Mr. B. Lawson, who participated fully in all phases of the experiments. Mr. G. Holz was responsible for the illustrations.

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DISCUSSION

W. J. Anderson[3]

The authors results are quite interesting, and may prove to be of considerable significance in the design of a variety of small, very high speed turbine driven machines. Air cycle machines and cryoexpanders are just two classes of machines in which the ability to operate at extremely high rotative speeds could be advantageous from the standpoints of both space utilization and economics. Compactness of size has obvious physical advantages, and, other factors being equal, it also has an economic impact through savings of materials. The capability of operating at very high speeds also implies the ability to utilize fully the potential capabilities of small turbines and compressors. Operation at tip speeds as high as 2000 ft per sec (610 m/sec) is sometimes desired. At 200,000 rpm components 2 in. (5.08 cm) in diameter could be used. The work output of such a machine would be extremely high for its size.

It is interesting to compare the authors' results with the capabilities of other types of bearing support systems. For rolling element bearings, operation of 0.5 in. (12.7 cm) bore bearings at 200,000 rpm represents a DN value of 2.5 million. This is considerably beyond the range of experience with small bore rolling bearings. Successful application of small rolling bearings at this DN value could not be guaranteed even with an extensive research program because of the severe centrifugal effects. The operation of liquid lubricated fluid film bearings at these speeds would, of course, be out of the question because of the high power consumption. The feasibility of using rigid geometry gas bearings is determined by their stability limit. Stability limits for several well designed herringbone-grooved journal bearings were determined in reference [24].[4] For the rotor used by the authors supported in two herringbone-grooved journal bearings, each 0.5 in. (12.7 cm) dia and 0.5 in. (12.6 cm) long, the following stability limits were determined using the experimental data of reference [24]:[4]

Diametral clearance	Whirl onset speed
500 μin. (12.7 μm)	25,000 rpm
400 μin. (10.2 μm)	41,000 rpm
350 μin. (8.9 μm)	93,000 rpm
250 μin. (6.3 μm)	95,000 rpm

Thus it is seen that a most optimistic stability limit for herringbone grooved journal bearings would be somewhat less than

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[4] Numbers in brackets designate Additional References at end of discussion.

100,000 rpm. The significance of the authors results can now be appreciated since other bearing types appear to be incapable of providing reliable, stable rotor support under these operating conditions.

The experimental program conducted by the authors was, by their own admission, brief in scope. Several questions arise on which, hopefully, the authors might elaborate:

1 It is assumed that the mylar foils shown in Fig. 4 in the multiple layered foil bearings are, as the authors state, for illustrative purposes only. Were metallic foils used in all the bearings, and what was the foil material?

2 What was the extent of the running time at 210,000 rpm for the various bearings?

3 Was there a pattern of polishing on the MoS_2 coatings or was any surface damage noted on either the tensioned or multiple layered foil bearings?

4 Since the dynamic behavior of the conical bearing-rotor system was somewhat poorer than that with either cylindrical bearing type, what is the author's opinion on the importance of the shorter bearing span with the conical bearings?

5 As a matter of interest success with fluid film bearings capable of supporting combined loads (i.e., conical, spherical, etc.) in other experiments has generally been poorer than with separate journal and thrust bearings. Was operation with the conical bearings free of nonsynchronous whirl over the speed range up to 100,000 rpm?

6 Bearings capable of taking thrust load still appear to be the weak link in gas bearing technoly. Would the authors care to comment on this?

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Dr. A. Tondl[5]

I should like to congratulate the authors on the successful research and development of foil bearings, and the new, spirally-wound foil bearings in particular. Such bearings are very suitable in many applications as supports of high-speed rotors because of their ability to suppress self-excited vibrations, which limit the operational speed range of many machines. The results are very promising, particularly in view of the imposingly high rotational speeds attained in the course of experiments.

I should like to make a number of observations and raise various points, the clarification of which would undoubtedly require further experiments and additional data. My remarks concern the occurrence of both main and secondary, nonlinear resonances of various orders, as well as of self-excited vibrations. For example, with reference to Fig. 9, it would be of interest to determine the dependence of frequency of the self-excited vibration on the angular speed. Because of the high resistance of these bearings to self-excited vibrations, the whirl ratio would be expected to be less than one half, particularly at speeds well above the onset of instability. The observed trend of whirl amplitude with increasing speed is similar to the behavior I have observed during experiments with hybrid bearings, mounted on damped, resilient supports [25].[6] The self-excited vibrations occurred in a limited speed interval, in which the amplitude increased rather quickly with increasing speed and subsided slowly after reaching a maximum. With proper tuning of the resilient support, the amplitude of self-excited whirl remained sensibly small, and the rotor could traverse the region of instability and operate smoothly at high rotational speeds. Qualitatively, the characteristics of foil bearings are similar to those of bearings mounted on damped, resilient supports.

The occurrence of vibrations containing components of motion at frequencies well below the speed of rotation, as in Fig. 7(*c*) and in Fig. 13(*b*) for example, is rather interesting. Are these frequencies less than the lowest natural frequencies of the system? If so, the occurrence of low-frequency components of motion at high speeds of rotation represents phenomena that have not been properly explained thus far. I have encountered such cases in the course of obtaining solutions on an analogue computer for nonlinear systems, capable of self-excitation with harmonic force input or parametric excitation. (Single-mass systems with one and two degrees of freedom, and two-mass systems with two degrees of freedom were programmed.) In most of these cases, the self-excitation was of the Van der Pol oscillator type. In some cases, vibrations occurred over a wide range of the exciting frequency, in which the amplitude of self-excited motion changed very little, but the corresponding frequency varied considerably. I was able to establish certain correlations. For example, in the case of a single-mass system with external, harmonic excitation, the following applied:

$$\Omega = |2\omega_0 - \omega|, \tag{1}$$

where ω_0 is the natural frequency and ω the frequency of excitation. For the two-mass system with parametric excitation, wherein one of the springs changes stiffness periodically at frequency 2ω, the following relations held true:

$$\Omega = |\omega_2 - 2\omega|, \tag{2}$$

$$\Omega = |\omega_1 - 2\omega|, \tag{3}$$

$$\Omega = \left|\omega_2 - \frac{1}{2}\omega_1 - \omega\right|, \quad (\omega_1 < \omega_2), \tag{4}$$

in which ω_1 and ω_2 were the natural frequencies of the linearized, undamped system. It may be inferred, therefore, that the frequency of the low-frequency component may be given by the relation

$$\Omega = |n_1\omega_1 + n_2\omega_2 - n\omega|/N, \tag{5}$$

where $n_1, n_2 = 0, \pm 1, \pm 2 \ldots$ and $n, N = 1, 2 \ldots$ With regard to very-low-frequency components (i.e., $\Omega \to 0$), it is thus possible to determine the excitation frequency ω, in the neighborhood of which these vibrations will occur.

I believe that the authors would contribute greatly if they were in a position to supplement their findings with additional experimental data and vibration records over the entire speed range, and thus provide means of differentiating between individual phenomena and types of motion. Such a contribution would be particularly valuable for an analysis of low-frequency vibrations (reported in this paper and also observed with other rotor systems), the physics of which are not yet well understood.

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W. A. Gross[7]

Congratulations to Dr. Licht and Mr. Branger for making a clear presentation of a thorough, well-conceived and conducted experimental investigation of some novel types of foil bearings that appear to have significant promise. Careful, thorough, and accurate experimentation is an art that is comparatively rare and, therefore,

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[6] Numbers & brackets designate References at end of discussion.

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deserves expressed appreciation from the technical community.

Foil bearings, with gas as a lubricant, have been investigated for about the last 15 years because of the promise of freedom from instability, simplicity of construction, forgivingness with respect to foreign particles, and forgivingness with respect to manufacturing imperfections. Dr. Licht and Mr. Branger have probably conducted more experimental analyses of foil bearings than anyone else. This novel bearing design they describe appears to offer substantial fabrication advantages. It also deals with the thrust problem in a way that preserves the advantages of the foil concept.

I believe the authors correctly point out the difficulty of building a mathematical model to analyze the bearings. It appears that simplifications which permit mathematical handling would inadequately represent the bearing, and that the descriptive problems would be too complex to justify the effort required for analysis. For this reason, the experimental results presented are of special value. Designers will have to use their best judgment in seeking to apply this bearing to their applications.

Can the authors give data or predictions about the manufacturing tolerances permitted in design of foil bearings such as these?

Authors' Closure

The authors wish to thank Mr. Anderson, Dr. Gross and Dr. Tondl for their interest in our modest contribution and for their comments.

The response of these bearing-rotor systems is rich in various nonsynchronous components of motion, and it would have been soul-satisfying to furnish a thoroughly documented experimental sequel to this work. Unfortunately, with limited time and support at our disposal, we were unable to accomplish more.

In reply to Dr. Tondl's questions, and to facilitate classification of fractional-frequency whirl of very low speed ratios, the synchronous resonances in Fig. 7*(a)*, relevant to the bearing shown in Fig. 3, were approximately $N_1 \approx 400$ rps and $N_2 \approx 560$ rps. In this sense, assuming $N_1 \approx 400$ rps to be close to the lowest natural freency of the linearized, undamped system, the small-amplitude whirl at 4170 rps, Fig. 7*(c)*, has a frequency the order of 208 rps, or approximately $\frac{1}{2} N_1$. In the case of conical bearings, the resonances in Fig. 13*(a)* are less clearly delineated, but all occur in the bandwidth approximately 180 to 360 rps, while the whirl speed at 3000 rps is only the order of 125 rps.

Continuing with observations made by Dr. Gross, the difficulty of mathematical modelling is due mainly to ill-defined surface tractions and end conditions for the foil, which undergoes initially-large deformations and is then subjected to film-shape dependent pressure. It is consequently difficult to arrange for satisfactory marriage between elastic and fluid partners, even in the provincial parish of steady rotation and one-dimensionality. On the other hand, a semi-empirical determination of bearing characteristics may allow for adequate representation of bearing forces and for extended studies of nonlinear rotor dynamics proposed by Dr. Tondl [25].

With reference to manufacturing tolerances, the tensioned foil bearing, Fig. 3, will accommodate with ease deviations of journal diameters several times larger than a typical clearance, or more. It is also quite insensitive to out-of-roundness. For the spirally-wound journal bearing, Fig. 4, we estimate approximately: $c/r = (c_0 - \frac{1}{2}\, nt)/r \sim 0.004$, and $|\Delta c|/r = (|\Delta c_0| + |\frac{1}{2}\, n \Delta t|)/r \sim 0.0004$, in which $2c$ = least diametral clearance (foil layers and retainer bore in contact), $2c_0$ = difference between foil retainer and journal diameters, t = foil thickness, n = maximum number of foil layers across any diameter, and $\Delta \equiv$ deviation from nominal value. In the presence of creases and corrugations, much will depend on precision and accuracy of the deformed layer, but appreciable accommodation is possible with preloading, which in turn depends on the compliance of the corrugated section. In general, conditions are less stringent than with rigid gas bearings, or rolling-element bearings, except that in the latter case everyone takes the accuracy and precision of races, balls, and bores for granted.

Mr. Anderson's remarks and comparisons constitute a most valuable complement of this paper. In reply to his questions:

1 While use of nonmetallic foils is not precluded in some applications, the foils used here were of precipitation-hardening, PH 17-4 steel ($1.503 \times 10^9 \mathrm{N/m^2}$ = 218,000 psi yield strength).

2 Running near maximum speed varied between approximately 15 and 30 min. Bearings neither seized, nor overheated.

3 Unlike in the course of experiments reported in references [21] and [22], time did not allow for endurance testing. Whatever little wear was apparent, was evidenced by an even distribution of polished high-spots of the MoS_2 coating. Polished weartracks of insignificant depth were observed just inboard of the foil edges.

There were no tears at the edges and no damage in the interior. Hexagonally crimped foils with one inner coil displayed fine burnishing along transverse lines, midway between creases, i.e., along lines of contact between inner and outer coil. The conical bearing displayed localized wear along a narrow track, at $\frac{1}{4}$ the frustrum height from the base.

4 While a shorter bearing span reduces the reaction moments of bearing forces and is generally conducive to pitching, we doubt that lengthening the span of conical bearings would have radically altered their characteristics. We share the point of view that it is generally advisable to apply the "doctrine of separation of radial and thrust loads" and to use specialized bearings.

5 The conical bearings were stable through 90,000 rpm, and whirl trajectories were contained within circles 5 μm (190 μ in.) in diameter through approximately 125,000 rpm, Fig. 13*(a)*.

6 Thrust bearings are frequently required to support larger unit loads than journal bearings. On the other hand, it is difficult to optimize and control the clearance topography by bending flexible surfaces to maximize the load. The choice is frequently between running at near-contact conditions with a more accommodating surface, or with a larger (and therefore safer) clearance between surfaces that do not take kindly to concentrated contact when it occurs. Claims of load capacity, made for thrust bearings using foils and thin reeds, have been notoriously exaggerated and stated without reference to minimum film thickness. Such bearings will be adequate for loads that are moderate in comparison with those carried by well designed, rigid bearings. A compromise suggested by Mr. Anderson, that of foil-mounted sliders, offers an attractive compromise. Other concepts may involve suitably contoured membranes, subjected to fluid pressure on the side opposite the bearing film.