

MEASUREMENT OF ROTOR SYSTEM DYNAMIC STIFFNESS BY PERTURBATION TESTING

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Specific aspects of the application of Modal Analysis to rotating machines are discussed in this paper. For lowest mode analysis, the circular-force perturbation testing gives the best results. Examples of application are presented.

1. MODAL ANALYSIS OF A ROTATING MACHINE

Experimental Modal Analysis or Modal Testing, as it is sometimes called, has become a popular method for studying practical vibration problems of mechanical structures. Application of Modal Testing for parameter identification and diagnostics of rotating machines, representing an important class of mechanical structures, has several specific aspects and requires a special approach. The results and predictions obtained by applying the classical "passive structure" Modal Testing to a rotating machine are usually incomplete and not sufficiently accurate for the most important modes, while providing information which is insignificant, for the rotating machine operating performance.

The specific aspects of rotating machines as subjects of Modal Analysis are discussed below.

1. All dynamic phenomena occurring during the performance of a rotating machine are closely related to the rotative motion of the rotor. The continuous supply of rotative energy makes the system "active." Numerous vibrational phenomena in rotating machines occur due to the transfer of energy from rotation (main performance) to vibration (undesirable side effects). Rotation of the shaft and all mechanical parts attached to it, as well as involvement in rotation of the working fluid (in fluid-flow machines), causes important modifications in modes and natural frequencies. In large turbomachines additional changes can be generated by thermal effects and foundation deformation. All these factors cause the results of Modal Testing of rotating machines "at rest" ("passive structure" approach) to differ significantly from the results of testing during machine operational conditions ("active structure" approach).

2. Rotors, which represent the main parts of rotating machines are equally constrained in two lateral directions; therefore, they perform vibrational motion which always has two inseparable lateral components (conventionally called "vertical" and "horizontal"). The result forms two-dimensional precessional motion of the rotor. Unidirectional impulse testing widely used in Modal Analysis, when applied to a rotating shaft, will undoubtedly result in a response containing both vertical and horizontal components.

3. In practical performances of rotors the precessional motion can contain multi-frequency components, each of them having a definite relation to the direction of rotation. In a most general case, each individual component can be either forward (direction of precession the same as direction of rotation) or backward (direction of precession opposite to rotation). Direction of precessional motion is vital to the rotor integrity. The net deformation frequency of the shaft is equal to the difference between rotative and precessional frequencies, taking into account their signs. During backward precession the shaft is therefore a subject of high frequency deformation (sum of both frequencies). This significantly increases a fatigue hazard. When measuring rotating machine vibrations it is very important to identify each vibrational frequency component whether it is forward or backward. Narrow band filtering and time base orbit analysis is extremely helpful for this purpose. In classical Modal Testing, "negative" frequencies have no meaning. Applied to rotating machines, "negative" frequency has a direct and very significant physical interpretation related to backward precession. Classical unidirectional impulse testing of a rotating shaft will result in a response containing elements of forward and backward precession.

4. Most important vibrational phenomena of rotating machines are related to the rotor lateral vibrations (sometimes coupled lateral/torsional/longitudinal vibrations). Each mode of rotor lateral vibration contains two components (vertical and horizontal), the characteristics of which are usually slightly different, as a result of elastic/mass nonsymmetry of the rotor and supporting structure in two lateral directions. Modal Testing of structures with closely spaced modes presents numerous difficulties. Rotating machines belong to this category. An alleviation of this problem consists, however, in close-to-symmetry modes; it is therefore reasonable to talk about "pair modes" in rotating machines (e.g., "first mode vertical" and "first mode horizontal").

5. Classical Modal Testing deals with a high number of modes of a structure in a wide spectrum of frequencies. In the performance of rotating machines, the most important are the lowest modes and low frequency precessional phenomena. This fact is related to the rigidity of the rotor system and to the relationship between the actual rotative speed and rotor precessional dynamic phenomena. Firstly, rigidity/mass characteristics of a rotor are always placed in a lower range of frequencies than those of the supporting structure. The lowest modes of the rotating machine correspond therefore to the modes of the rotor itself. Secondly, the rotating machine has its own continuously active forcing function, the unbalance, an inseparable feature of the rotating system. The frequency of this forcing is equal to the rotor actual rotative speed. The resulting motion is referred to as synchronous precession. A rotating machine operational speed, even if it represents dozens of thousands of rpm, seldom exceeds third balance resonance frequency; therefore, main interest is concentrated on investigating the rotor first two or three modes, as the rotating machine has to survive resonances of the lowest modes during each start-up and shutdown. The amplitudes of rotor deformation at low modes are the highest; therefore, they are of the greatest concern.

6. Another aspect of importance focused on rotor lowest modes is the fact that all self-excited vibrational-precessional phenomena occurring during a performance of a rotating machine are characterized by low frequencies, always located in the sub-synchronous region (frequencies lower than synchronous frequency). The self-excited vibrations occur when rotative speed is sufficiently high, and they are often referred to as rotor instabilities, significantly affecting the machine operation. The frequency of self-excited vibrations is either equal to a fraction of the actual rotative speed, and the same ratio to rotative speed is maintained if the rotative

speed varies (oil whirl, partial rub) or it is equal to the rotor first bending mode natural frequency (oil whip, full annular rub). Due to the specific role of internal friction, subsynchronous vibrations of rotating machines are always characterized by much higher amplitudes than supersynchronous vibrations [1].

7. During classical Modal Testing when dealing with high number of modes, the accuracy of the phase angle readings is usually low. In rotating machines the phase angle represents an extremely important parameter. It not only gives the information on the force/response relationship, but also can be related to the shaft rotative motion. It also yields significant information for the identification of modal parameters. Limiting Modal Analysis to the lowest modes permits one to increase accuracy in the phase angle readings.

8. Finally the most important aspect: the results of Modal Testing of rotors during operational conditions in low-frequency regions reveal the existence of special modes, unknown in "passive" structures. These modes are generated by solid/fluid interaction, e.g., in fluid-lubricated bearings and seals. During rotating machine performances, these modes show their activity through rotor self-excited vibrations (e.g., "oil whirl" is the rotor/bearing system self-excited vibration; "oil whirl resonance" and "oil whirl mode" are revealed by perturbation testing [2-7]).

Summarizing all these aspects, Modal Analysis and Modal Testing of rotating machines have to be focused on the lowest bending modes and applied to the rotor during normal operational conditions. The sophisticated Modal Testing, as used in case of "passive" structures, is not the most efficient for this purpose. Better results can be obtained by applying limited frequency sweep circular-force perturbation testing.

2. PERTURBATION TESTING OF ROTATING MACHINES

Perturbation modal testing is one of the commonly-used methods for parameter identification of mechanical modal structures. The method requires perturbing the dynamic equilibrium of the structure, represented by a "black box," by a known input force, and measuring the dynamic response of the structure. Most often the response represents motion, measured in terms of displacement, velocity, or acceleration, as functions of time (Fig. 1). The input force is applied to a selected point of the structure. Output measuring devices can be installed in several points of the structure, giving a set of structure point responses. Changing the point of force application, and measuring the structure response again, one can eventually get a matrix of responses corresponding to the 'vector' of inputs.

In the case of mechanical structures, the input is usually a force; the output is usually motion. The "black" (or "grey") box, representing the structure should then be described in terms of the structure Transfer Function with the units [Motion]/[Force]. In the linear case the Transfer Function represents a matrix, whose components could be real or complex numbers. The matrix order is equal to the number of points of force application and to the number of the assumed degrees of freedom of the structure.

Perturbation technique is widely applied for identification of "passive" structures, i.e., the structures whose dynamic equilibrium represents the static equilibrium. For "passive" structures, such as bridges, masts or buildings, the dynamic equilibrium means "no motion." A different situation occurs in the case of "active" struc-

PERTURBATION TECHNIQUE

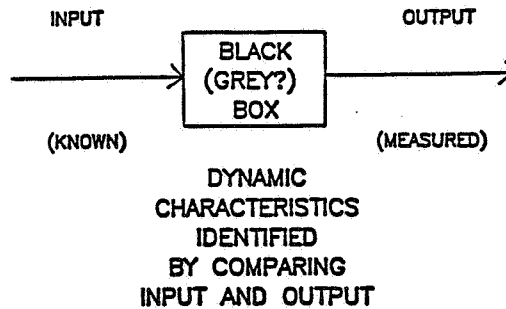


Figure 1. - Block diagram of a mechanic structure dynamic perturbation.

tures, such as rotating machines. Their dynamic equilibrium means rotation at a particular rotative speed, i.e., in the "active" structures there exists a continuous flow of kinetic energy. It is well known that due to several physical mechanisms the rotational energy of shafts can be transferred into various forms of vibrational energy of the shafts themselves (e.g., shaft lateral vibrations), supporting pedestals, cases, foundations, etc. It is then very important to know "how stable" the rotating machine dynamic equilibrium is. The perturbation method can give the answer to this question. Perturbation method applied to the "active structures" at their dynamic equilibrium give better evaluation of the structure parameters, as their value becomes affected by the shaft/rotational speed.

In "active" structures such as rotating machines, the best input force is a circular rotating force applied to the shaft in a plane perpendicular to the shaft axis. This force perturbs the shaft simultaneously in two lateral directions. Shaft motion in these two directions is closely coupled; a unilateral perturbation would result in shaft motion in both directions. An important advantage is related to the circular rotating force: the direction of its rotation can be chosen, i.e., the force can rotate "forward," in the direction of the shaft rotation, or "backward," in the opposite direction. For rotating structures the modes "forward" and "backward" are different. A unilateral shaft perturbation (or a perturbation applied to the pedestals or other nonrotating elements of the rotating machine) would result in mixed "forward" plus "backward" perturbation of the rotating shaft [8]. The shaft response will then contain both modes and the results might be ambiguous.

Another advantage of a circular rotating force is the ease of control of the force magnitude and phase by applying a controlled unbalance in the perturbation system and the ease of controlling its frequency, usually varying from about 20 rad/sec to a chosen value (sweep-frequency type of perturbation).

The perturbation system of a rotating machine can consist of an unbalanced rotating free spinner driven by compressed air blow (Fig. 2a) or can consist of an unbalanced perturbing shaft driven by a separate motor and attached to the main rotating machine shaft through a pivoting bearing (Figure 2b). These types of systems allow for "non-synchronous" shaft perturbation, i.e., the frequency (angular speed) of the perturbing force is entirely independent of the rotational speed of the main shaft. They also allow one to perturb the shaft either in a forward or a backward direction. Such perturbation systems also give very good results when the shaft does not rotate.

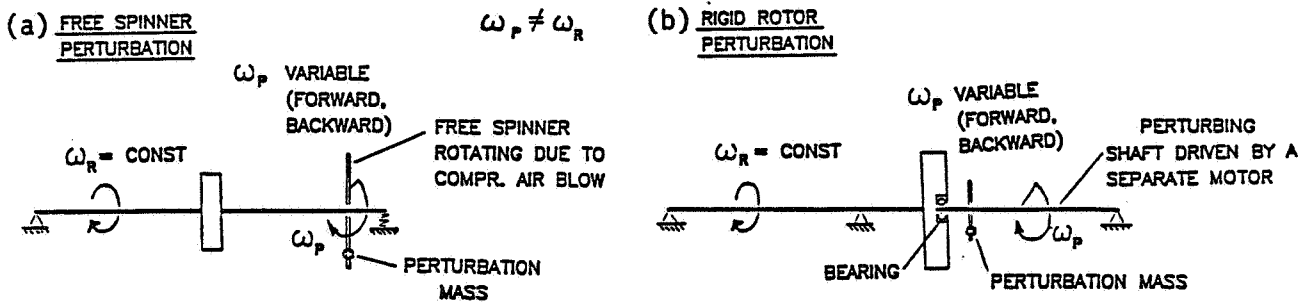


Figure 2. - Nonsynchronous perturbation of rotors.

"Nonsynchronous" perturbation, applied to a rotating machine, requires additional perturbing devices which, in case of big heavy machines, might be difficult to install. For some machines "synchronous" perturbation will give sufficient and very important information.

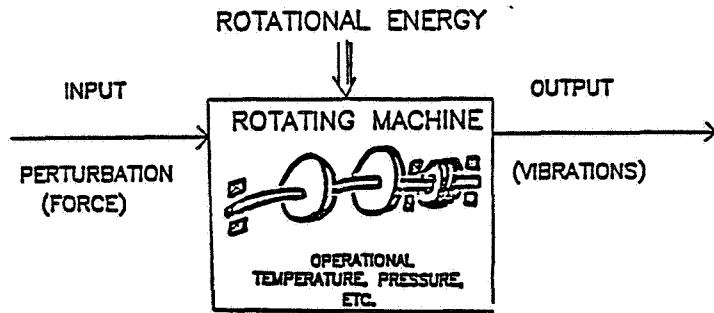
The "synchronous" perturbation force is created by a controlled unbalance introduced directly to the machine. The system response filtered to the synchronous component ($1 \times$), is measured during machine start-up or shutdown (for instance, this type of synchronous perturbation is used during balancing). By comparing the input force and the rotor output response, the rotor "synchronous" characteristics can be identified. In particular, in the case of the calibration weight balancing for one particular speed, the matrix of the rotor influence vectors representing the rotor transfer function as well as correcting weights can be calculated. Synchronous perturbation allows one to identify the rotor synchronous modal mass, stiffness and damping, by applying the Dynamic Stiffness Method [2,9]. This synchronous testing should be recommended for all rotating machines, as a routine practice.

In summary, the modal perturbation method is very efficient in the identification of dynamic characteristics of rotating machines (Fig. 3). "Nonsynchronous" perturbation should be superimposed on the rotational motion of the machine, while the machine operates at its normal conditions (including temperature, pressure, etc.). The best perturbing input for rotating machines is a rotating, circular unbalance force applied directly to the shaft. This force has slowly variable frequency (sweep method) and it can have direction "forward" or "backward" relative to the direction of the shaft rotation. The "synchronous" rotor perturbation, during start-up or shutdown, by a controlled unbalance, introduced directly to the rotor does not require any additional devices and it allows one to identify very important rotor "synchronous" modal dynamic characteristics.

3. RESULTS OF PERTURBATION TESTING OF ROTORS

3.1 Identification of Natural Frequencies as Functions of Rotative Speed for Rotors with Strong Gyroscopic Effects

Rotative energy of shafts has a significant influence on rotor dynamic characteristics. In particular, rotating shaft natural frequencies differ from nonrotating shaft natural frequencies. The perturbation testing has been used for determining the relationship between the rotor natural frequencies and rotative speed. The rig with a rotor showing strong gyroscopic effect presented in Fig. 4 yields the results of forward and backward perturbation given in Fig. 5 [2,10].



- PERTURBATION BY CONTROLLED UNBALANCE IS EASY TO GENERATE, CONTROL, AND MEASURE
- CIRCULAR PERTURBATION IN THE PLANE PERPENDICULAR TO SHAFT AXIS PROVIDES THE BEST ROTOR DYNAMIC BEHAVIOR INSIGHT
- CIRCULAR PERTURBATION CAN BE APPLIED TO THE ROTOR IN FORWARD OR REVERSE DIRECTION (RELATIVE TO SHAFT ROTATION)

Figure 3. - Perturbation technique for rotating machinery.

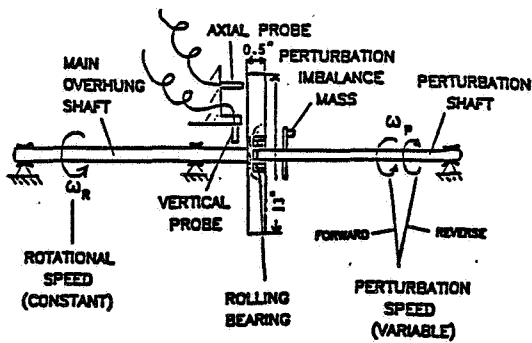


Figure 4. - Gyroscopic effect test rig (nonsynchronous perturbation method).

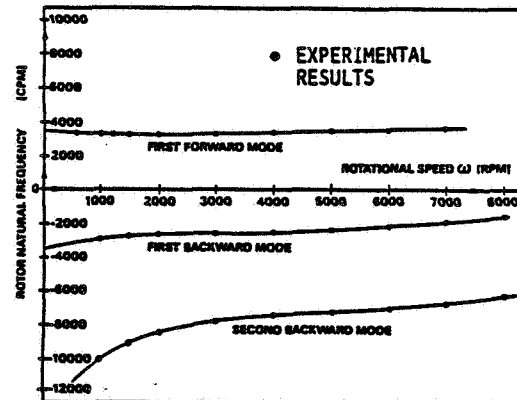


Figure 5. - Rotor natural frequencies versus rotative speed. Second forward mode in higher range of frequencies.

3.2 Identification of Rotor First Mode Parameters by Perturbation Testing (Dynamic Stiffness Method)

Synchronous perturbation combined with Dynamic Stiffness Method testing was successfully used for the identification of Rotor First Bending Mode generalized parameters (Fig. 6) [2,9]. When applied to a system with two degrees of freedom (rotor horizontal and vertical displacements), the Dynamic Stiffness Method clearly yields the system modal (generalized) parameters. Figs. 7 and 8 present the results for the vertical mode. The controlled unbalance was introduced to the rotor disk at a chosen location for the first run, then the same unbalance was placed at the same radius, 180° from the previous location for the second run. The results of these two runs are then vectorially subtracted. This eliminates all existing residual synchronous effects, and the results of the perturbation testing are very clean.

3.3 Identification of Bearing Fluid Dynamic Forces

The nonsynchronous perturbation method proved to be very efficient for the identification of bearing fluid dynamic forces [2,7]. This method was also applied by several researchers in order to identify the radial and tangential fluid forces acting on impellers in centrifugal flow pump [11,12]. Fig. 9 presents the rotor rig used for identification of bearing fluid dynamic forces. Figs. 10 and 11 illustrate some results of low frequency perturbation, covering oil whirl resonance. The perturbation testing yielded several important conclusions, concerning dynamic behavior of rotor/bearing/seal systems. These include a determination of oil whirl resonance frequency as being a rotor/bearing system natural frequency and existing only for forward directions, a specific relationship between bearing coefficients (such as "cross" stiffness proportional to radial damping, rotative speed, and average oil swirling ratio), a significant fluid inertia effect, stability margin, stabilizing effect of high oil pressure (Fig. 12), as well as allowed for full identification of the bearing fluid force coefficients.

The perturbation testing covering higher frequency range reveal both oil whirl and oil whip resonances as the rotor/bearing system characteristics as well as the specific whirl mode, governed by bearing radial damping. The Dynamic Stiffness Technique allowed for the identification of the rotor/bearing system parameters, and creation of an adequate rotor/bearing model [7] (Figs. 13-15).

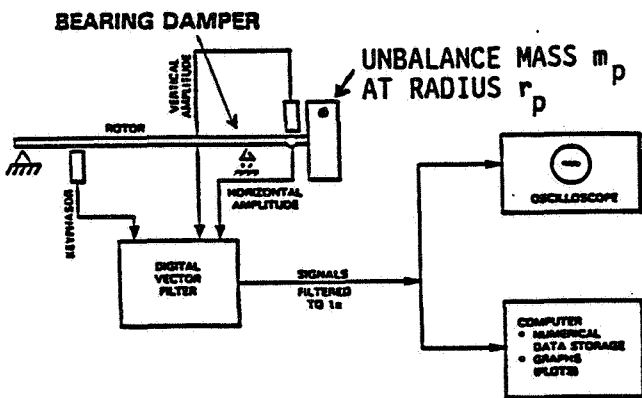


Figure 6. - Synchronous perturbation and dynamic stiffness method applied for identification of rotor first bending mode vertical and horizontal parameters.

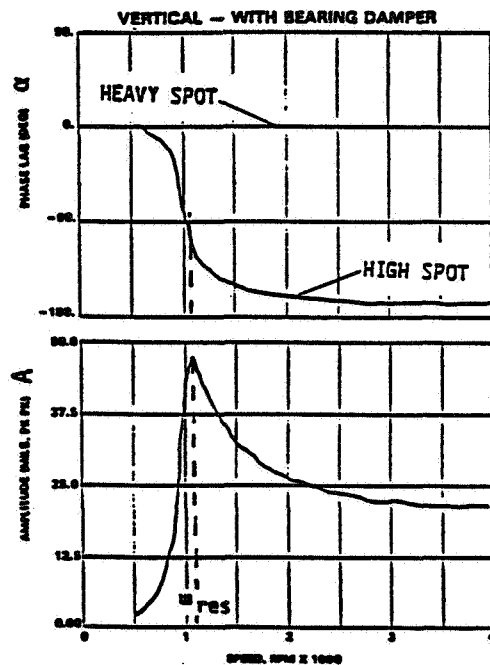


Figure 7. - Rotor vertical response: phase and amplitude versus rotative speed.

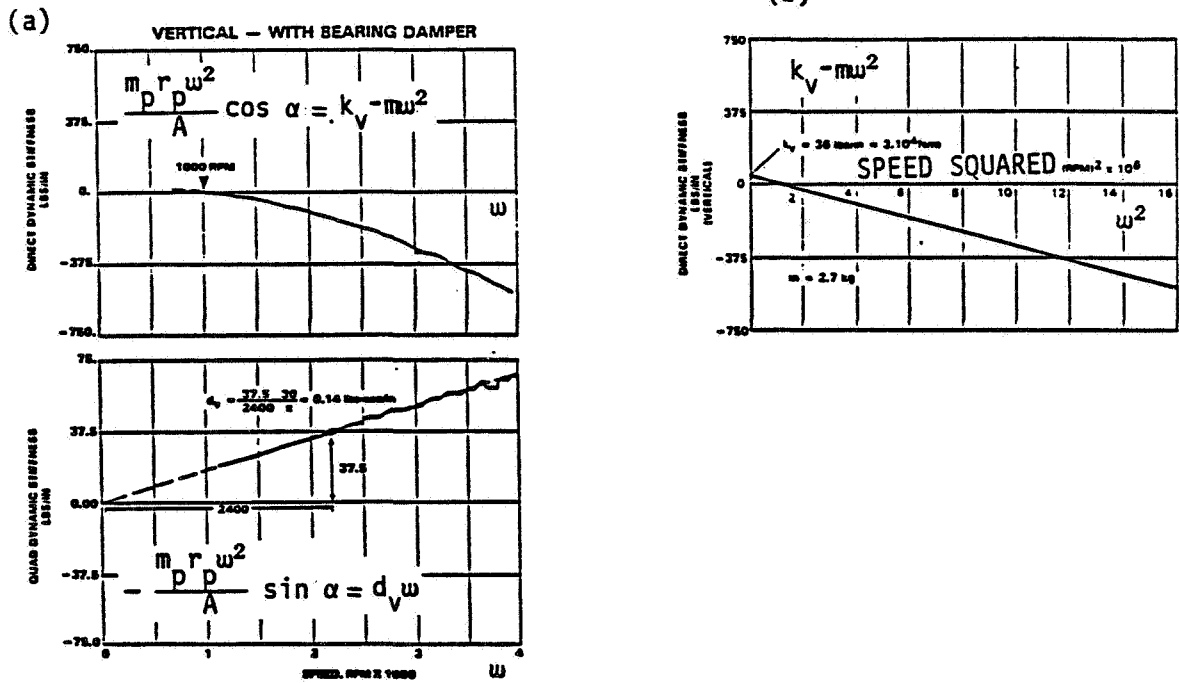


Figure 8. - Vertical dynamic stiffnesses versus rotative speed (a) and versus rotative speed squared (b). Identification of modal vertical damping coefficient d_v , stiffness k_v , and mass m .

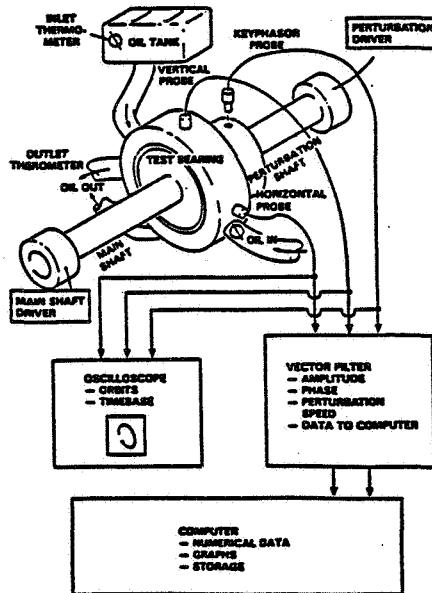


Figure 9. - Scheme of the perturbation testing for bearing fluid dynamic force identification.

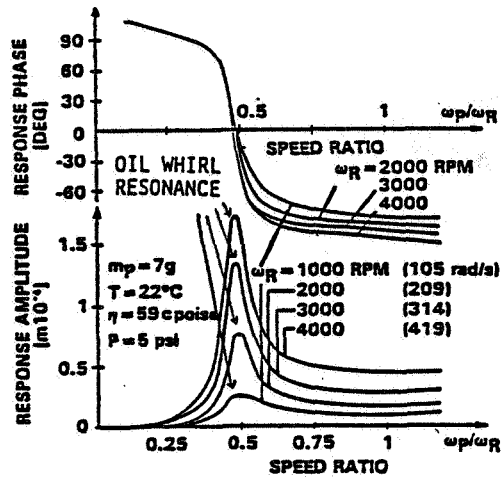


Figure 10. - Phase and amplitude of rotor response to forward perturbation versus perturbation speed ratio for several values of rotative speed ω_R . Oil whirl resonance revealed at frequency of approx $0.48 \omega_R$. At low perturbation speeds phase of response is leading (force phase angle, 0°).

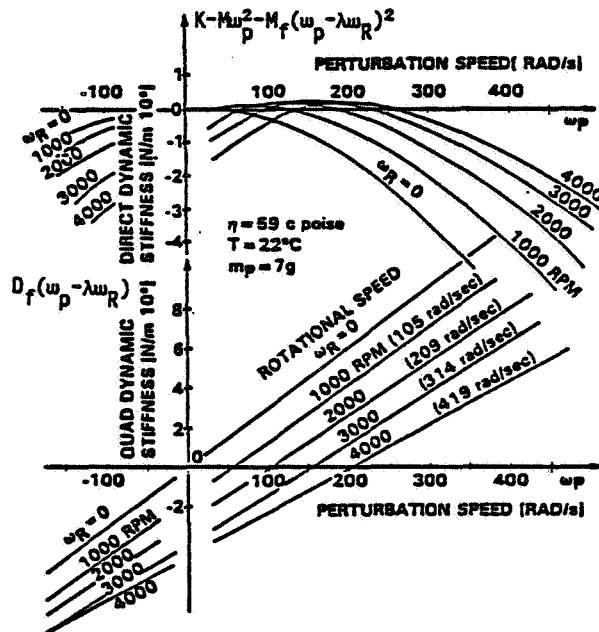


Figure 11. - Direct and quadrature dynamic stiffnesses versus perturbation speed: identification of rotor/bearing system parameters. Cross stiffness generated by radial damping; cross damping a function of fluid inertia. (M_f denotes fluid inertia; D_f , fluid radial damping coefficient; λ , fluid average swirling ratio; K and M , rotor modal parameters.)

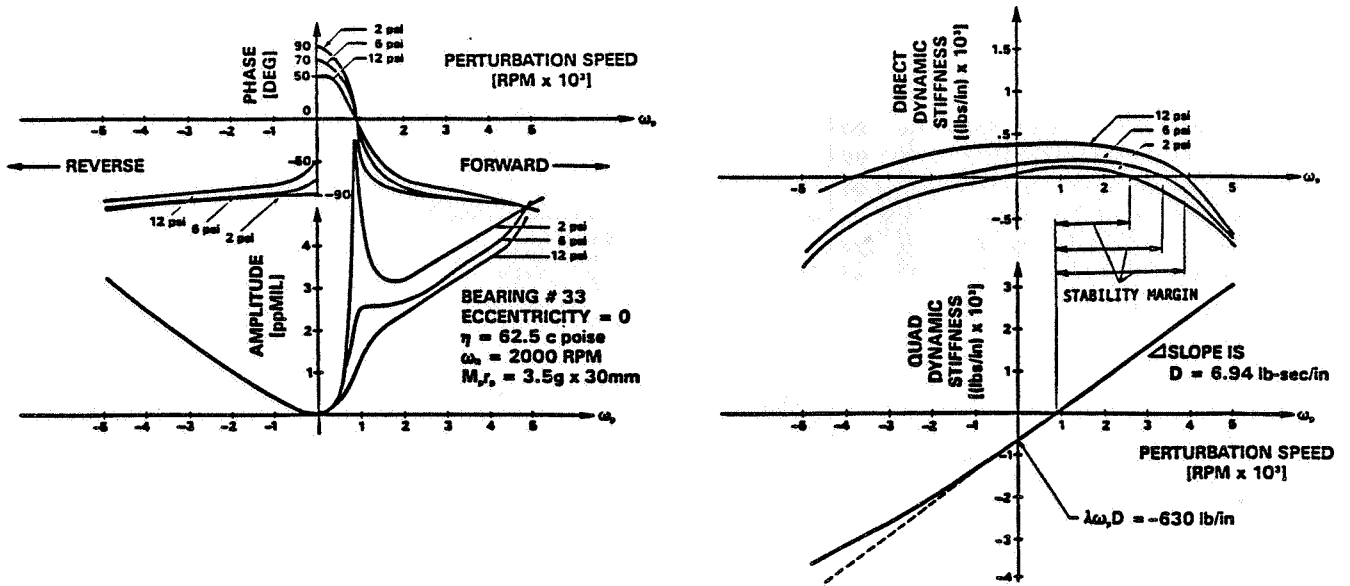


Figure 12. - Phase, amplitude, and dynamic stiffnesses versus perturbation speed for several values of oil pressure: increase of stiffness and stability margin for higher pressure.

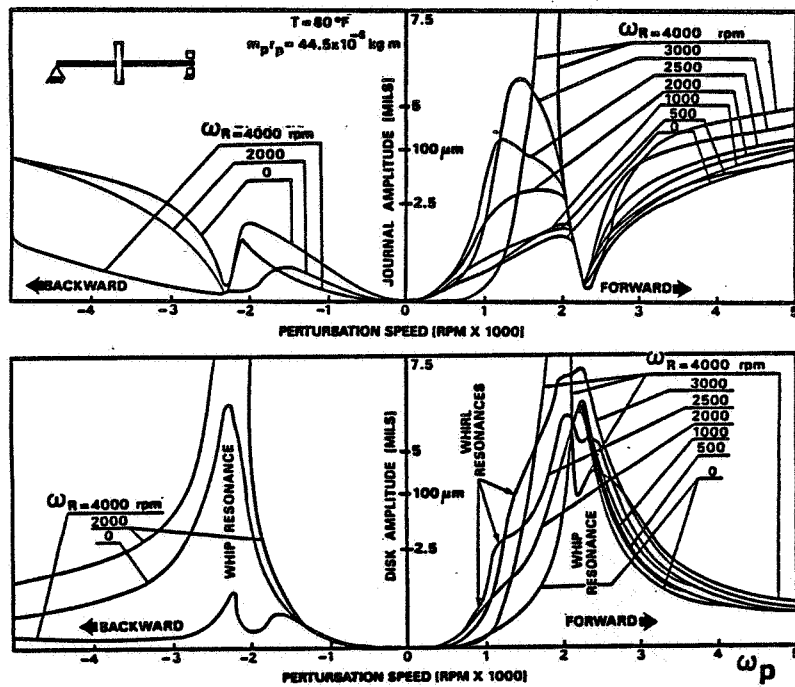


Figure 13. - Journal and disk response amplitude versus perturbation frequency ω_p for several values of rotative speed ω_R . Whirl resonance exists only for forward perturbation. Whip resonance (corresponding to rotor first bending mode) exists for both forward and backward perturbation.

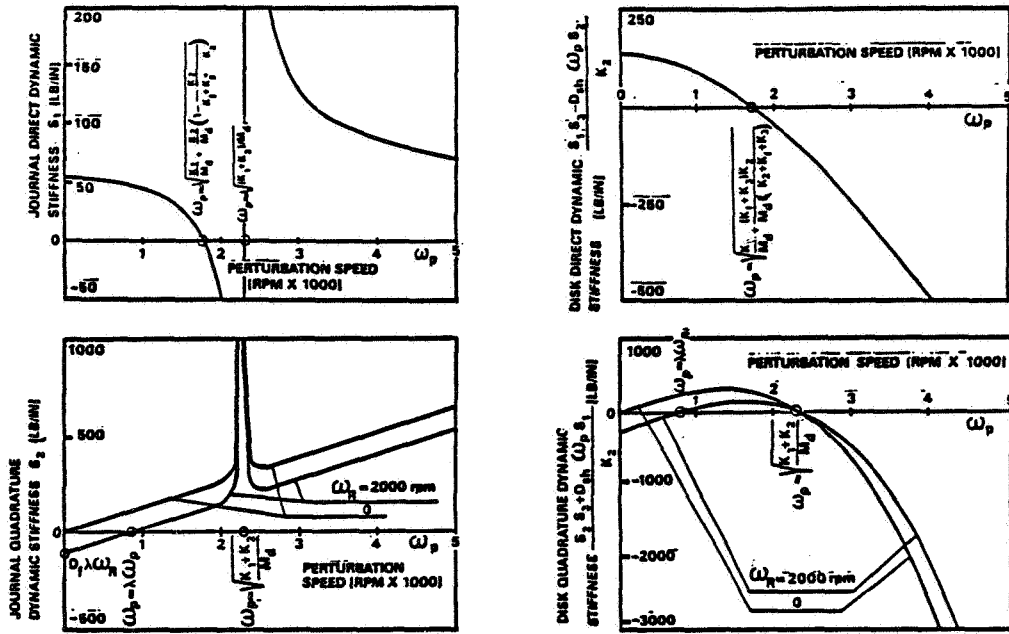
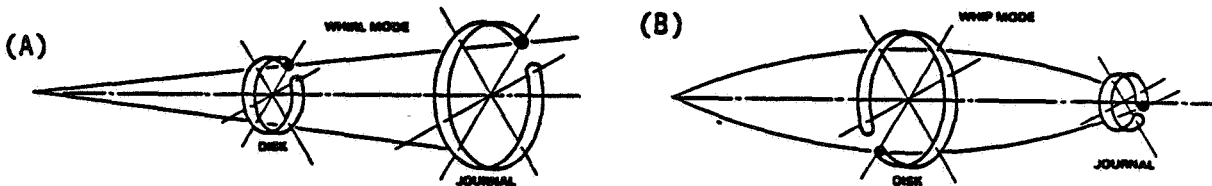


Figure 14. - Journal and disk dynamic stiffnesses versus perturbation frequency ω_p . Rotative speed ω_R affects quadrature dynamic stiffness only.

$$[S_1 = K_f + K_3 + K_2 - M_f(\omega_p - \lambda\omega_R)^2 - M_b\omega_p^2 - K_2^2 S_3 / (S_3^2 + D_{sh}^2 \omega_p^2), S_2 = D_f(\omega_p - \lambda\omega_R) + K_2^2 D_{sh} \omega_p / (S_{sh}^2 + D_{sh}^2 \omega_p^2), S_3 = K_1 + K_2 - M_d \omega_p^2].$$



(a) Whirl mode, disk and journal motion in phase.
 (b) Whip mode, journal 90° ahead of disk motion.

Figure 15. - Modes of rotor/bearing system revealed by perturbation testing.

SYMBOLS

A	Rotor amplitude of response
D_f, K_f, M_f	Bearing fluid damping, stiffness and inertia coefficients
D_{sh}	Rotor external viscous damping coefficient
K, K_1, K_2	Shaft stiffnesses
K_3	External spring stiffness
m_p, r	Perturbation mass and radius of unbalance correspondingly
M_b^p, M_d^p, M_d	Modal masses of the rotor

α	Phase of Response
η	Oil dynamic viscosity
λ	Oil average swirling ratio
w	Perturbation speed
w_R^p	Rotative speed of the main rotor

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