$$F\beta_r = \beta_1 \left[1 + \frac{\cos \psi_e}{\epsilon/u} \right] \qquad (13)$$

$$F\beta_t = \beta_1 \frac{\sin \psi_e}{\epsilon/u} \qquad (14)$$

The effective stiffness, damping, and transmissibility occurring during the tests can be determined from (10), (11), and (12) by substitution of (13) and (14).

A6 Consideration of the onset of cavitation suggests that there is a direct relation between the static oil pressure and the speed at which cavitation starts. In convenient, nondimensional terms this suggests a relation between P and ω/ω_s. As shown in Fig. 15, a number of experimental points plotted on log-log paper indicate a linear relationship which is given by:

$$P = 2(\omega/\omega_s)^5 \qquad (15)$$

If the static pressure is less than required by (15) cavitation can be expected. The validity of this expression remains to be checked for a wide variety of the variables.

DISCUSSION

J. M. Vance[2]

This paper gives experimental results from a high speed damper rig, data which is needed and useful for the following reasons:

1 The limited amounts of experimental data which have previously been reported are mostly from low speed rigs where principles of similitude must be relied upon for extrapolation of the results to higher speeds. It would be useful to learn if there are variables pertinent at high speed which have been overlooked in the low speed tests.

2 Data from the low speed rigs generally have supported the trends of the theory, but do not show accurate quantitative predictions necessary for design optimization. Although it is believed by the discusser that the wide data scatter mentioned by the author is generally more severe in the high speed rigs due to rotor dynamics effects, any data which can lead to the development of a more accurate prediction model will be a valuable contribution.

One of the most important and interesting conclusions of this paper is that cavitation produces nonsynchronous whirling and, therefore, limits the maximum speed at which dampers should be used. This conclusion directly contradicts even the basic trends shown by all existing bearing theories and design practices. Ordinary journal bearings, which exhibit nonsynchronous instability at high speed, can be made more stable by inducing cavitation. In a damper bearing, with no journal rotation, the effect of cavitation should be simply a reduction in the damping force. Past calculations by the discusser have indicated that practically all existing dampers in modern turbomachinery must be operating cavitated.

The discusser wonders if the reported nonsynchronous motion might be induced by one or more dynamic characteristics of the rig at high speed. Some possibilities are:

1 Gyroscopic precession and/or dynamic (as opposed to static) unbalance of the rotor. The rotor appears to have an extremely small stiffness in the rocking or pitching mode. Also, motion in which the journal does not remain aligned with the bearing axis has been shown to radically alter the pressure distribution in a bearing.

2 Internal friction. The rotor appears to be a built-up assembly with threads at the most flexible part, and also appears to have a spherical spline coupling at the drive shaft connection. Such sources of friction in rotating parts are known to produce nonsynchronous whirling.

3 Driving torque. In a paper to be presented at the ASME Design Conference in Chicago (April, 1976), the discusser will show that driving torque can produce nonsynchronous whirl in a precessing rotor, with the least amount of torque required in a rotor of the configuration of this rig.

Can the author give his opinion as to the possibilities of any of these or other mechanisms producing the nonsynchronous motion?

The 10 percent estimate of the inertia load of the nonrotating sprung mass is also of interest and is critical to the accuracy of the reported results. Can the author elaborate on the derivation of this estimate?

[2] Department of Mechanical Engineering, University of Florida, Gainesville, Fla.

P. R. Trumpler[3]

The writer has used oil-film dampers to effect dramatic improvement in whirl threshold of industrial turbomachines, but informal reports from industry suggest that such dampers often perform badly. Whether this is due to inadequate theory, bad engineering judgment about how to use the oil-film damper, or lack of reproducibility because we do not or cannot control certain so-called secondary factors, is an open question. Experimental studies are thus most welcome, and the author should be encouraged to continue his work and report his findings. The paper suggests certain questions, namely:

1 In early experiments it was found that internal friction of rotors (shrink-fitted disks, for example) was a clear cause of whirl instability, and this has been theoretically confirmed. Unless experiments are conducted to assure the negligible influence of the shaft screw joint, would it not be wise to construct an experimental rotor without such a joint?

2 What is the evidence that the spikes in the pressure diagram (Fig. 7) are produced by oil-film cavitation alone? The orbits of Fig. 8 as they change with increasing speed appear very similar to those produced by the onset of whirl in situations where cavitation does not enter. Could the spikes and orbits be produced by a phenomenon that is primarily dynamic in origin?

3 The prediction of performance using a linearized theory (β constant) should perhaps be justified. Oil film load-deflection functions are strongly nonlinear at high eccentricity ratios and this writer would expect the predicted curves of Figs. 11 and 12 to rise sharply as critical speed is approached if, as expected, film eccentricity ratio assumes values much above 0.5 at the high loads.

4 Are the functions of Figs. 13 and 14 intended to be general or to apply only to the particular apparatus and one particular set of operating parameters? In general, both damping and stiffness are functions not only of speed but also of force (or displacement) amplitude. Since force is a function of unbalance, the damping and stiffness are functions of the rotor balance accuracy. It is a common fallacy among designers to seek some one pair of values for oil-film stiffness and damping in the calculation of rotor response. Even if a different pair is taken for each speed, the effect of amplitude is almost always ignored and can be serious.

[3] Trumpler Associates Inc., Wayne, Pa.

Author's Closure

The author appreciates the comments from both discussers and offers the following response to the points that were raised.

1 The source of the nonsynchronous motion may be internal friction in the bolted rotor joint, in the bearing, or in the spline drive. No simple alternate designs were available. A more detailed analysis of the nonsynchronous motion has not yet been performed. It is expected to show half-frequency whirl and, possibly, frequency components of higher order. The rotor is not responsive to dynamic unbalance, the pitch mode occurring at very low speed due to the small stiffness. It is possible that precession occurs, but this would show up as deflections at very low frequency. The results of further data reduction will be reported, together with new data obtained on other geometries.

2 The results of efforts by others to establish the effects of driving torque and friction in spline joints are expected to be useful and may well provide a satisfactory explanation for the nonsynchronous motion. The reasons for linking the nonsynchronous motion to cavitation in the paper were not based on any special knowledge of cavitation, but on the observations that the onset speed of nonsynchronous motion was always slightly higher than the speed at which pressure spikes first occur, and that these pressure spikes occur at speeds which depend on the static oil pressure (see Fig. 15).

3 The estimate for the inertia load of the nonrotating sprung mass is derived from the measured load transmitted through the load cell and from the calibration data for the load cell which relates load to deflection and, hence, to acceleration.

4 The paper is concerned with the damper performance in cases of perfectly centered orbits at constant speeds. In those cases the eccentricity of the orbit is constant, the radial velocity of the shaft center is zero, and the eccentricity can be predicted in a simple analysis as shown in the paper. For values of the eccentricity smaller than 0.5, which is true for all test points in Figs. 11 and 12 and for all practical purposes in engine applications, the available oil-film damper theory can be approximated by a linear relation between load and deflection (β constant). The test points shown in Figs. 11 and 12 indicate that at speeds close to the critical the rotor behavior then does not confirm the theory. This may be due to cavitation or to other effects.

5 Figs. 13 and 14 are, like Figs. 11 and 12, valid for the particular set of parameters. For the linearized case studied in the paper the equations given in the Appendix show that the effective stiffness and damping are independent of the unbalance. This is also true for the transmissibility, but of course not for the damper deflection and the transmitted load. The common view that damping and stiffness are strong functions of eccentricity is not true according to the theory for synchronous behavior at eccentricities smaller than 0.5. If experience tells us otherwise it must be because it relates to rotors which respond in a nonsynchronous manner, e.g., due to gravity in horizontal rotors, or due to cavitation.