grooves or feed ports (more applicable to fixed geometry bearings), and,

3. The temperature rise of the bearing based on calculated power loss and flow. A further check associated with this calculation should be the design adequacy of the lubricant supply system.

In addition, one should carefully review the thrust system to establish the adequacy of the selected thrust bearing type and size. Such a review would include establishing the direction and magnitude of the unbalanced aerodynamic thrust on the rotor and size the thrust bearing system accordingly. Although pressure (psi) loadings have

LOAD THROUGH PIVOTS    BEARING NO. 1    PRELOAD= -.0

BEARING LENGTH= 1.8900E+00    INLET TEMP(F) = 1.4000E+02    CL. RATIO C/R = 1.9600E-03
BEARING DIAM = 4.3300E+00    TEMP.RISE PRO.= 6.0000E-01

(a) STEADY STATE CHARACTERISYICS

| SPEED (RPM) | LOAD (LB) | TEMP RISE DEG.F | MEAN TEMP | VISCOS. MICRO-REYNS | 1/S | ATTITUDE ANGLE (DEG) | EFF. ECCEN RATIO | MIN FILM THICKNESS (MILS) | HP LOSS | INLET FLOW (GPM) |
|---|---|---|---|---|---|---|---|---|---|---|
| 2000 | 509 | 1.3 | 140.8 | 1.625 | 4.410 | .00 | .671 | 1.395 | .2976 | 2.9443 |
| 2500 | 509 | 1.5 | 141.0 | 1.619 | 3.542 | .00 | .627 | 1.584 | .4415 | 3.6818 |
| 3000 | 509 | 1.7 | 141.2 | 1.615 | 2.959 | .00 | .587 | 1.752 | .6115 | 4.4195 |
| 3500 | 509 | 2.0 | 141.3 | 1.611 | 2.542 | .00 | .552 | 1.900 | .8071 | 5.1573 |
| 4000 | 509 | 2.2 | 141.4 | 1.607 | 2.230 | .00 | .517 | 2.050 | 1.0282 | 5.8954 |
| 4500 | 509 | 2.4 | 141.6 | 1.604 | 1.986 | .00 | .485 | 2.186 | 1.2744 | 6.6336 |
| 5000 | 509 | 2.6 | 141.7 | 1.600 | 1.792 | .00 | .455 | 2.315 | 1.5456 | 7.3719 |
| 5500 | 509 | 2.8 | 141.8 | 1.596 | 1.633 | .00 | .424 | 2.443 | 1.8422 | 8.1103 |
| 6000 | 509 | 3.0 | 141.9 | 1.593 | 1.500 | .00 | .395 | 2.566 | 2.1638 | 8.8487 |
| 6500 | 509 | 3.3 | 142.1 | 1.589 | 1.388 | .00 | .368 | 2.682 | 2.5101 | 9.5872 |
| 7000 | 509 | 3.5 | 142.2 | 1.586 | 1.292 | .00 | .341 | 2.796 | 2.8818 | 10.3258 |
| 7500 | 509 | 3.7 | 142.3 | 1.582 | 1.208 | .00 | .314 | 2.909 | 3.2788 | 11.0644 |
| 8000 | 509 | 3.9 | 142.5 | 1.578 | 1.135 | .00 | .289 | 3.018 | 3.7008 | 11.8030 |
| 8500 | 509 | 4.1 | 142.6 | 1.575 | 1.071 | .00 | .264 | 3.123 | 4.1480 | 12.5417 |
| 9000 | 509 | 4.3 | 142.7 | 1.571 | 1.014 | .00 | .239 | 3.227 | 4.6207 | 13.2803 |
| 9500 | 509 | 4.5 | 142.8 | 1.568 | .962 | .00 | .215 | 3.331 | 5.1191 | 14.0190 |
| 10000 | 509 | 4.8 | 143.0 | 1.564 | .916 | .00 | .190 | 3.435 | 5.6433 | 14.7576 |

*Table No. IIa. Bearing Steady State Characteristics*

(b) DYNAMIC CHARACTERISTICS

| SPEED (RPM) | KXX (LB/IN) | KXY (LB/IN) | KYX (LB/IN) | KYY (LB/IN) | ωBXX (LB/IN) | ωBXY (LB/IN) | ωBYX (LB/IN) | ωBYY (LB/IN) |
|---|---|---|---|---|---|---|---|---|
| 2000 | 6.563E+05 | .0 | .0 | 9.071E+04 | 4.541E+05 | .0 | .0 | 8.683E+04 |
| 2500 | 5.821E+05 | .0 | .0 | 1.075E+05 | 4.239E+05 | .0 | .0 | 1.035E+05 |
| 3000 | 5.262E+05 | .0 | .0 | 1.230E+05 | 3.977E+05 | .0 | .0 | 1.186E+05 |
| 3500 | 4.820E+05 | .0 | .0 | 1.376E+05 | 3.755E+05 | .0 | .0 | 1.329E+05 |
| 4000 | 4.466E+05 | .0 | .0 | 1.511E+05 | 3.595E+05 | .0 | .0 | 1.470E+05 |
| 4500 | 4.160E+05 | .0 | .0 | 1.633E+05 | 3.465E+05 | .0 | .0 | 1.607E+05 |
| 5000 | 3.900E+05 | .0 | .0 | 1.749E+05 | 3.355E+05 | .0 | .0 | 1.742E+05 |
| 5500 | 3.700E+05 | .0 | .0 | 1.869E+05 | 3.271E+05 | .0 | .0 | 1.879E+05 |
| 6000 | 3.552E+05 | .0 | .0 | 1.996E+05 | 3.214E+05 | .0 | .0 | 2.018E+05 |
| 6500 | 3.453E+05 | .0 | .0 | 2.132E+05 | 3.188E+05 | .0 | .0 | 2.161E+05 |
| 7000 | 3.387E+05 | .0 | .0 | 2.269E+05 | 3.181E+05 | .0 | .0 | 2.304E+05 |
| 7500 | 3.347E+05 | .0 | .0 | 2.407E+05 | 3.190E+05 | .0 | .0 | 2.447E+05 |
| 8000 | 3.326E+05 | .0 | .0 | 2.543E+05 | 3.213E+05 | .0 | .0 | 2.589E+05 |
| 8500 | 3.323E+05 | .0 | .0 | 2.678E+05 | 3.247E+05 | .0 | .0 | 2.730E+05 |
| 9000 | 3.337E+05 | .0 | .0 | 2.813E+05 | 3.293E+05 | .0 | .0 | 2.871E+05 |
| 9500 | 3.367E+05 | .0 | .0 | 2.948E+05 | 3.351E+05 | .0 | .0 | 3.011E+05 |
| 10000 | 3.414E+05 | .0 | .0 | 3.083E+05 | 3.421E+05 | .0 | .0 | 3.152E+05 |

*Table No. IIb. Bearing Dynamic Characteristics*

been stated for various types of thrust bearings, high thrust loadings should be limited to thrust bearing systems which have equalizing ability for tilt-pad thrust bearings or gimbaled systems for the fixed geometries. In the absence of these features thrust loadings should be kept very conservative based upon detailed analysis of the thrust bearing.

It should be noted that in the case of the thrust bearing, a significant penalty is paid for including abnormally large sizes. This penalty results from the added weight and inertia which the rotor must carry as well as the fact that the power loss of the thrust bearing varies as the radius to the fourth power.

This is also a good time to review the mechanical configuration with an eye towards the fitting practice of the elements mounted to the shaft, the location of high temperature regions, etc.

### *Critical Speed Determination and Interpretation*

The calculation of rotor critical speeds is now quite common [3]. However, the techniques and the interpretations vary significantly. It is extremely important that the model be verified before going into extensive analysis. Thus, an effective critical speed calculation procedure will determine rotor weight and lengths which can be checked. An effective presentation technique utilized by the author's firm is the generation of a critical speed map. This presents on log-log graph paper the critical speed location as a function of support stiffness. This technique shows the rigid body or rigid shaft behavior as a straight line with a slope of 1/2 for the first two critical speeds. Thus, the rigid rotor and flexible rotor regions are readily apparent. Superposition of the bearing stiffness vs. speed characteristic on the map gives the location of the undamped rotor natural frequency. Fig. 2 presents a map of an actual rotor system and will be used to illustrate some of the observations which can be made by this technique.

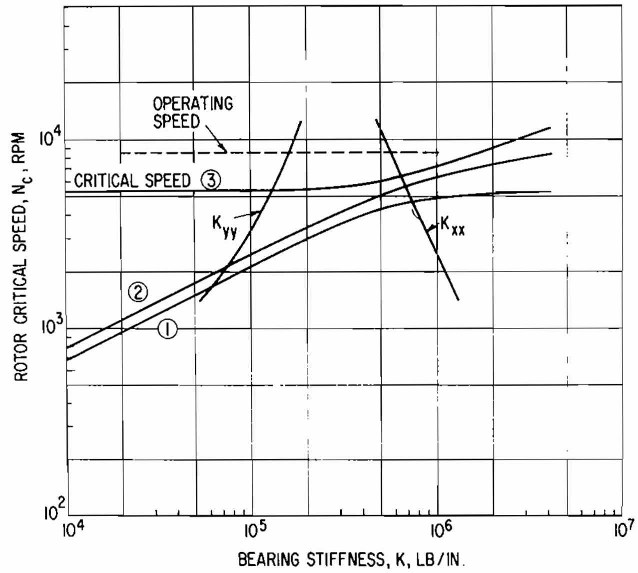


*Figure 2. Critical Speed Map*

Prior to discussing these, it should be noted that another output of this type of analysis is the mode shapes of the rotor at the critical speeds. This further indicates the degree of rotor bending and presents information relative to natural nodes of the rotor. This latter gives a qualitative indication about the effectiveness of the bearing damping. For example, if a bearing is located at a nodal location, very little bearing motion can result to utilize the inherent damping of the system. Where this is the case, amplitudes at the criticals will be unacceptably high. Fig. 3 shows typical mode shapes for the natural frequencies of such a

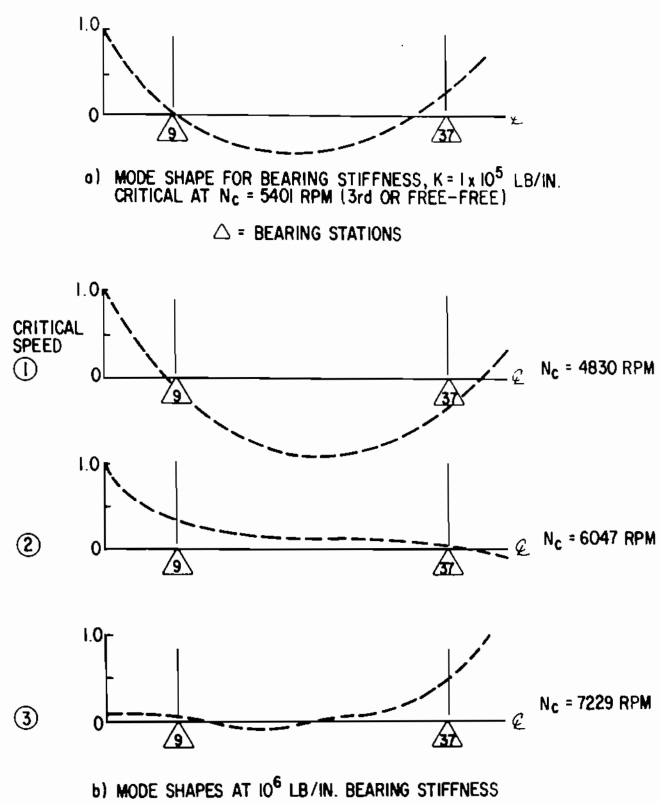


*Figure 3. Rotor Mode Shapes*

rotor. It will be noted that the mode shape for the first and third criticals suggests a bearing location very near a natural node. This rotor would, in fact, give unacceptably high vibration in passing through these criticals. A minor bearing location change resulted in a shift of the bearing relative to the subject node with resulting improvement in overall performance.

The above, although it can be useful as a guide, is rarely used exclusively, as will be discussed in the next section on "Rotor Response to Unbalance". Other observations may be similary drawn from the critical speed map. These generally are associated with rotor instability tendencies. These will be discussed further in the section on "Rotor Stability Analysis".

*Rotor Response to Unbalance*

This analysis [4] basically utilized the same rotor model described previously. For this analysis, however, the bearing spring and damping coefficients (all eight coefficients) will be utilized. Based upon the mode shapes previously described unbalance locations are selected to accentuate particular behavior. For example, to determine the sensitivity at the first critical, unbalance at midspan would be selected. Generally, unbalances at the shaft ends 180° out of phase with the midspan unbalance will further accentuate this critical and the third critical. The subject analysis determines the amplitude of response for each of the rotor locations as a function of rotor speed. Another result from this analysis is the bearing dynamic forces and system dissipated energy level.

In order to provide continuity of discussion, the unbalance response characteristics for the rotor discussed in the critical speed section are shown in Figs. 4 and 5. By switching the overhung thrust bearing with the journal bearing, which removed the natural rotor node from the journal bearing location, the rotor response was greatly reduced.

This type of analysis can further be used as a sensitivity analysis. In this case, a unit unbalance would be located sequentially at various rotor locations to determine which locations are particularly influential in generating high amplitudes of the rotor vibration at critical locations. Generally speaking, however, one would not arbitrarily introduce an unbalance at arbitrary distance increments. Rather, the locations would be selected to coincide with components added to the rotor, such as couplings, impellers, balance pistons and thrust runners.

Another value of the subject analysis is that it can establish the effect of small modifications in design such as component weight reduction, reduction of overhangs, and as has been previously discussed, bearing location or bearing span.

*System Stability Evaluation*

The evaluation of rotor bearing system stability utilizes the same mass elastic model as previously described and bearing spring and damping coefficients, as well as other system elements which can exhibit cross-coupling, or destabilizing behavior. The latter includes such elements as oil-buffered seals, aerodynamic excitation from impellers, and some influence from both a stiffening and a cross-coupling standpoint from long labyrinths. It should be noted that many of the mechanisms which exhibit cross-coupling are not completely defined. However, significant empirical data based upon field experience has proven that this approach can be completely viable.

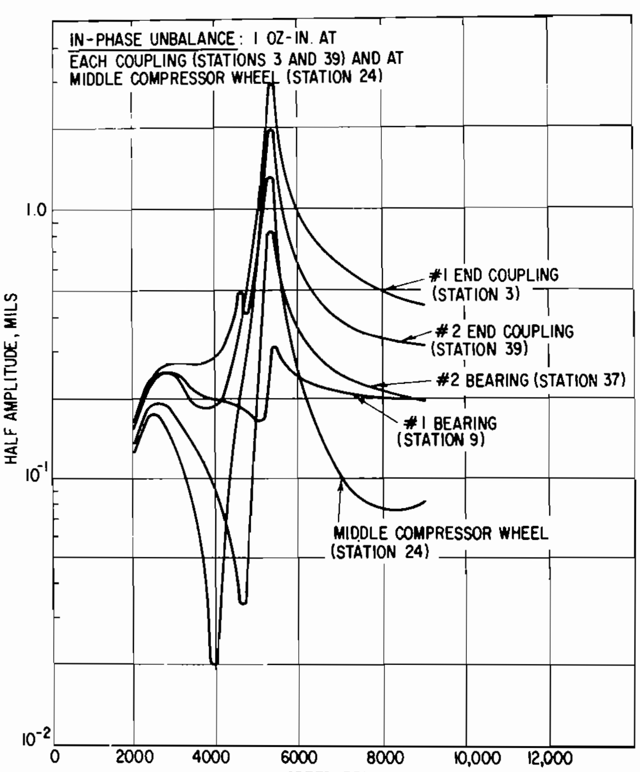


*Figure 4. Rotor Response of Unmodified Rotor Bearing System*

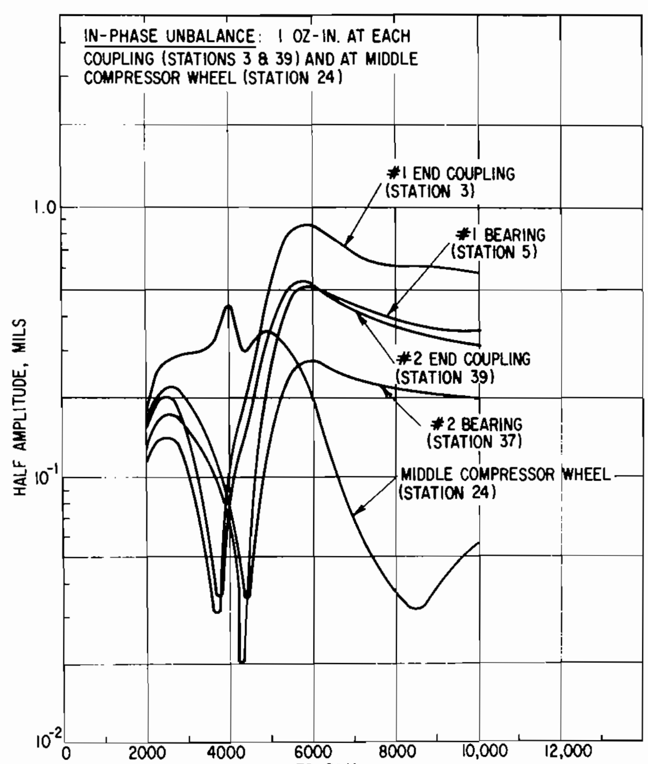


*Figure 5. Rotor Response After Switching Journal Bearing and Thrust Bearing Locations*

The procedure employed by the author's firm calculates the system damped natural frequencies and determines the output at these frequencies in the form of a logarithmic decrement. Reference 5 gives some of the theoretical background for the subject treatment. In summary, the results of this analysis are the system damped critical speeds and the sensitivity of the system towards supporting nonsynchronous vibrations. Generally, the first undamped critical speed (frequency) will be the primary one of interest in determining the stability margin of the system. However, each of the damped critical speeds, eigen values, will be accompanied by a system log decrement to guide this interpretation.

At any operating speed of a rotor-bearing system, there is a system natural frequency $\nu$(in general, nonsynchronous with the rotational speed) which can be excited and which has an amplitude decay exponent. Amplitude growth is described by the expression $e\lambda t$, where t is the time. In the commonly referred to logarithmic decrement, $\delta = -2\pi\lambda/\nu$, if $\delta$ is positive ($\lambda$ negative) the system is stable and adequately damped. In this case, any induced vibrations will decay. As $\delta$ becomes zero, and then negative, the rotor instability threshold speed becomes equal to the running speed. The rotor then becomes unstable and highly susceptible to any internal or external excitation forces.

Normally, the basic rotor model is first evaluated with the bearing system alone. This is valuable in that most available field data is more complete in the definition of rotor model and bearing system whereas the definition of the other elements in the system is often unknown or unattainable. Our experience has shown that log decrement values above 0.5 will generally portend a stable system. Negative values indicate definite instability while positive values below 0.25 indicate marginal behavior. Values between 0.25 and 0.5 represent a gray area which must be more thoroughly evaluated in terms of the other destabilizing forces in the system. Once having generally ascertained the sensitivity of the system, estimates are made for the effects of oil-buffered seals, aerodynamics, etc. In the case of the oil-buffered seal, two effects must be considered. First, there is a hydrostatic influence based upon the usually high pressure drop across the subject seals. Where the high friction exists, this provides a secondary support which makes the inherent bearing damping far less effective. In addition, the subject seals exhibit cross-coupling behavior which directly reduces the stability margin. The aerodynamic forces, although far less understood, have been represented as radial cross-coupling influences which are proportional to the torque level of the individual stages.

Figs. 6 and 7 are critical speed maps of two rotors on which extensive field experience has been gathered. It will be noted on the two figures that the critical speed map indicates additional guidelines as follows:

1. Operation is well above the first critical speed,
2. The rotor is operating in an extremely flexible regime.

For the two rotors, the log decrements at the first forward precessional mode were approximately 0 and 0.1, respectively. Further, initial oil seal and aerodynamic effects resulted in values less than zero. In point of fact, both rotors did exhibit unstable behavior, and corrective action was necessary to improve operating capability.

Based upon the large number of field observations, guidelines based upon the initial critical speed analysis have been formulated. These are listed below:

1. A rotor is generally more vulnerable to large subsynchronous response when the operating speed is more than a factor of 1.8 times the infinite support, ($10^7$ lb/in or greater) first critical speed.
2. Normally, the bearing stiffness range is from $10^5$ lb/in to $10^7$ lb/in. A rotor is more vulnerable to large subsynchronous response if it is so flexible that the first critical speed calculated with $10^7$ lb/in supports is not larger, by again, a factor of 1.8, compared with the first critical speed calculated with $10^5$ lb/in supports.

The remedies for such behavior can be multiple, depending upon the stage of development, whether the analysis is before the fact treatment or trouble-shooting. Some of the possible remedies in order of increasing effectiveness are as follows:

1. Modify the bearing characteristics to achieve more effective system damping. It will be noted here that a change as simple as a rotation of the bearing for between-pad operation from on-pad operation was sufficient in some borderline cases.
2. Modify rotor geometry in terms of bearing span, reduction of overhangs and in some instances, larger shaft sections.
3. Employ the use of an external damper in series with the bearing to stabilize the rotor system performance.

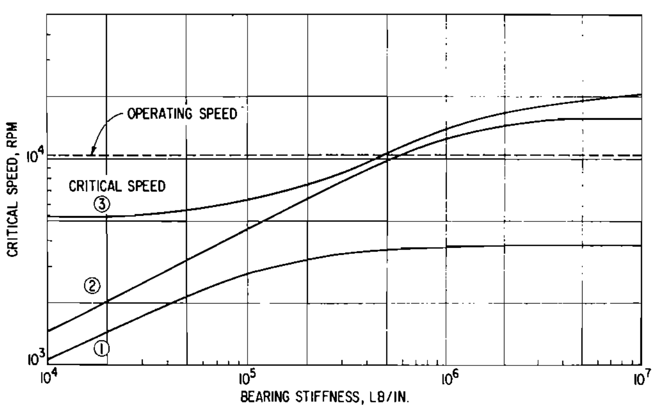


*Figure 6. Critical Speed Map of Very Flexible Rotor*

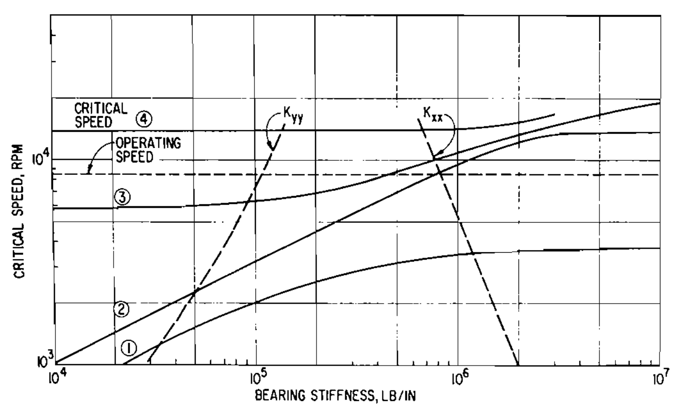


*Figure 7. Typical Critical Speed Map of Industrial Machine Rotor*

*System Response to Shock Load (Seismic Effects)*

This analysis is a rather new analysis developed to answer the question of whether certain machines can exist in earthquake prone environments. The same rotor-bearing model, as used previously, is analyzed to determine its sensitivity, or transient response, to a unit impulse, or shock pulse. This analysis [6] is an extension of the analysis used in evaluating system stability, and is based on the orthogonal complex modal functions associated with the eigen values of the system. The results of the analysis include the actual amplitude of the rotor at selected rotor stations as a function of time and input acceleration impulse magnitudes. The results of this analysis are thus used to determine a rotor's response to a given input, which unlike the unbalance response analysis, can have any frequency value.

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