N86-30191

REMARKS ON ROTOR STABILITY (A CONTRIBUTION TO DISCUSSION DURING SYMPOSIUM ON INSTABILITY IN ROTATING MACHINERY)

E. Krämer
Technical University of Darmstadt
Darmstadt, Federal Republic of Germany

The remarks refer to two subjects:

1. Influence of nonlinear factors in fluid-lubricated bearings on rotor vibrations. (Figures 1 through 8)

2. Computation of threshold of stability for systems with multiple degrees of freedom. (Figures 9 and 10)

Notes on the first subject are taken from [1]. The thesis [1] is written in German; therefore, it is useful to present here some major contributions from this publication. The reason for this investigation was to learn more about the influence of tilting pad bearings on vertical rotors. The eigenvalues, the stability threshold, the response on the Heaviside-function and the unbalance response were calculated for a rigid rotor, a Jeffcott rotor and for the finite element model of a real rotor. The comparison of the linear and nonlinear calculations shows the limits of the linear calculations.

The second contribution is suppose to show, that the model of the Jeffcott rotor is not sufficient in most practical cases to determine the stability threshold. This is known since some years, when computed results were compared with the behavior of real machines. That's why a program was developed, which is roughly described in the following:

Finite Element Program "MADYN"

Bar elements, rigid bodies, journal fluid-lubricated bearings. Elements may be arbitrarily located and orientated in space. Free matrices for special cases. Maximum 2000 degrees of freedom.

Computations

Eigenvalues (complex), eigenvectors, harmonically forced vibrations, transient vibrations, earthquake excitation, random vibrations.

Procedures

Householder, Hessenberg, Inverse Vector Iteration, Complex Gauss, Modal Analysis (right-hand and left-hand eigenvectors), step by step methods.

Developed at the Institute of Machine Dynamics of the Technical University of Darmstadt during 1972-1982 (leader of the project: H. D. Klement, Ph.D.).

As an example fig. 10 shows the real part of the eigenvalue, determining the stability threshold of the turbogenerator shown in fig. 9 as a function of the rotational speed. A positive real part means that the rotor is unstable. It can be seen that the threshold depends to a high extent on how the foundation was taken into account for the calculation.

Final remark:

My opinion about instability problems in rotating machinery is that the most efficient way to solve them is to consider both analytical computations and measurement results.

When computation is not used, very limited insight to the problem is gained. On the other hand, without measurements one cannot learn about the dynamic behavior of real machines. Both, measurements and computations, are necessary. An intelligent comparison and correlation of measurement and computational results will certainly positively contribute to the increased knowledge of the causes of instability in rotating machinery.

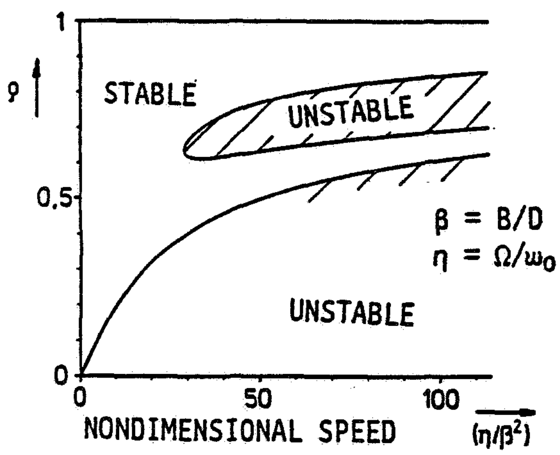


Figure 1. Stability map for a rigid shaft with circular fluid-lubricated bearings. η/β^2 is nondimensional speed, ρ is eccentricity ratio.

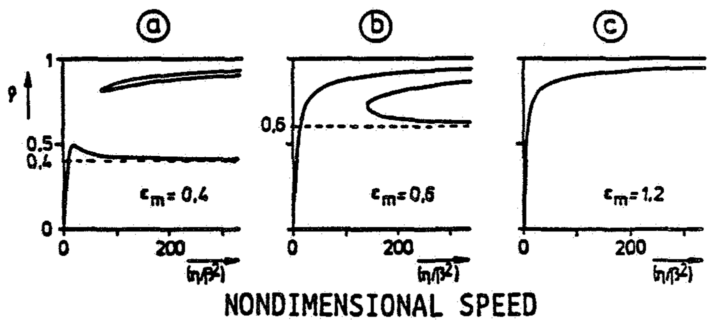


Figure 2. Radius of shaft orbit versus rotative speed. Possible cases. Rigid shaft. Zero static load. ε_m = mass eccentricity/clearance.

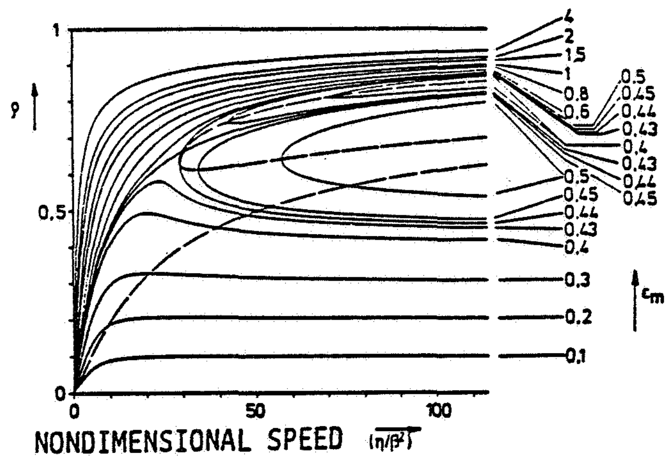


Figure 3. Details of the cases presented in Figure 2.

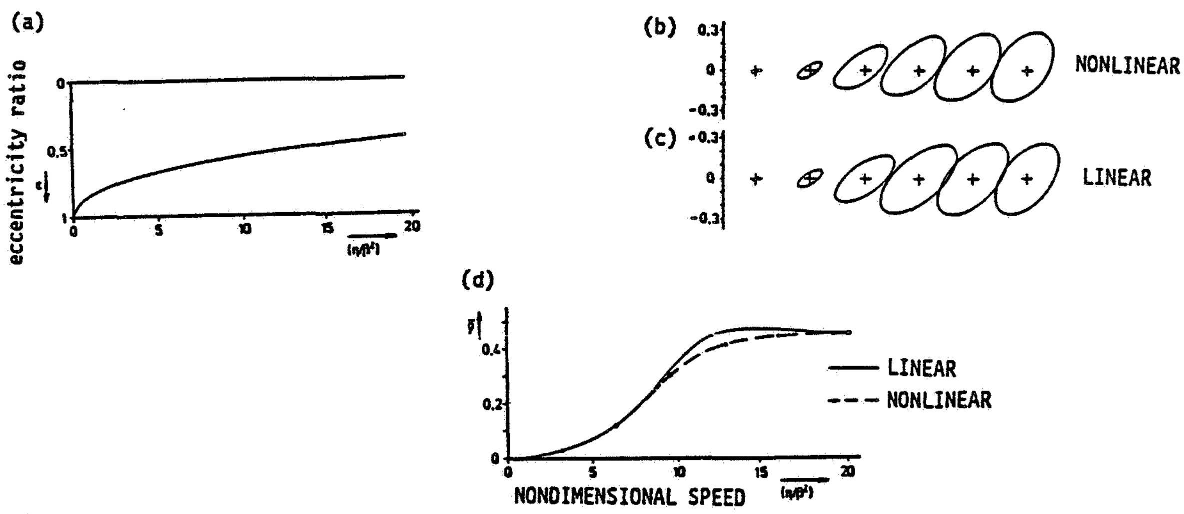


Figure 4. Orbits of a rigid shaft versus rotative speed. Circular fluid-lubricated bearings. Effect of unbalance (ε_m=0.2) and static force. (a) Location of orbit center, (b) Orbits (nonlinear), (c) Orbits (linear), (d) Mean orbit radius.

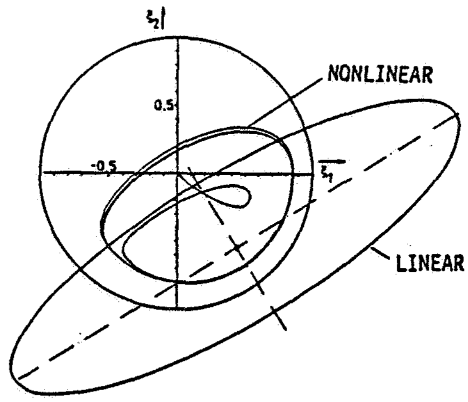


Figure 5. Orbit of a rigid shaft with unbalance and static force. Comparison between the results yielded by linear and nonlinear models.

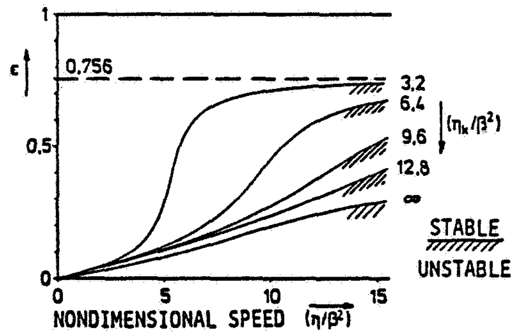


Figure 6. Stability chart for elastic shafts with circular fluid-lubricated bearings. Static position of shaft center.

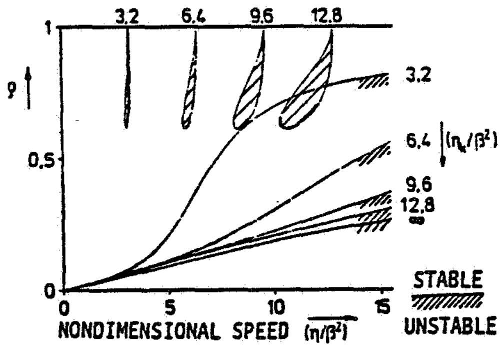
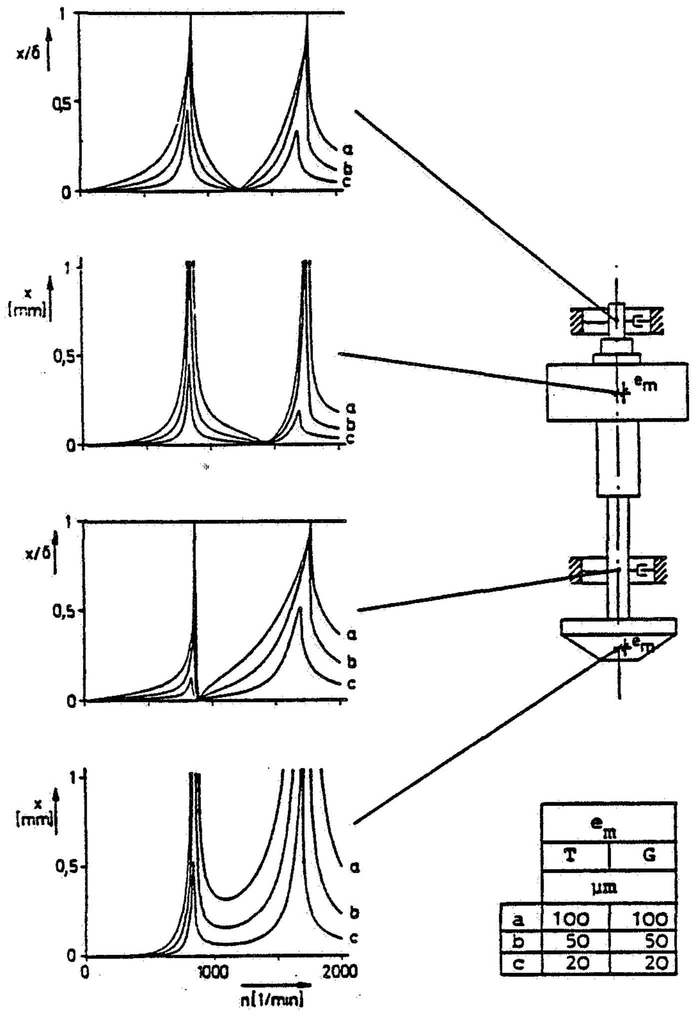


Figure 7. Stability chart for elastic shafts with fluid-lubricated bearings. The shaft center moves on a circular concentric path with relative radius ρ.



	e_m	
	T	G
	µm	
a	100	100
b	50	50
c	20	20

Figure 8. Amplitudes of a 200 MW pump turbine. Results of computation for nonlinear cases a, b, c, corresponding to various amounts of unbalance.

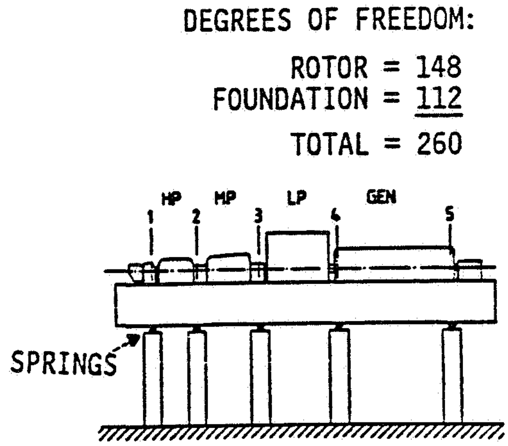


Figure 9. 350 MW turbogenerator. Model: Shaft and foundation have 260 degrees of freedom (condensed). Length = 32 m.

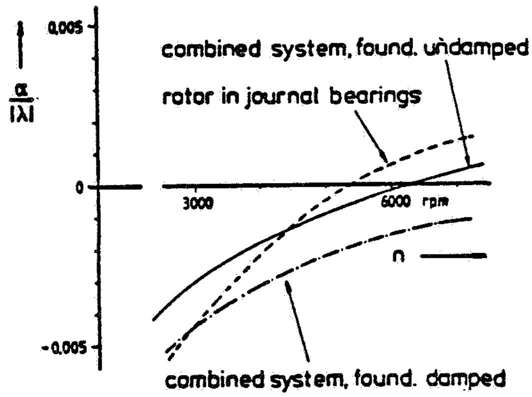


Figure 10. Oil whip instability of system presented in Figure 9 for different models of foundation. Real part of eigenvalue determines stability threshold.