to unbalance.

2 Bearings, pedestals, seals, couplings, and so on, play a major role in dynamic response of turbomachinery and therefore must be included in analysis.

3 Rigorous dynamic analysis provides direct insight into machinery reliability, acceptance testing, direction of development program, location of balancing planes, and instrumentation.

4 Machinery such as high-pressure, multistage compressors of barrel construction have difficult access to midplanes for in-the-field multiplane balancing.

5 Machinery which is well balanced in the factory can get out of balance with time in the field as a result of design and environment conditions; e.g., corrosion, erosion, and deposits.

6 Rotors supported in pivoted-shoe bearings are not necessarily inherently stable. Method of shoe restraint, pivot stiffness, shoe mass, and inertia must be carefully considered in assessing stability.

7 Viscous seals improperly designed can cause severe rotor instability even with properly designed pivoted-shoe bearings.

Recommendations

1 Rigorous dynamic analysis of the machine and the coupled system should be performed by the manufacturer in order to insure adequacy of design.

2 In applications where reliability and long-life operation is required, the contractors and the machinery users should impose both mechanical and aerodynamic performance specifications on the machinery builders.

3 Adequate, rugged instrumentation located in the proper places should be used for monitoring machinery performance in the field. It is preferable for the builder to install this instrumentation during machinery construction. In this way, it can be used in acceptance testing and will be directly correlated with field experience.

4 In machines which have access only to two balancing planes, e.g., barrel compressors, every effort should be exercised to design them to operate below the second flexural critical. If this is not possible, then careful attention must be maintained in assessing the rotor response to unbalance.

5 Maintenance specifications and provision for in-the-field balancing should be factored into the design. Adequate damping must be designed to tolerate the vibration of an unbalanced rotor.

References

1 Sternlicht, B., Poritsky, H., and Arwas, E., "Dynamic Stability Aspects of Cylindrical Journal Bearings Using Compressible and Incompressible Fluids," First International Symposium—Gas Lubricated Bearings, ONR/ACR-49.

2 Lund, J. W., and Sternlicht, B., "Rotor-Bearing Dynamics With Emphasis on Attenuation," *Journal of Basic Engineering*, Trans. ASME, Series D, Vol. 84, No. 4, Dec. 1962, pp. 491–502.

3 Muster, D., and Sternlicht, B., *Proceedings—International Symposium on Lubrication and Wear*, McCutchan Publishing, 1965.

4 Lund, J. W., "Self-Excited, Stationary Whirl Orbits of a Journal in a Sleeve Bearing," PhD thesis, Rensselaer Polytechnic Institute, May 16, 1966.

5 Prohl, M. A., "A General Method for Calculating Critical Speed of Flexible Rotors," *Journal of Applied Mechanics*, Vol. 12, No. 3, Trans. ASME, Vol. 67, Sept. 1945, pp. A-142–A-148.

6 Myklestad, N. O., "New Method of Calculating Natural Modes of Uncoupled Bending Vibration of Airplane Wings and Other Types of Beams," *Journal of the Aeronautical Sciences*, Vol. 11, No. 2, Apr. 1944, pp. 153–162.

7 Boeker, G. F., and Sternlicht, B., "Investigation of Translatory Fluid Whirl in Vertical Machines," Trans. ASME, Vol. 78, No. 1, Jan. 1956, pp. 13–19.

8 Lund, J. W., "The Stability of an Elastic Rotor in Journal Bearings With Flexible, Damped Supports," *Journal of Applied Mechanics*, Vol. 32, No. 4, Trans. ASME, Vol. 87, Series E, Dec. 1965, pp. 911–920.

9 Hagg, A. C., "The Influence of Oil-Film Journal Bearings on the Stability of Rotating Machines," *Journal of Applied Mechanics*, Vol. 13, No. 3, Trans. ASME, Vol. 68, Sept. 1946, pp. A-211–A-220.

10 Poritsky, H., "Contribution to the Theory of Oil Whip," Trans. ASME, Vol. 75, No. 6, Aug. 1953, pp. 1153–1161.

11 Pinkus, O., and Sternlicht, B., *Theory of Hydrodynamic Lubrication*, McGraw-Hill, New York, 1961.

12 Pinkus, O., "Note on Oil Whip," *Journal of Applied Mechanics*, Vol. 20, Trans. ASME, Vol. 75, No. 3, Sept. 1953, pp. 450–451.

13 Pinkus, O., "Experimental Investigation of Resonant Whip," Trans. ASME, Vol. 78, No. 5, July 1956, pp. 975–983.

14 Hagg, A. C. and Warner, P. C., "Oil Whip of Flexible Rotors," Trans. ASME, Vol. 75, No. 7, Oct. 1953, pp. 1339–1344.

15 Newkirk, B. L., "Varieties of Shaft Disturbances Due to Fluid Films in Journal Bearings," Trans. ASME, Vol. 78, p. 985, 1956.

16 Sternlicht, B., "Investigation of Translatory Fluid Whirl of Vertical Shafts," PhD thesis, Columbia University, 1954.

17 Lund, J. W., discussion of paper by P. C. Warner and R. J. Thoman, "Effect of 150-Degree Partial Bearing on Rotor-Unbalance Vibration," *Journal of Basic Engineering*, Trans. ASME, Series D, Vol. 86, No. 2, June 1964, pp. 337–347.

18 Warner, P. C. and Thoman, R. J., "The Effect of the 150-Degree Partial Bearing on Rotor-Unbalance Vibration," *Journal of Basic Engineering*, Trans. ASME, Series D, Vol. 86, No. 2, June 1964, pp. 337–347.

19 Feldman, S., "Dynamic Balancing for Noise Reduction," BuShips Report No. 371, Vol. 24, Apr. 1955.

20 Federn, K., "Unwuchttoheranzen Rotierender Korper," Werkstatt and Bltrieb, Vol. 86, No. 5, May 1953.

21 Arwas, E. G., Sternlicht, B., and Wernick, R. J., "Analysis of Plain Cylindrical Journal Bearings in Turbulent Regime," *Journal of Basic Engineering*, Trans. ASME, Series D, Vol. 86, No. 2, June 1964, pp. 387–395.

22 Orcutt, F. K., "Steady-State and Dynamic Properties of the Tilting-Pad Journal Bearing in Laminar and Superlaminar Flow Regimes," MTI-65TR32.

23 Orcutt, F. K., and Ng, C. W., "Steady-State and Dynamic Properties of a Floating-Ring Bearing in Laminar and Superlaminar Flow Regime," NASA, AFSS—A Third Topical Report.

24 Lund, J. W., and Orcutt, F. K., "Calculations and Experiments on the Unbalance Response of a Flexible Rotor," Journal of Engineering for Industry, Trans. ASME, Series B, Vol. 89, No. 4, Nov. 1967, pp. 785–796.

DISCUSSION

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Establishing shop tolerances for residual unbalance in rotors in terms of in-oz per weight of rotor goes way back to the early 1920's, at least. As I recall, we at the Westinghouse South Philadelphia Turbine Plant had set up a rule for balancing in the dynamic balancing machine down to within one in-oz per 1000 lb of rotor, for the sum at the two ends. For a big 1800 rpm rotor, say 50 tons, this represented three or four oz at the balance plug radius.

It is interesting to note that Graphs (1) and (2) and the Navy Graph (3) in the authors' Fig. 11 chart all agree fairly well with this early practice for rotors at 1800 to 3600 rpm.

For the final shop balancing at rated speed and overspeed, in the heater box more nearly simulating service conditions, vibration amplitudes at the bearings became criteria for tolerance, with frequency a factor.

In the final analysis, vibration amplitudes under operating conditions in the customer's plant become the yardstick, rather than in-oz, or microin of c.g. displacement. For this purpose, vibration tolerance charts in terms of amplitudes versus frequency have appeared from time to time. One early Machinery Vibration Tolerance Chart that has been widely used as a guide was prepared by this writer and published as Fig. 4 in the article "Vibration Tolerance," in the November 1939 issue of *Power Plant Engineering*. The latest reproduction of this chart as slightly revised some time ago appears in ASME Paper No. 67–PEM-14, "Vibration Tolerances in Industry," presented at the 1967 Plant Engineering and Maintenance Conference in Detroit by R. L. Baxter and D. L. Bernhard.

The authors are to be commended for their thorough analysis of critical speed and whirl phenomena, and their work on instability.

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Authors' Closure

The authors wish to thank Mr. Rathbone for his comments. We fully agree that the final criterion for acceptability of equipment is the performance under operating conditions in the customer's plant. Vibration amplitudes must be acceptable and the vibration tolerance charts suggested by Mr. Rathbone can be a valuable guide. The importance of degree of unbalance cannot, however, be minimized or overlooked. The rotor response analysis described by the authors permits evaluation of the sensitivity of the system to location and amount of unbalance. By this is meant that most machines over a period of time will exhibit changes in balance by virtue of creep, material relaxation, erosion, corrosion, deposits, as well as other causes. The degree of balance initially may permit acceptance under the conditions of operation; however, often very short periods of time can demonstrate that the adequacy is very short lived.

A very important implication of Mr. Rathbone's comments should be emphasized. That is, a single guide specification such as the degree of unbalance in ounce inches or micro inches of displacement of a rotor, is in itself incomplete. It is only when this factor can be considered by such tools as a rotor dynamic analysis that the sensitivity of the rotor system and its long time operating behavior can be deduced. The reference levels given are intended primarily as a guide with the implication that values well in excess of those recommended will, in all likelihood, result in higher than desired equipment vibration.