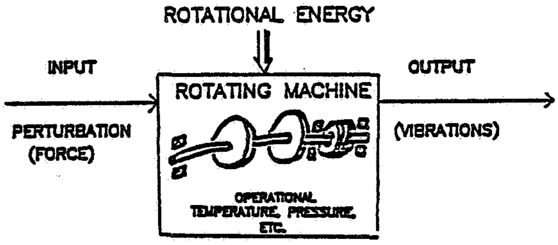
N86-30201

ROTOR/BEARING SYSTEM DYNAMIC STIFFNESS MEASUREMENTS

BY NONSYNCHRONOUS PERTURBATION

Agnes Muszynska
Bently Rotor Dynamics Research Corporation
Minden, Nevada 89423

Sweep perturbation testing as used in Modal Analysis when applied to a rotating machine has to take into consideration the machine dynamic state of equilibrium at its operational rotative speed. This stands in contrast to a static equilibrium of nonrotating structures (Fig. 1). The rotational energy has a significant influence on rotor dynamic characteristics. The best perturbing input for rotating machines is a forward or reverse rotating, circular force applied directly to the shaft (Fig. 2).



ADVANTAGES OF PERTURBATION BY USING A ROTATIONAL HARMONIC FORCE

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

Figure 1. - Perturbation technique for rotating machines.

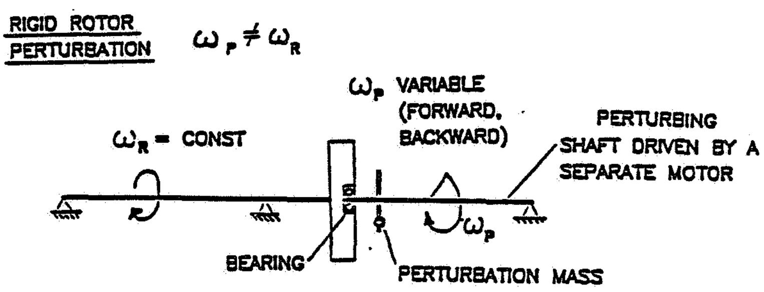


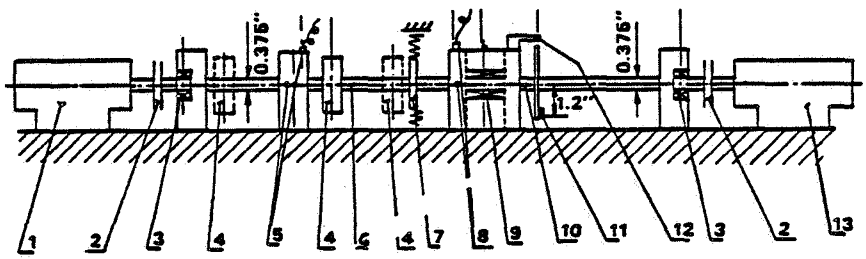
Figure 2. - Nonsynchronous perturbation of rotating machine through auxiliary perturbing shaft with controlled unbalance.

OBJECTIVE

Determination of Dynamic Stiffness Characteristics of the rotor bearing system by nonsynchronous perturbation of a symmetric rotating shaft supported in one relatively rigid and one oil lubricated bearing.

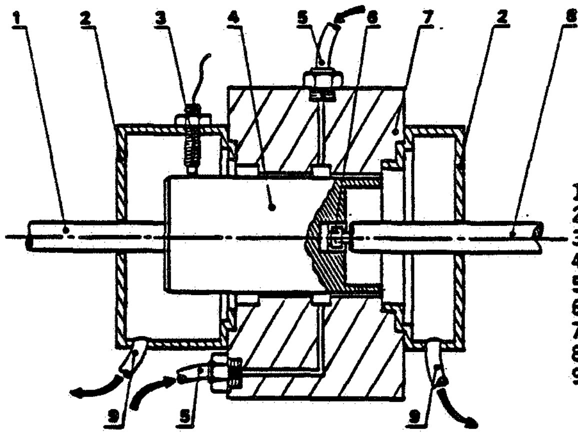
EXPERIMENTAL ROTOR RIG

The experimental rotor system consists of the main shaft rotating at constant angular speed (ω_R) and the auxiliary shaft generating a rotating perturbing force with sine-sweep variable frequency (ω_p). This force is applied to the main journal (Figs. 3 and 4).



1-MAIN ROTOR; 2-FLEXIBLE COUPLINGS; 3-RIGID SUPPORTS WITH PIVOTING BEARINGS; 4-DISK WITH THREE OPTIONAL POSITIONS ON THE SHAFT; 5-HORIZONTAL AND VERTICAL NONCONTACTING PROBES; 6-MAIN SHAFT; 7-ROTOR WEIGHT BALANCING SPRINGS; 8-HORIZONTAL AND VERTICAL PROBES; 9-OIL LUBRICATED BEARING; 10-PERTURBATION SHAFT; 11-PERTURBATION UNBALANCE MASS; 12-KEYPHASOR PROBE; 13-PERTURBATION MOTOR

Figure 3. - Rotor rig for perturbation testing.



1-MAIN SHAFT
2-PLEXIGLAS END CAPS
3-VERTICAL PROBE
4-MAIN SHAFT JOURNAL
5-OIL INLET
6-KNUCKLE PIVOTING BEARING
7-ALUMINUM BEARING HOUSING
8-PERTURBATION SHAFT
9-OIL OUTLET

Figure 4. - Oil bearing used for perturbation testing.

INSTRUMENTATION

Primary instruments:

- Four eddy current proximity probes for measuring the journal and disk horizontal and vertical displacements
- Keyphasor narrow beam optical probe for phase measurements
- Digital Vector Filter for filtering 1× response during perturbation testing
- Spectrum analyzer for yielding amplitude/frequency spectra of nonperturbed rotor runs
- Oscilloscope for continuous observation of the rotor motion
- Hewlett Packard 9836 computer with Bently Nevada software for data acquisition, storage, reduction, analytical computation and displays.

Peripheral instruments:

- Speed controllers for rotation and perturbation frequency
- Oil supply system including filter and pressure control
- Oil heating/cooling device
- Three thermometers (at oil inlet, outlet and in bearing housing).

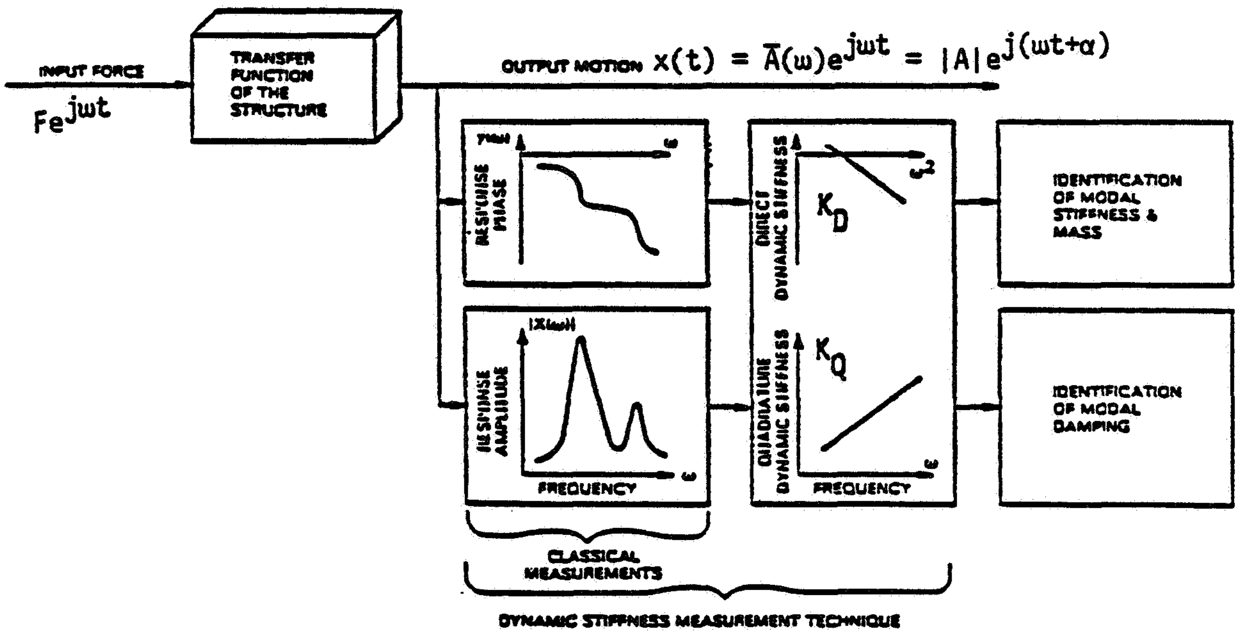


Figure 5. - Dynamic stiffness measurement technique by perturbation testing.

HINTS TO MAINTAIN DATA ACCURACY

- Rigid rotor support, concrete foundations, rigid auxiliary fixtures (elimination of additional degrees of freedom).
- Balanced rotor system.
- Balanced perturbation system, with precisely known perturbation imbalance introduced later.
- Rotor centered in the test bearing by adjustment of the external supporting springs for all deliberately centered tests.

- Control of oil temperature at the bearing to ±1°C.
- Accurate control of constant rotative speed ω_R.
- Slow ramp of the perturbation speed.

EXPERIMENTAL PARAMETERS

- Main rotor rotational speed: 0 (squeeze film test) 500, 1000, 1500, 2000, 2500, 3000, 3500, 4000 rpm.
- Perturbation speed range: -6000 to 6000 rpm (minus indicates reverse direction).
- Perturbation imbalance masses: m_p = 0.75, 2.35, and 7 grams. (Low mass was used for better evaluation of resonant amplitudes; high mass was used to increase the sensitivity in the out-of-resonance regions.)
- Perturbation radius: r_p = 1.2" (0.03 m)
- T-10 oil density = 794 kg/m^3
- Oil temperature: 80°F (26.7°C)
- Corresponding oil dynamic viscosity: 7.25×10^{-6} lbs s/in^2 (50 c poise).
- Oil pressure: 3 psi (20684 Pa). The bearing is center fed and has two-directional axial flow. A circumferential oil feed groove and relatively low, well controlled pressure enable a uniform axial pressure to be maintained in the bearing.
- Main steel shaft weight (including aluminum journal): 0.614 lbs.
- Perturbation aluminum shaft weight: 0.181 lbs.
- Disk weight: 1.79 lbs (disk mass 4.64×10^{-3} lbs s^2/in = 0.81 kg).
- Aluminum bearing radial clearance: 0.0075" (190 μm).
- Bearing length: 0.5" (0.013 m)
- Bearing diameter: 1.0" (0.0254 m)
- Bearing radial clearance to radius ratio: 0.015.

RESULTS (Example)

The perturbation testing yields response data in form of vibration amplitude and phase, which are eventually used to obtain Direct and Quadrature Dynamic Stiffness characteristics of the rotor/bearing system. The latter permits the identification of the system modal stiffness, mass and damping.

The results of the perturbation tests are presented in Figures 6 through 10 in the form of Bodé plots, polar plots, and Dynamic Stiffness versus perturbation speed plots.

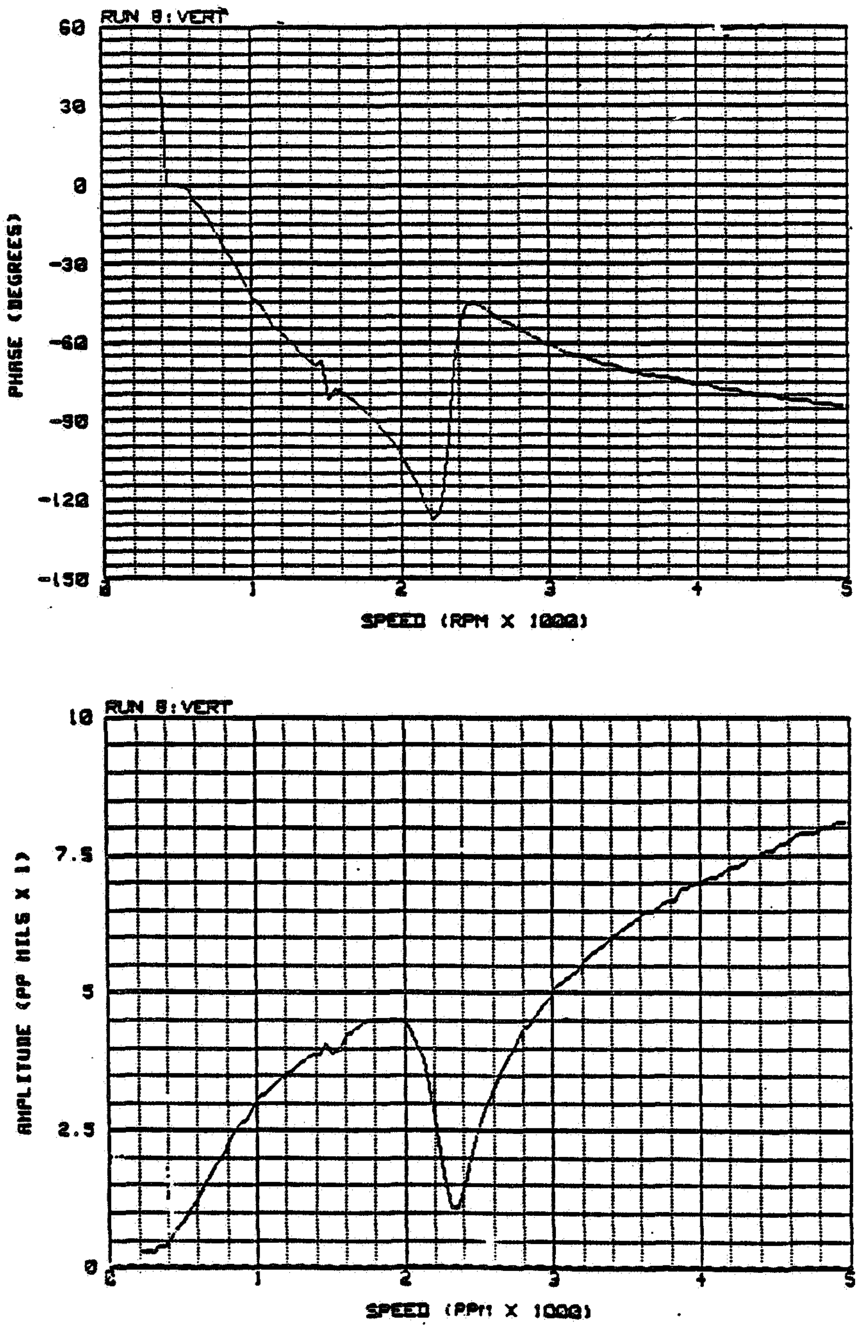


Figure 6. - Journal phase and amplitude of vertical response. Rotative speed, 1500 rpm; m_p, 1.6 g; T, 80 °F; forward perturbation.

ORIGINAL PAGE IS
OF POOR QUALITY

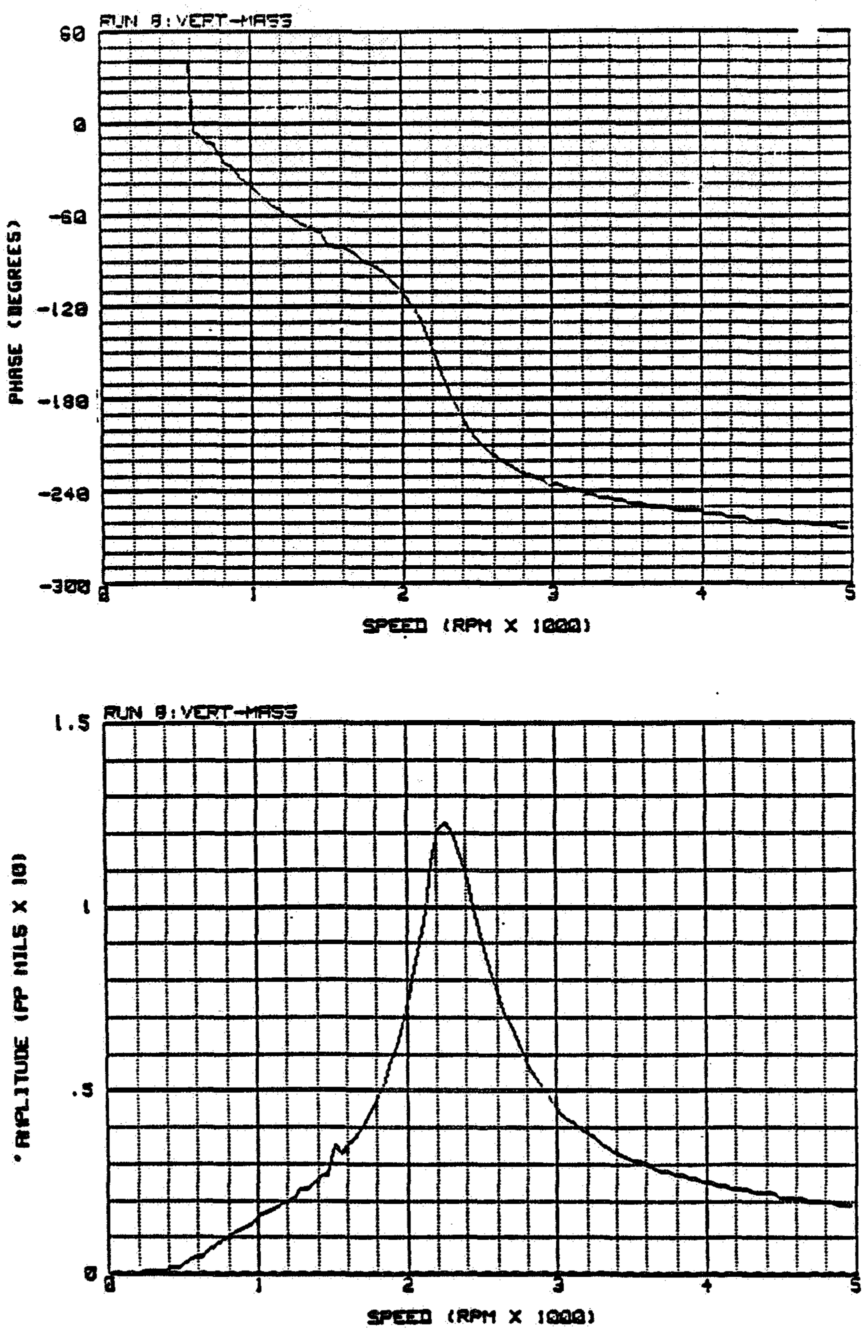
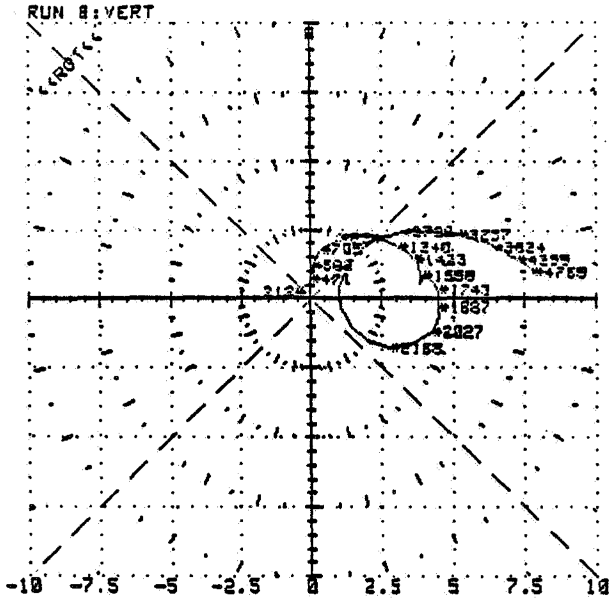


Figure 7. - Disk phase and amplitude of vertical response (parameters same as in fig. 6).



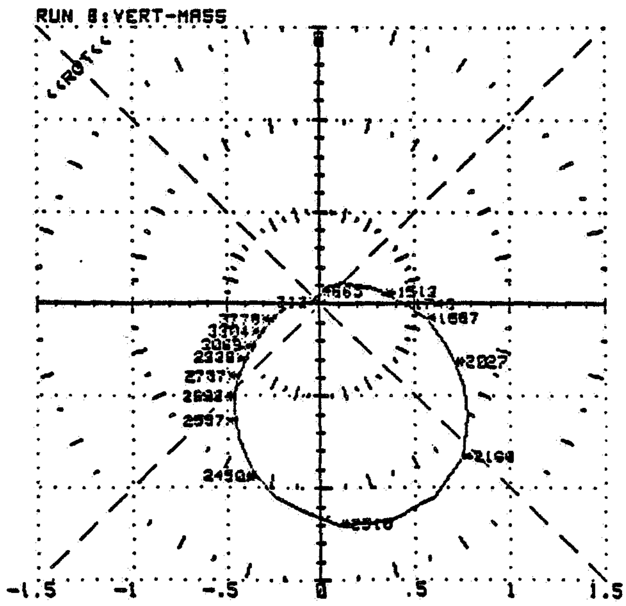


Figure 8. - Journal and disk response in form of polar plots (parameters same as in fig. 6).

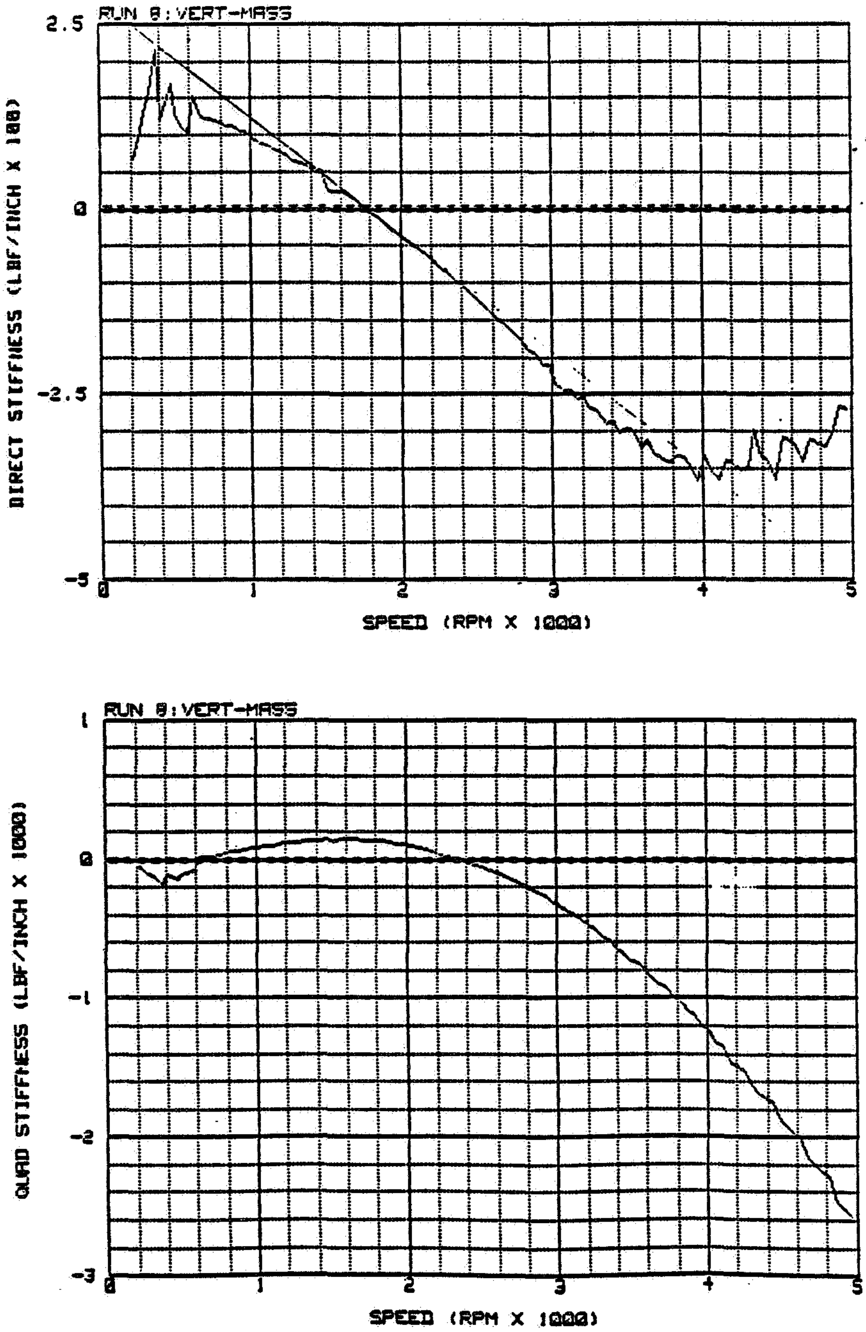


Figure 9. - Rotor/bearing dynamic stiffnesses (journal) (parameters same as in fig. 6).

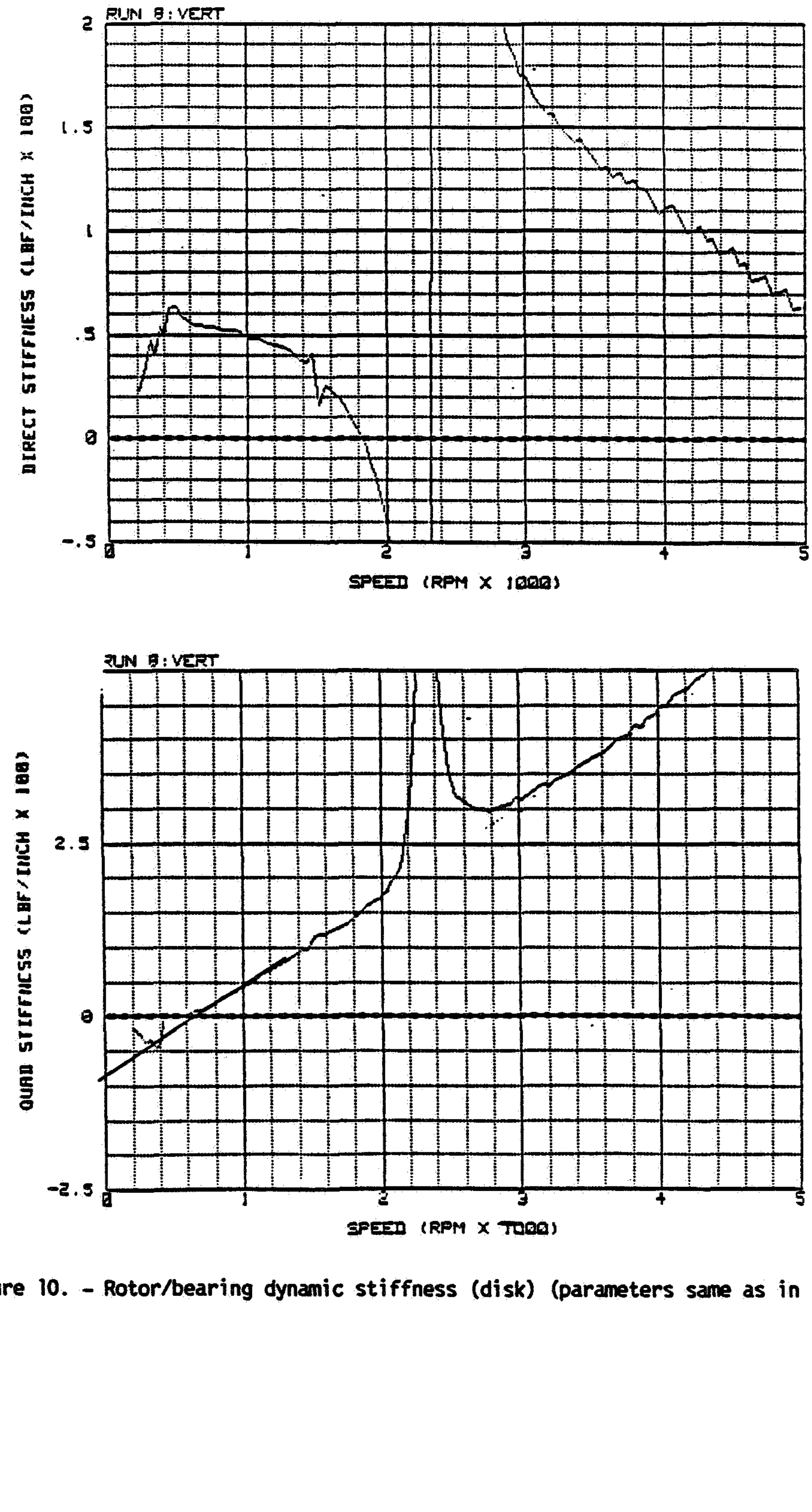


Figure 10. - Rotor/bearing dynamic stiffness (disk) (parameters same as in fig. 6).

REFERENCES

1. Muszynska, A.: Dynamic Stiffness Measurements for Better Mechanical System Identification. Bently Nevada Corporation, Paper presented at EPRI-Nuclear Power Division Pump Workshop, Toronto, Canada, August 1982.

2. Muszynska, A.: Application of the Perturbation Method to Rotating Machines. BNC Senior Mechanical Engineering Seminar, Carson City, NV, June 1984.

3. Bently, D.E., Muszynska, A., Olas, A.: Identification of the Parameters of a Rotor with the Strong Gyroscopic Effect by Perturbation Testing, Bently Nevada Corporation, 1984.

4. Bently, D.E., Muszynska, A.: Stability Evaluation of Rotor/Bearing System by Perturbation Tests, Rotordynamic Instability Problems in High Performance Turbomachinery, Proc. of a Workshop, Texas A&M University, College Station, TX, 1982.

5. Bently, D.E., Muszynska, A.: Oil Whirl Identification by Perturbation Test. Advances in Computer-aided Bearing Design, ASME/ASLE Lubrication Conference, Washington, DC, October 1982.

6. Bently, D.E., Muszynska, A.: Perturbation Tests of Bearing/Seal for Evaluation of Dynamic Coefficients. Symposium on Rotor Dynamical Instability, Summer Annual Conference of the ASME Applied Mechanics Division, Houston, TX, June 1983.

7. Bently, D.E., Muszynska, A.: Perturbation Study of Rotor/Bearing System : Identification of the Oil Whirl and Oil Whip Resonances. Tenth Biennial ASME Design Engineering Division Conference on Mechanical Vibration and Noise, Cincinnati, OH, September 1985.

8. Bently, D.E., Muszynska, A.: The Dynamic Stiffness Characteristics of High Eccentricity Ratio Bearings and Seals by Perturbation Testing. Workshop on Rotordynamic Instability Problems in High Performance Turbomachinery, Texas A&M University, May 1984.