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would bring about closer correlation between theory and experiment for the smaller values of η?

Author's Closure

Mr. Zaid's correction of the author's force term on the center area of the plate is logical and it gives values of force under the limiting conditions of zero plate separation and large tube diameter which are more realistic than the author's. The fact, that the resulting characteristic curves as shown in Mr. Zaid's Fig. 2 depart from the experimental results even more than the original curve, is believed to indicate that a more detailed analysis of the flow in the transition from the tube to radially outward flow is necessary. Such an analysis would indicate a stagnation point at the center of the upper plate and a velocity gradient across the stream, with maximum velocities occurring along the bell-mouthed exit from the tube. With small plate separation, even with low flow rate, the velocity along this boundary may reach the velocity of sound, with the possibility of the formation of a shock wave. Even with subsonic flow there is a strong possibility of flow separation, as stated in the paper. This may be expected to occur along the bell-mouthed exit from the tube, resulting in an effectively narrower flow stream between the plates. It would account for the fact that for intermediate values of flow rate the experimentally determined values of force on the upper plate are higher than those predicted by the equations. At high flow rates, probably both flow separation and shock waves occur, resulting in the fact that the flow-rate factor fails to continue to increase with plate separation. It is believed that shock waves and flow separation are major factors as compared with variation in the pipe friction factor in causing the difference between the experimental results and the simple theory.

Some Dynamic Properties of Oil-Film Journal Bearings With Reference to the Unbalance Vibration of Rotors[1]

P. R. Trumpler.[2] The limitations of ordinary critical-speed calculations, which ignore bearing-support complexities and other significant factors, are becoming rather evident to many turbomachinery designers. The authors have presented important new information on the manner in which bearings may influence rotor motion. The writer agrees with the authors in their feeling that the proportioning of bearings for low vibration levels is an important new design concept.

The linearization of the bearing problem is a good first step. With the use of modern computers the removal of this limitation should not be too difficult. Linearization of the dynamic bearing equations leads to some important questions which cannot be answered until better solutions are available. The oil-whip phenomenon, for instance, is quite important and cannot be explained by equations of the form of the authors' Equation [1].

Stodola felt that fluid friction at the disk surfaces provided the damping which limited rotor motion at critical speeds; the authors say that damping in bearings does the job. It would be interesting to consider the applications in which each form is predominant. Also, the effect of seals and internal (hysteresis) damping may not be negligible.

[1] By A. C. Hagg and G. O. Sankey, published in the June, 1956, issue of the Journal of Applied Mechanics, Trans. ASME, vol. 78, pp. 302–306.

[2] Professor of Mechanical Engineering, Illinois Institute of Technology Chicago, Ill. Mem. ASME.

P. C. Warner.[3] The authors are to be congratulated on their fine paper. The data they present fill a large gap in the rotor-vibration picture, permitting practical calculations of vibration data which would not be possible otherwise. An understanding of the influence of journal bearings on rotor vibration is of prime importance not only in rotor design and rotor-stator-clearance specifications, but also in such later situations as field balancing, and vibration trouble-shooting in general.

Inasmuch as an extensive experimental program designed to test the validity of calculations using these data has been unavoidably delayed, we have made as many checks as possible utilizing such test data as were available. The range of bearing sizes checked runs from 3 in. diam to 20 in. diam, with a range of rotor weights from several hundred pounds to 90 tons. In each case, reasonable estimates of actual critical speeds were obtained by using the bearing data reported by the authors. This, in some cases, was somewhat surprising since the range of bearing operation involved was below the range tested by the authors, forcing us to use the dashed sections of the curves. While this is perhaps somewhat risky, there is in fact currently no alternative. While we still expect to do rather extensive testing,

[3] Medium Turbine Engineering, Steam Division, Westinghouse Electric Corporation, Philadelphia, Pa. Assoc. Mem. ASME.

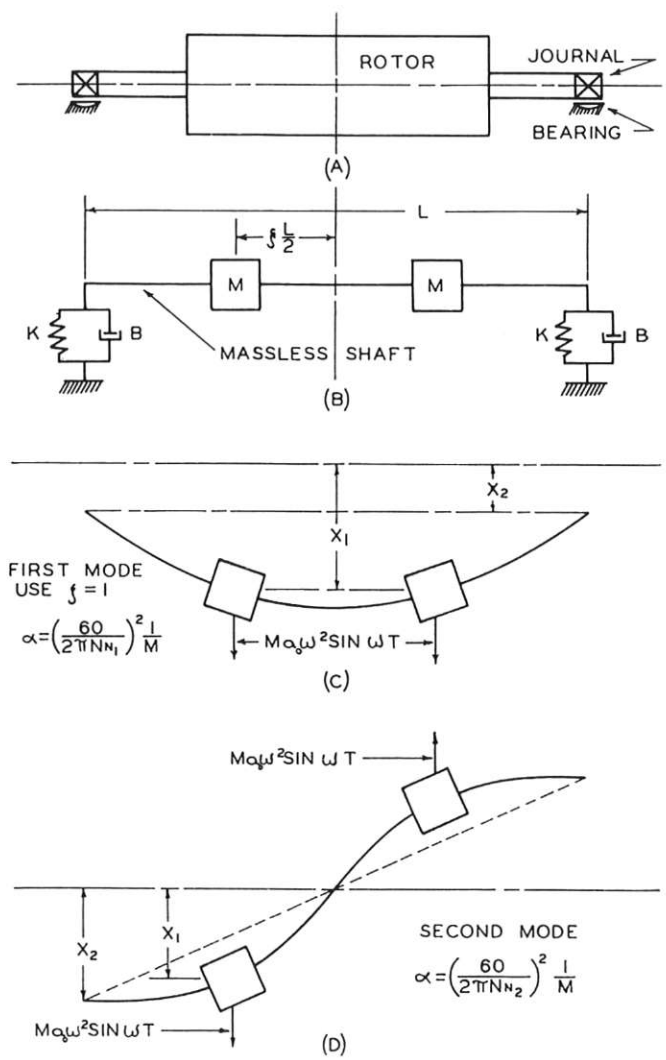

Fig. 1

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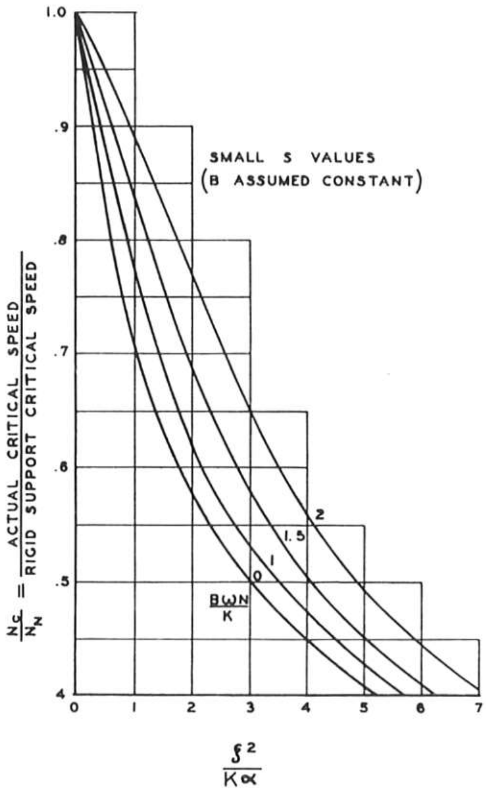

Fig. 2(*a*)

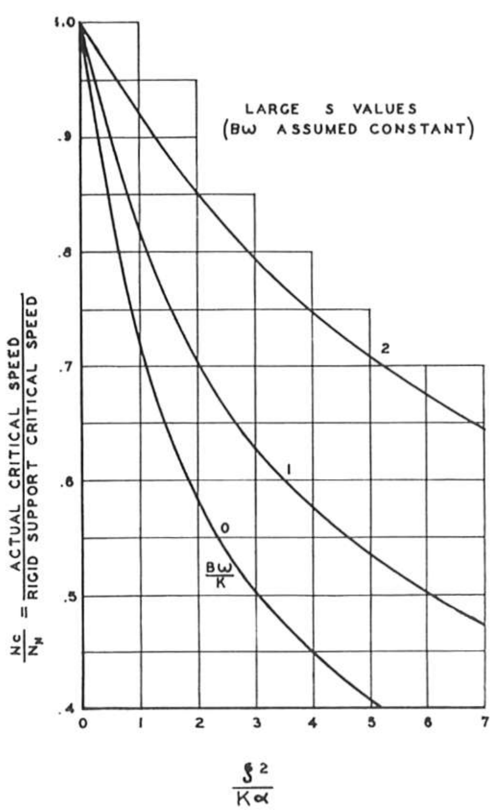

Fig. 2(*b*)

we feel at this time that realistic rotor-vibration calculations may be made using the authors' data.

The writer would like to take issue with the statement by the authors, that the inclusion of bearing conditions in critical-speed calculations was probably not practical until the advent of modern high-speed computers. It is strongly felt that simplified calculations of certain one and two-degree-of-freedom systems are not only sufficiently accurate when representing many actual rotors, but also afford a valuable insight into the rotor-vibration picture. This greatly increases the value of the bearing data since many people who are engaged in rotor design do not have a computer readily available. In the only cases available to the writer at this time, single-degree-of-freedom calculations of critical speeds deviated from high speed-digital computer calculations by 1.7 and 2.9 per cent.

The writer has considered the vibration of a rotor system similar to that considered by the senior author in reference (5), extending the calculations to include the second mode of vibration. Fig. 1 (*a* through *d*), herewith, indicates the system considered, and Figs. 2 (*a*) and (*b*) indicate some results. Two curves are required because of the essential difference in bearing action at high and low values of the Sommerfeld variable, as evidenced in the authors' Figs. 3 and 4.

The writer has found the charts of Fig. 2 of this discussion to be quite useful, particularly when a quick, reasonable estimate is needed. It is agreed however, that in those cases where a multirotor unit with many bearings is to be considered, a high-speed computer is a necessity. Even in such cases, if each rotor is supported in its own bearings, surprisingly good estimates may be made from Fig. 2.

Again, the authors are to be congratulated for presenting a paper of immediate practical importance which is of interest to such a large group of mechanical engineers.

Authors' Closure

The authors are grateful for the comments and suggestions of Prof. Trumpler and Mr. Warner, both of whom have made important contributions toward a better understanding of the vibration of rotating machines.

Equation [1] and the tests were designed to provide "equivalent" linear properties of oil films for the specific case of unbalance vibration. Stability considerations were deliberately avoided, and in the application of these dynamic properties to vibration calculations it is assumed that we have a stable system. This approach may have its defects, as Prof. Trumpler suggests; however, it seems that more general procedures await the time when operation of real bearings are more accurately specified.

Professor Trumpler also points out that sources of rotor damping other than the bearing oil films may be important. Friction of the surrounding medium can certainly be significant, as Stodola suggests, when the geometry of the rotor is favorable and when the density of the surrounding medium is high. A centrifugal water pump is an example in this category. On the other hand, the vibration magnification of a machine operating in air, hydrogen, or steam is usually far lower than would be expected on the basis of damping other than that provided by bearing oil films. Modern machines have quite rigid bearing supports, and hysteresis damping in nonrotating parts is believed to be very small. Rotor hysteresis is, of course, inoperative for unbalance vibration and has no effect on vibration magnification.

The experience reported by Mr. Warner is encouraging and lines up with our own experience in East Pittsburgh. Mr. Warner suggests that high-speed computers are not always essential, and he includes interesting and valuable charts based on simplified rotor systems. This should prove to be of especial interest to engineers concerned with critical speed calculations.

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