COMPULSATOR ROTORDYNAMICS AND SUSPENSION DESIGN

B. T. Murphy, S. M. Manifold and J. R. Kitzmiller
Center For Electromechanics, The University of Texas at Austin, Austin, Texas

***ABSTRACT*—High speed compulsator rotors utilizing high strength composite bandings pose unique problems from a rotordynamic standpoint. This article describes the basic design approach for rotordynamics used at The University of Texas at Austin Center for Electromechanics at (UT-CEM). As an example, the CCEML compulsator rotordynamic design is presented. The key considerations are seen to be; 1) mass and stiffness properties of the fully assembled rotor, 2) selection of rotor support bearings, 3) bearing supporting structure, 4) proper placement of rotor critical speeds, 5) adequate attenuation of rotor response at all speeds, and 6) bearing load capacity to react large discharge forces. Due to large mechanical and thermal shocks which occur during discharge, a primary design goal is to maximize tolerance to rotating imbalance. Another primary design goal is avoidance of destructive whirling instabilities which can occur with high speed rotors possessing large amounts of damping within the rotating assembly.**

INTRODUCTION

Air core compensated pulsed alternators (compulsator or CPA) are practically ideal power supplies for mobile electromagnetic launchers (EML). An air core CPA is a light weight, high speed, high energy rotating machine capable of satisfying gigawatt-plus power needs of modern electric guns. In terms of both power per pound and power per unit volume, an air-core CPA has the best potential for providing a practical field transportable power system.

This paper presents the basic design approach used at UT-CEM for optimizing the rotordynamic aspects of air core CPAs. As an example to illustrate the concepts and results achieved, the CCEML CPA will be discussed in detail. Results of the rotordynamic analysis and mechanical spin testing will be presented. To date, CCEML CPA has been mechanically spin tested to 70% of maximum design speed, including numerous traversals of a rotor critical speed.

Manuscript received April 8, 1996.

B.T. Murphy, e-mail b.murphy@mail.utexas.edu, phone number (512) 471-4496, fax (512) 471-0781; S.M. Manifold, e-mail ezab065@hermes.chpc.utexas.edu, phone number (512) 471-4496, fax (512) 471-0781; J.R. Kitzmiller, phone number (512) 471-4496, fax (512) 471-0781.

This work was supported by CCEML under contract no. DAAA21-92-C-0060 and subcontract no. F55305-62.

ROTORDYNAMIC DESIGN OF AIR CORE COMPULSATORS

The overall approach to rotordynamic design for many rotating machine layouts is to:

1. Create a rotordynamic model of the given shaft assembly, and use it to
2. compute the rotor critical speeds as a function of support stiffness. The results of which are used to
3. decide on placement of rotor critical speeds with respect to the operating speed range. This is done with due consideration to what can be practically achieved in the way of bearing stiffness so that one can
4. select a bearing technology/design for radial and axial rotor support. To achieve the desired properties identified in step 3, it may be required to
5. design bearing supports, so that the rotor "sees" the desired stiffness and damping properties between itself and the stator. In conjunction with this effort,
6. compute the response of the rotor to various amounts and distributions of residual mass imbalance. This helps identify how well the rotor should be balanced initially, and how much imbalance the system should withstand.
7. If warranted, perform a damped eigenvalue analysis to check the rotordynamic stability of the system with regard to any significant destabilizing mechanisms.

The above series of steps are often performed in an iterative fashion. The overall goal is to arrive at a practical rotor/bearing design which is robust and reliable.

In the case of air core CPAs, emphasis is placed on maximizing tolerance to residual mass imbalance. Experience with other composite rotors indicates that air core CPA rotors can be successfully precision balanced. However, because of their composite construction together with large transient forces associated with electrical discharge, the ability to retain precision balance during use has not yet been fully demonstrated. Thermal transients which sweep through the rotor during and after discharge(s) are also causes of concern regarding balance retention.

ROTOR MODEL

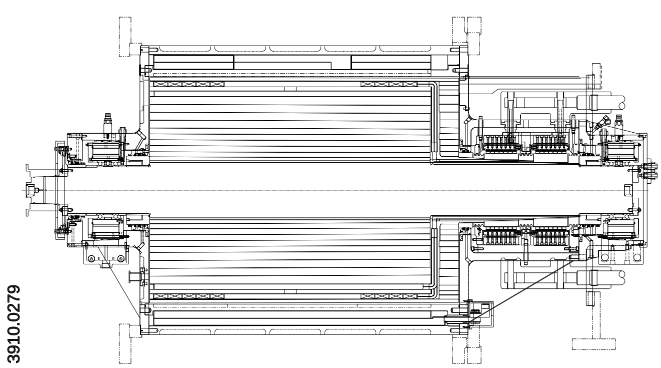
Fig. 1 shows the rotor top assembly drawing for the UT-CCEML CPA, and the corresponding rotordynamic model. The rotordynamic model was constructed using 3 node isoparametric cylindrical beam elements of constant cross section [1]. Two dimensional beam element models like this are entirely adequate for rotordynamic analysis. The beam elements fully model the mass and elastic properties of the rotor assembly [2]. The model was analyzed with a special purpose, rotordynamic analysis code, TXROTOR, developed at UT-CEM expressly for performing all facets of rotordynamic analysis. The program includes the effects of gyroscopics and other speed dependent rotor/bearing properties. TXROTOR computes both damped eigenvalues and response to harmonic and transient excitations. Thus, it can be used to compute rotor critical speeds, dynamic stability, as well as loads and deflections due to residual mass imbalance. These various analysis are done as functions of rotational speed. Also, TXROTOR is not limited to just rotors: machine cases and bearing support structures can also be modeled along with the rotor.

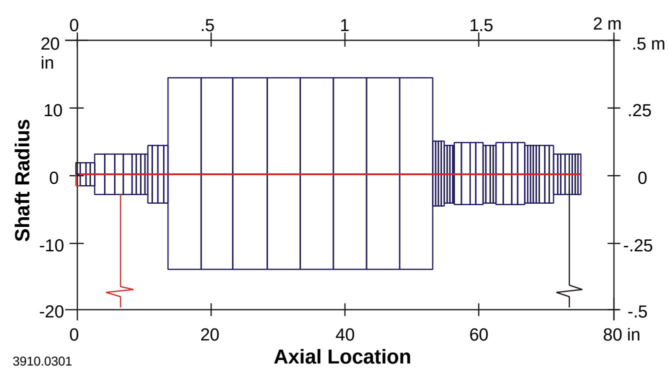
From the rotor model shown in Fig. 1, an undamped critical speed map vs. support stiffness was computed, and is shown in Fig. 2. Fig. 2 shows that the CCEML rotor must run above the first critical speed, but could run between the first and second with a net support stiffness of 2.0×10^6 lb/in. (3.5×10^8 N/m). A net bearing stiffness value this high would be extremely difficult to achieve in practice. Also, “hard mounting” a large high speed rotor in such a fashion would lead to a very small balance tolerance requirement for reliable operation.

Referring again to Fig. 2 an attractive alternative is to “soft mount” the rotor with a support stiffness of around 2×10^5 lb/in. (3.5×10^7 N/m). Since the rotor must then run through two rigid rotor critical speeds, sufficient damping in the supports would be required to limit resonant response at the criticals. The first bending mode of the rotor is seen to always be well above the maximum operating speed. This aspect makes the “soft mount” approach even more attractive. This is the design approach adopted for the CCEML rotor.

BEARING SELECTION

Since the CCEML machine is targeted for use in mobile environments, the bearings must be robust and must adequately support the rotor while it is rotating and at rest. Rolling element bearings are a logical first choice. The actual bearing selection process is driven by the high speed of the rotor, and the potentially high transient forces which can occur during electrical discharge. The high transient forces call for large bearings, but maximum allowable speed decreases as size increases for rolling element bearings. A useful speed-





CCEML Subscale Rotor
Rotor weight = 2,100 lb
Speed range = 8,000 to 12,000 rpm
Rotor energy = 40 MJ, max

Fig. 1. CCEML CPA rotor and the corresponding rotordynamic finite element model

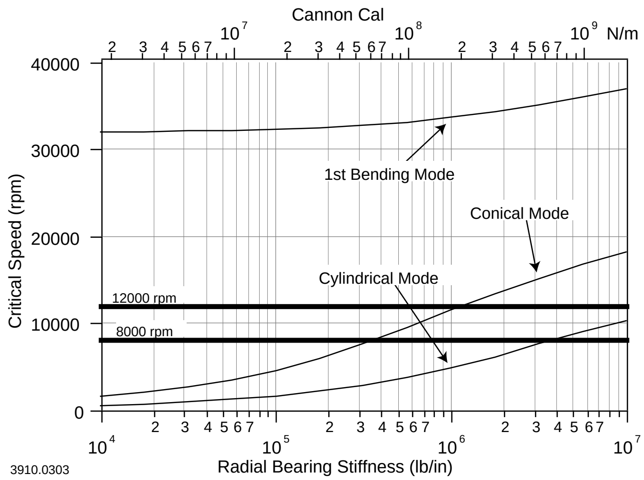


Fig. 2. Undamped synchronous critical speeds as a function of support stiffness, for the CCEML compulsator rotor

size parameter for bearings is DN, where D = bore size in mm and N = rotor speed in rpm. For many years the practical upper limit on DN for reliable operation of rolling element bearings was about 1×10^6. The past 5 to 10 years has seen many advances in manufacturing tolerances, elastohydrodynamic lubrication, lightweight ceramic bearing parts, and the design and materials used for cages. The practical DN limit is now well beyond 2×10^6 for angular contact ball bearings. The selected bearings for the CCEML rotor are:

Manufacturer:	Split Ball Bearing
Type:	15° angular contact, DB duplex
Model:	5HAZ 1830-PT
Size:	150 mm (5.91 in.) bore
Lubrication:	continuous oil stream, 1 gpm/pair (3.785 lpm)
Races:	M50 bearing steel
Balls:	SiN (ceramic, 7/16 in., 37/bearing)
Cage:	silicon iron bronze
DN:	1.8 x10^6 @ 12,000 rpm

The ultimate static load capacity of these bearings has been calculated to be on the order of 40,000 lb (1.78×10^5 N) per bearing for either axial or radial loads. Transient radial loads would be shared by all four bearings supporting the shaft. Axial loads would be reacted solely by one bearing. Transient loads produced during discharge cannot be predicted accurately. Actual loads are due to slight imbalance of large distributed electromagnetic forces acting throughout the conductive coils on the rotors (armature for CCEML). Rough estimates of the loads are well within the calculated radial bearing capacity, and are comparable to the axial bearing capacity. The brief transient nature of the discharge loads, together with the compliance of the rotor, bearings, and end plates should reduce the actual peak bearing reaction forces.

Bearing Supports

The radial stiffness of the selected ball bearings has been calculated to be around 2.0 x 10^6 lb/in. (3.5×10^8 N/m) per bearing. This stiffness prediction is based on a nonlinear bearing analysis code which has been experimentally verified specifically for use with high speed angular contact ball bearings [3]. To reduce the support stiffness to the 2×10^5 lb/in. (3.5×10^7 N/m) range, a compliant bearing structure is required. Support damping is also needed for resonant attenuation during critical speed traversal. This dual requirement is satisfied by a squeeze film damper (SFD) [4]. SFDs are widely employed on aircraft gas turbine engines. Through the use of SFDs, gas turbine engines are able to safely operate on critical speeds, and can tolerate large imbalances resulting from loss of a turbine blade. The CCEML design does not allow the axial space typically required by standard SFD designs. So a special compact design (Fig. 3) was obtained from KMC, Inc. of West Greenwich, Rhode Island [5]. The damper is primarily a one part structure, which is wire electro-discharged machined from a single piece of titanium.

The stiffness and damping properties of the SFDs can be independently set within wide ranges of values. Referring to Fig. 3, the radial stiffness value is produced by bending of the eight L-shaped beams shown. The radial damping value is produced by squeezing oil in the four preset clearance spaces.

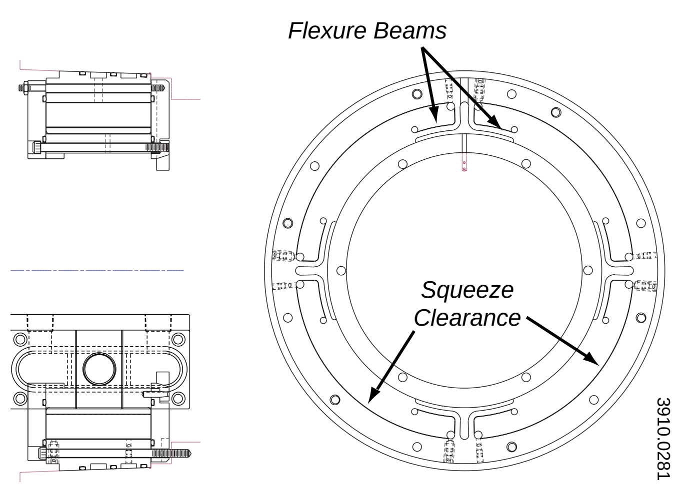


Fig. 3. Squeeze film damper used for bearing support

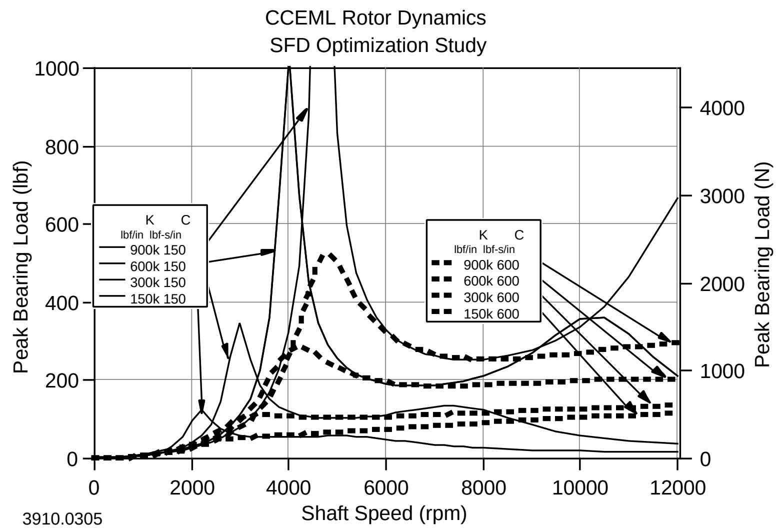


Fig. 4. Sample imbalance response plots used for SFD optimization.

Response to Imbalance

An important aspect of the rotordynamic design process when SFDs are involved, is determining optimum stiffness and damping properties for the SFDs. The purpose behind using SFDs is to achieve an overall reduction of rotor response to residual mass imbalance. SFDs are then optimized through a series of imbalance response studies with the rotordynamic model. Fig. 4 shows some sample results from the studies performed on the CCEML CPA rotor. This figure shows predicted bearing loads vs. running speed for a particular mass imbalance distribution. Results for ranges of different SFD properties are shown.

In the optimization study it is necessary to consider the effect of different imbalance distributions. Also, in addition to bearing loads, deflections of the rotor, bearings and dampers must be taken into account [6]. Fig. 4 illustrates the general trend that lower stiffness results in lower bearing loads. Making the dampers too soft, however, leads to excessively large rotor displacements. Higher damping helps reduce bearing loads, but only at the critical speeds. At high speed excessive damping can lead to increased bearing loads.

From results like those shown in the figure, "optimum" SFD radial stiffness and damping values of 1.5×10^5 lb/in. (2.6×10^7 N/m) and 150 lb-s/in. (2.6×10^4 N-s/m), respectively, were selected.

Rotor Stability

Rotordynamic instability is always a concern with high speed rotating machinery which operates above any critical speeds [7], [8]. In an overall sense, rotordynamic instabilities occur when destabilizing forces exceed the damping forces. In the case of high speed pumps, compressors and turbines, the destabilizing forces often come from fluid/solid interaction at impellers and blades, whereas the damping forces generally act at the bearings [9]. These types of instabilities, when they occur, are often termed an "aerodynamic instability" in reference to their source. These types of instabilities do occur in practice, and can be very damaging to the machine, and enormously expensive to remedy [10], [7].

Certain classes of hydrodynamic bearings can also be sources of rotordynamic instabilities [11]. These instabilities are often called "bearing whirl instabilities" or "oil whip". A classic example is a cylindrical sleeve journal bearing supporting a rotor at a speed above about twice the first rotor critical speed. Such conditions nearly always produce a rotordynamic instability. These instabilities are more common than aerodynamic instabilities, but can usually be corrected by a change to the bearing without effecting the remainder of the machine.

In the case of a composite CPA rotor, the main destabilizing forces of concern are due to dissipative effects within the rotating assembly. This type of instability, when it occurs, is called an "internal friction" instability [12]. Damping mechanisms within a rotating assembly are always benign in subcritical machines. That is, they have no significant effect on the dynamic behavior of the machine. In supercritical rotors, however, rotor-borne damping actually becomes a source of instability. In conventional rotating machines the typical sources of dissipation are splines and couplings, loose or marginally tight press fits, and interfaces in axially built up rotors. If the rotor contains any viscoelastic material like rubber, that also can be a source of destabilizing rotor damping. In industrial machinery, internal friction instabilities are extremely rare as they are easily avoided by adhering to proper methods of design and fabrication.

Since an air core CPA has no impellers or turbine stages, and for purposes of this discussion are being supported by ball bearings, only an internal friction instability is considered possible. On a composite CPA rotor, significant dissipative effects can possibly occur due to any or all of the following:

1. Material damping within the composite/epoxy matrix.
2. Movements of the potted conducters with respect to the rotor.
3. Slippage or creep at the cylindrical interfaces between layers of the built up rotor.

The most effective way to avoid an internal hysteresis instability is to not operate the machine at a speed above any natural mode which is appreciably damped by rotor dissipative mechanisms. A free-free rap test on the rotor can provide a quantification of rotor damping on a mode by mode basis. Such "ring down" measurements were conducted on the CCEML rotor. Results showed that rotor damping was comparable to solid metal rotors. This, together with the fact that the two critical speeds that the CCEML rotor runs above are rigid body modes with negligible rotor bending, eliminates concern that an internal friction instability might occur.

TEST RESULTS FOR CPA ROTOR RESPONSE

When the SFD dampers were delivered to UT-CEM, they were tested for their stiffness and damping values. They were found to be overly stiff at approximately 1.2×10^6 lb/in. (2.1×10^8 N/m). The damping was also measured to be higher than optimum, approximately 1,200 lb-s/in. (2.1×10^5 N-s/m). The high stiffness is seen to place the rigid rotor critical speeds in or near the operating speed ranges. The high damping, however, was predicted to be adequate to permit each machine to actually run on the critical speeds. In fact, the damping is close to being sufficient to suppress any noticeable resonance. This is not as good as the design optimum, but was acceptable as opposed to performing extensive modifications to the existing damper hardware.

For the CCEML rotor, the first rotor critical speed with the current SFDs is predicted to be near 5500 rpm. The level of damping produced be the SFD should permit a well damped, but noticeable, response while traversing the first critical speed. The second critical speed is predicted to be close to 12,000 rpm. The response at this critical speed should be heavily damped. To date, mechanical spin up tests have been performed to a planned maximum of 8500 rpm. Speed will be increased when the CPA's electrical systems are fully installed. The plot in Fig. 5 shows rotor deflection data measured adjacent to the thrust end bearing pair.

Other features about the composite CPA rotors were noted during mechanical testing.

1. Measured rotor response was very consistent on repeat runs of the same configuration. That is, no shifting or creep occurred on either rotor at any time during their respective test programs.
2. Although a very good state of balance had been achieved on a low speed balance machine prior to assembly within the stator, in-situ trim balancing was able to significantly reduce residual rotor imbalance
3. The CCEML rotor is a working example that a com-- posite rotor can traverse a rotor critical speed with no detrimental effects.

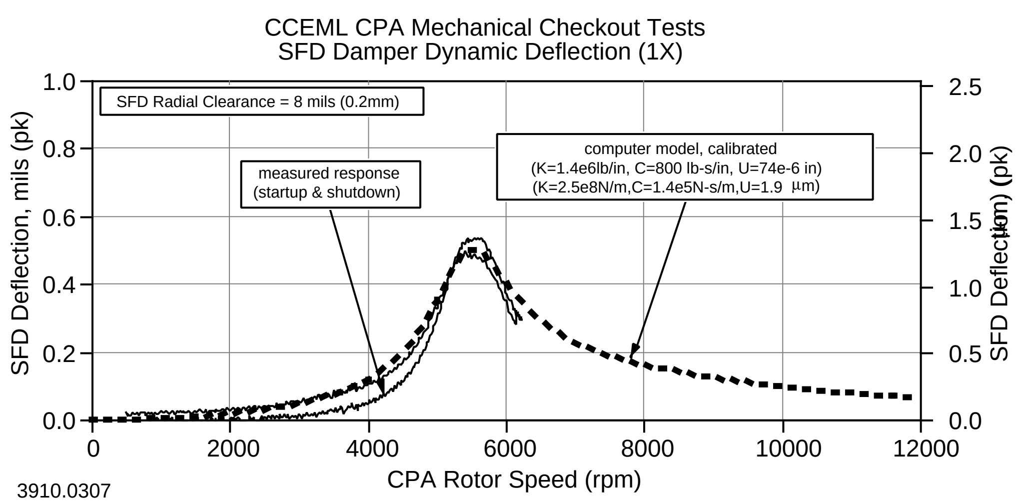


Fig. 5. Measure test data for the CCEML CPA rotor

CONCLUSIONS

State of the art rotordynamic analysis methods have been successfully applied to the design of air core compulsators. Advanced finite element techniques were used to model the rotating assembly, and a special purpose computer program was used to calculate rotordynamic critical speeds and imbalance response. The rotordynamic model was used to help select a bearing technology best suited to the application (angular contact ball bearings in conjunction with squeeze film dampers). The rotordynamic model was also used to calculate optimum properties for the squeeze film dampers to minimize the machine's sensitivity to residual imbalance throughout its speed range. The rotordynamic model was again used to assess the effect of using squeeze film dampers which had been measured to possess stiffness and damping in excess of the optimum design values.

Mechanical spin up tests on a completed CCEML CPA rotor assembly verified that the composite rotor could be precision balanced, and that it would retain that level of balance from run to run. The CCEML rotor has also been run multiple times through a rotor critical speed, showing no effect on vibration performance. The SFD bearing supports have performed as required by limiting measured vibration amplitudes to benign levels, thus permitting safe and repeatable operation.

ACKNOWLEDGMENTS

Funding for this work was provided by the U.S. Army and Marine Corps, under contract Cannon Caliber Electromagnetic Launcher, contract #DAAA21-92-C-0060 (subcontract #F55305-62).

REFERENCES

[1] Bathe, K. J., *Finite Element Procedures in Engineering Analysis*, Prentice Hall, Englewood Cliffs, New Jersey, 1982.

[2] Nelson, H. D., and McVaugh, J. M., "The dynamics of rotor bearing systems using finite elements," *Journal of Engineering for Industry,* pp. 594-600, May 1976.

[3] Murphy, B. T., and Butner, M. F., "Influence of mounting compliance on the dynamic stiffness of ball bearings: analysis and test results," *ASME Journal of Vibration and Acoustics.*

[4] San Andres, L. A., and Vance, J. M., "Experimental measurements of squeeze film bearing force coefficients for circular centered orbits," ASLE Paper No. 86TC402, ASME/ASLE Tribology Conference, Pittsburgh, October 1986.

[5] Zeidan, F. Y., "Application of squeeze film dampers," *Turbomachinery International*, Vol. 36, No. 6, 1995, pp. 50-53.

[6] Barret, L. E., Gunter, E. J., and Allaire, P. E., "Optimum bearing and support damping for unbalance response and stability of rotating machinery," *Journal of Engineering for Power*, pp. 89-94, January 1978.

[7] Ehrich, F. F., and Childs, D. W., "Self-excited vibration in high-performance turbomachinery," *Mechanical Engineering*, May 1984, pp. 66-79.

[8] Lund, J. W., "Stability and damped critical speeds of a flexible rotor in fluid film bearings," *Journal of Engineering for Industry*, Vol. 96, No. 2, pp 509-517, 1974.

[9] Alford, J. S., "Protecting turbomachinery from self-excited rotor whirl," *Journal of Engineering for Power,* June 1965, pp. 333-344.

[10] Childs, D. .W., "The space shuttle main engine high-pressure fuel turbopump rotordynamic instability problem," *Journal of Engineering for Power*, pp. 48-57, January 1978.

[11] Newkirk, B. L., and Taylor, H. D., "Shaft whipping due to oil action in journal bearings," *General Electric Review*, Vol 28, pp 559-568, 1925.

[12] Ehrich, F. F., "Shaft whirl induced by rotor internal damping," *Journal of Applied Mechanics*, June 1964, pp. 279-282.