The paper is unusual in analyzing phenomena which have been observed and mentioned in the literature for many years. As the author indicates, it seems to be intuitively wrong that an increase in viscosity will negate damping effects in a hydrodynamic bearing.

In his own paper, the discusser touches upon these phenomena at some length. Unfortunately, the subject paper did not provide quite enough information to see if there was a quantitative confirmation of the stability parameter (called the “dynamic load number”) in the discusser’s paper. However, it appears that there is good correlation. Ruzicka also provides an insight with respect to the effect of too much damping in systems having distributed elasticity. The combination of the subject paper and the two additional references provides a very consistent presentation of important practical aspects of vibration technology.

Author’s Closure

The reference of Mr. H. J. Wood to his recent paper about the dynamic load number of the bearing is to be appreciated. The bearing capacity number as defined by Du Bois and Ocvirk for the modified bearing number due to the finite bearing length, has the following values for the bearing dealt with in the paper of the author:

\[ \Lambda' = \left\{ \frac{L}{D} \frac{W}{\mu_n} \left( \frac{C}{G} \right)^\pi \right\} \left( \frac{L}{D} \right)^3 - \Lambda \left( \frac{L}{D} \right)^2 = 0.79 \Lambda \]

\[ = 0.0335 \text{ for bearing oil at 25 deg C} \]

\[ = 0.0600 \text{ for bearing oil at 12 deg C} \]

For the running speed of \( n = 16.6 \text{ cps} \) for the rotor concerned, the bearing capacity number will be 0.50 and 1.0 for 25 and 12 deg C, respectively. From these we obtain the eccentricity ratio of the journal—\( e = 0.21 \) and 0.13, respectively (see Fig. 1 in the paper of Du Bois and Ocvirk); and the eccentricity itself—\( e = 0.032 \)

\[ \frac{1 - S^2}{S} \lambda_0 = 0.05 \lambda_0 = 8.2 \text{ in/sec}^2 \text{ for 25 deg C} \]

\[ = 6.8 \text{ in/sec}^2 \text{ for 12 deg C} \]

These values are much smaller than the critical one of about 100, so that a strong resonance at the critical speed is to be expected, as confirmed by the measurement. The two values differ by 20 percent for these two different bearing oil temperatures. As the difference of the vibration amplitudes of the bearing at these temperatures amounts to 30 percent for their maximum value and 23 percent for their minimum value (see Fig. 6 in the author’s paper), the dynamic load number \( \lambda \) offers, therefore, a useful measure for characterizing the damping of the journal bearing.

For proving the effect of the unequal elasticity of the bearing pedestals on the instability of the rotor, a further test has been carried out. The shaft of the turboblower shown in Fig. 1 was machined down to a smaller diameter, so that the natural frequency of the rotor was reduced to 13.2 cps. The vibrations of the bearings at the oil temperature of 25 deg C, as measured with this new rotor, are presented as curves \( a \) in Fig. 10. The beating character of the vibration is evident. Afterwards the bearing pedestals were reinforced to an equal stiffness and the same rotor was tested again. The vibration of the bearing pedestals was reduced to a great extent, as shown by curves \( b \). No essential beating exists any more. This test clearly shows the unfavorable character of the unequal elasticity of the bearings.