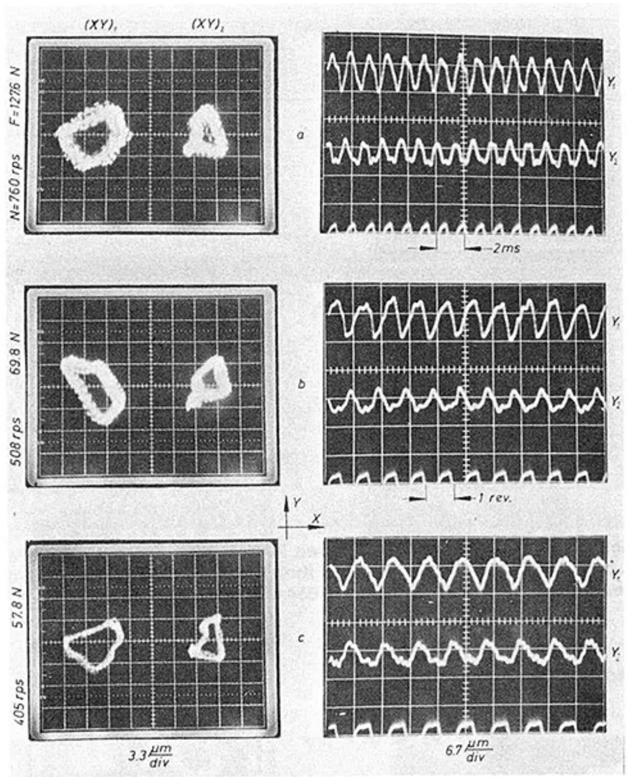

Fig. 15 Trajectories and time-base records of motion in $(XY)_{1,2}$ and $(Y_1 Y_2)$ planes with asymmetric unbalance U_1 = 19.3 μm N and U_2 = 23.2 μm. N – journal foil bearing Design B (octagonally bent backing and 2-1/4 inner turns, one steel foil per bearing, $L \times D \times t_f$ = 20 × 30 × 0.054 mm. journal diameter = 29.87 mm, foil-cartridge bore = 30.61 mm). thrust loads and speeds as indicated

10 Trippett, R. J., Oh, K. P., and Rohde, S. M., "Theoretical and Experimental Load-Deflection Studies of a Multileaf Journal Bearing," ASME Publication 100118, 1979.

11 Licht, L., "Resilient Foil Journal Bearing," U. S. Patent No. 4,133,585, January 9, 1979.

12 Licht, L., "Foil Bearings for Axial and Radial Support of High Speed Rotors - Design, Development and Determination of Operating Characteristics," NASA CR-2940, Jan. 1978.

13 Licht, L., Anderson, W. J., and Doroff, S. W., "Design and Performance of Compliant Thrust Bearings with Spiral-Groove Membranes and Resilient Supports," To be presented at the ASME-ASLE International Lubrication Conference, San Francisco, California, August 18–21, 1980, ASME Paper No. 80-C2/Lub-36.

14 Tondl, A., and Licht, L., "Nonlinear Resonances of Rotors and their Identification," *Acta Technica ČSAV* (Czechoslovak Academy of Science), No. 1, 1976, p. 74.

15 Licht, L., "Resilient Foil Thrust Bearings," U. S. Patent No. 4, 116,503, September 26, 1978.

DISCUSSION

V. Castelli[2]

The authors present an informative and thorough study of two interesting types of foil bearings.

Due to the present lack of reliable theoretical modeling of these bearing configurations and due to their attractive manufacturing and performance characteristics, experimental information is certainly welcome and useful.

This writer would be grateful if the authors answered a few questions.

1. Design B uses a foil, the outer loop of which is prefolded into a polygon. Are the folds annealed? If they are not, it would seem difficult to avoid erratic results due to stress relaxation.
2. What criteria did the authors use to choose the thickness, length, and number of turns of the foils, particularly in Design B?
3. Both the damping and stiffness of the foils are affected by friction between foil layers. How reproducible is the behavior of these bearings from run to run and from bearing to bearing?
4. Can most of the differences in behavior between Design A and Design B (such as fewer subharmonic and superharmonic vibrations) be ascribable to the softer elastic suspension of Design B?
5. Were stiffnesses measured?
6. Some of the amplitude versus speed oscilloscope traces are not symmetrical. Is that a sign of some electronic trouble?

W. D. Waldron[3]

The bearings described in Dr. Licht's papers are both interesting and innovative. The coiled journal bearing with the preformed backing is the first published conscious attempt to combine a highly compliant integral substructure with a multi-layered foil to provide for Coulomb damping. The thrust bearing support systems described integrate well, at

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least in principle, the superior load carrying capability of a spiral grooved geometry with the "forgiveness" of a compliant mount.

The journal bearing performance was impressive in that when half frequency whirl set in at almost exactly twice the first critical, it was totally contained within an amplitude envelope that was essentially independent of speed. This containment of fractional frequency amplitudes attests to the superior damping characteristics inherent to the configuration.

The fact that the amplitude of the synchronous orbit was essentially independent of speed above "inversion" is not surprising as this amplitude was essentially equal to the mass unbalance. Although the author speaks of "excessive unbalance", the unbalance force at the maximum test speed was only about 10 psi; not an excessive load by today's gas bearing standards. It would be interesting to know how the bearing's fractional frequency whirl characteristics are affected by higher radial loads, either static or dynamic.

With respect to the thrust bearing, the maximum applied load that the author reported was equivalent to approximately 5 psi. Here too, it would be interesting to know how it performs in the 15 to 20 psi range where competitive rigid types tend to get into trouble from thermal and misalignment effects.

Authors' Closure

The authors are rewarded by the interest of Dr. Castelli and Mr. Waldron in this developmental study.

In reply to the purposive questions posed by Dr. Castelli:

1. The foils were full-hard (as rolled) and were not stress-relieved either before, or after cold-forming of the quasi-octagonal support. A useful method of "fitting" a foil element was to start with a more acute angle of bend (e.g. 120 deg, rather than 135 deg), insert the coil into the retainer, and force into the bore several taper-nosed arbors of diameters progressively larger than the journal size, thereby decreasing the initial curvature at the vertices by "reverse deformation." The response of the rotor supported in this type of foil bearings is *not* overly sensitive to changes of preload and working clearance of magnitudes corresponding to dimensional deviations caused by stress relaxation.

In the course of a later investigation, hard and hard-coated foils (Inconel 718 and X-750, Ni-Be and AM-350) with similar, cold-formed spring supports were subjected to over 18,000 (and later to over 30,000) start-stop cycles. Scans of the response up to 50,000 rpm, obtained prior to, during and after the cycling tests, indicated only minor differences. Scans obtained following a 120 hour run at a foil-bearing temperature of 330°C (625°F) at the turbine end were substantially identical with those recorded prior to the "hot" test. Differences in resonant speeds and trajectory size observed between runs at room and at elevated temperatures were reduced by replacing the foil retainer with one having a higher coefficient of expansion, without changing the foil element (Inconel 718).

It is recognized that careful but *properly weighted* consideration must be given to the choice of foil materials, particularly for high speed and temperature applications. Indeed, as in many similar applications, the problems of mechanically and thermally induced fatigue, of tendencies to crack propagation, corrosion and fretting at points of stress concentration, of decrease of yield strength, hardness and elastic modulus at elevated temperature, should be minimized. Foils are thin, and distortions as well as loss of desirable properties imparted in rolling should be considered with reference to heat treatment *after* cold forming. Fortunately, the choice of suitable materials is quite large and residual stresses can be dealt with. More difficult is the problem of wear and of effective surface conditioning.

2. The simplicity of fabrication of these foil bearings obscures the geometrical intricacies and complexities of friction-influenced displacements and vibrations. Realistic mathematical modeling of such bearings is a Sisyphean undertaking, even for a very proficient analyst with considerable hardware experience and astute powers of observation in the laboratory. Not only is the number of parameters large, but their variability and functional dependence (e.g. friction) are as difficult to define as the relevant boundary conditions.

Only very simple and approximate calculations were made. For a given bearing size, the thickness, width and properties of the foil were considered in regard to static stiffness and deflection, preload and starting torque, permanent deformation, wear compatibility and many other bearing characteristics. The coiled structure contributes to frictional damping and distributes the contact forces in the vicinity of rounded vertices over a wider bearing area. The number of turns, generally two to six, depends on bearing size, foil thickness and spacing of rounded vertices.

The addition of the polygonal spring-support to the coil was intentional, of course, since it was anticipated that this geometry would be very conducive to frictional damping (due to relative motion between the spreading and contracting "thighs" of the deformed polygon and the adjacent surfaces of the foil stack, particularly in the vicinity of rounded vertices, Fig. 5). No attempt was made to control the frictional properties of relevant surfaces, or to describe the local variation of friction with frequency and amplitude of motion.

Both experience and intuition that comes with it were helpful in zeroing in on various parameters, and both the test rig and the bearings were designed to facilitate fast "mechanical breadboarding". A foil element could be preformed within 15 minutes and replaced within 30 minutes in the test rig between runs.

3. Experiments conducted with both hexagonal and octagonal supports indicated no appreciable change in the basic character of the response. Replacement of one set of foil elements with another set of the same kind produced only very minor changes of the response. The variability between runs with the same foil elements was quite insignificant.

4. The suppression of resonant amplitudes and of rotor instabilities in foil bearings of Design B (octagonal spring-support) can undoubtedly be ascribed to the fact that compliance facilitates motion, in the course of which appreciable damping forces are generated with the aid of a geometry conducive to the frictional dissipation of energy. A secondary contribution to damping is due to the multi-clearance squeeze films (which may be dominant in a liquid-lubricated foil bearing). Compliance, in the absence of dissipation, is not conducive to stability.

5. Both stiffness and breakaway friction-torque were measured, but rather perfunctorily. We regret that we were unable to locate the pertinent records in a large volume of raw, unreported data. Stiffness increases considerably at large displacements of the journal from the concentric position. A hystertic effect (due to friction) could be observed in the course of static determinations of load-deflection characteristics.

6. The scans of amplitude components with frequency were recorded in the ac-mode and the probe outputs do not reflect changes of the *mean* journal-positions with speed. The responses, however, contain both even and odd harmonics and are aperiodic in several intervals, as depicted in numerous photographs of trajectories and wave forms (also recorded in the ac-mode). There is no reason for the scans to be sym-

metric and the asymmetries are not a consequence of instrument malfunction.

In regard to comments made by Mr. Waldron:

1. We concur that the journal bearings with the polygonal spring supports are endowed with very effective damping, so that resonant amplitudes are reduced, instabilities suppressed, and the response quite flat and typical of highly damped systems.

2. In the journal bearings with coiled, plain-foil elements (Design A) the damping is still sufficient to contain the resonant and unstable trajectories within very acceptable limits. Regions of instability are traversable and envelopes of trajectories remain fairly constant, or even diminish with increasing speed. In this highly nonlinear, four-degrees-of-freedom, asymmetric system excited by a large pitching unbalance and influenced by gyroscopic forces, the onset-speed of whirl cannot be so simply related to twice the "first critical" speed. These journal foil-bearings generally operate at average clearances that are larger, and minimum gaps that are smaller than in rigid bearings. It is doubtful if an increase in radial load to enhance stability would be advisable when limit-trajectories of unstable motion are so well contained. For the case at hand, it would appear that the unbalance has a destabilizing influence, Fig. 9.

3. The added unbalance *was excessive* and its asymmetry most disadvantageous. Expanders, turbocharges, small gas turbines, centrifuges and other high-speed machines have maxima of total unbalance specified in the approximate range 15 to 750 μin.oz/lb. An average value of the order of 350 μin.oz/lb may be taken as typical, while a more stringent U. S. Navy specification calls for a maximum of 230 μin.oz/lb at 45,000 rpm. The present unbalance of 1420 μin.oz/lb was four to six times as large and this *is excessive*, particularly in view of recent advances in balancing techniques.

4. The following comments apply to rigid progenitors of the compliant, spiral-groove thrust bearings. An optimized and otherwise ideal, perfectly aligned, flat spiral-groove bearing, operating at 45,000 rpm, would support a unit load of 20 psi at a clearance of only ~ 200 μ in. (5 μm). How this bearing would fare in the presence of a "non-excessive" pitching unbalance of 1420 μin.oz/lb, even if the massive plate remained flat and were flexibly supported, or gimbal mounted, is *quite predictable*! It is unrealistic to expect that any air bearing in a turbomachine would reliably support a unit load of 20 psi, or even 15 psi, especially if a 30 to 50% margin of safety were required. On the other hand, 10 psi is a reasonable, realistic and realizable expectation in regard to the compliant thrust bearings discussed in the companion paper [7] and tested concurrently with the present foil journal-bearings.

Our objective, rather than to better load-lifting records attainable on paper and sometimes in well-controlled laboratory tests, was to design a compliant thrust bearing that would be safer and less likely to fail in a real machine than the existing types of both rigid-surface and foil thrust-bearings of equal dimensions, when operated under the same conditions.

In conclusion, we wish to thank Dr. Castelli and Mr. Waldron for their stimulating discussions and for their interest in our papers.