$$= 0.125 \left(\frac{2.000}{0.003}\right)^{1/2} (1)^{1/2}$$

$$= 3.22$$

(*e*) *Misaligned attitude*, m_a

$m_a = 0.72$ [from Fig. 10(*a*) for $C_m = 3.22$ and $\mathsf{S} = 0.119$]

(*f*) *Maximum eccentricity ratio*, n_e

$$n_e = m_a(1 - n) + n$$
$$= 0.72(1 - 0.45) + 0.45$$
$$= 0.846$$

By placing the pulley centrally on the hub, the misaligning couple would be reduced to zero, and the maximum eccentricity ratio would be $n = 0.45$ with a correspondingly larger minimum film thickness. It may be observed that as small a misalignment as 12.5 per cent almost doubles the eccentricity ratio at the end of the bearing, demonstrating the relatively low capacity of a bearing to support misalignment.

Illustrative Example 3

Twisting Misalignment. The idler gear with helical teeth in Fig. 14 is an example in which the tooth forces impose the condition of twisting misaligning. The tangential tooth-force component of F_t and the axial component is $F_a = F_t \tan \psi$ where ψ is the helix angle. $P = 2F_t$ is the radial load in the bearing, and the twisting couple is $M_t = F_aD$. The following data are assumed in the calculations of misaligned attitude

$F_t = 300$ lb, $P = 600$ lb, $\psi = 15°$, $F_a = F_t \tan \psi = 300 \tan 15° = 80.4$ lb

$D = 4$ in., $M_t = F_aD = 80.4 \times 4 = 321.6$ in-lb

$l = d = 2.0$ in., $c_d = 0.004$ in., $N' = 40$ rps, $\mu = 1.5 \times 10^{-6}$ reyn (SAE 20 oil at 180 F)

$p = 600/4 = 150$ psi

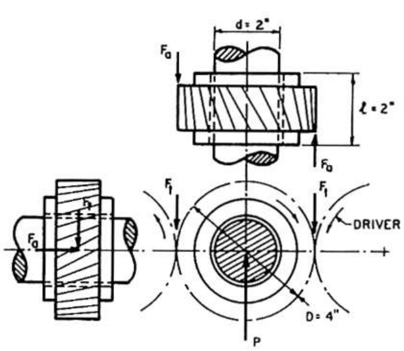

Fig. 14 Idler Gear With Helical Teeth Having Axial Tooth-Force Components Producing Twisting Misalignment

(*a*) *Sommerfeld number*, S

$$\mathsf{S} = \frac{\mu N'}{p}\left(\frac{d}{c_d}\right)^2 = \left(\frac{1.5 \times 10^{-6} \times 40}{150}\right)\left(\frac{2.000}{0.004}\right)^2 = 0.10$$

(*b*) *Couple variable*, C_m

$$C_m = \frac{M_t}{Pl}\left(\frac{d}{c_d}\right)^{1/2}\left(\frac{l}{d}\right)^{1/2}$$

$$= \frac{321.6}{600 \times 2}\left(\frac{2.000}{0.004}\right)^{1/2} (1)^{1/2}$$

$$= 5.98$$

(*c*) *Misaligned attitude*, m_a

$$m_a = 1.0\text{—from Fig. 10}(b)$$

For $m_a = 1.0$, metallic contact would occur at the end of the bearing.

Discussion

L. F. Kreisle.[8] The authors are to be commended highly for their excellent experimental investigations and correlation of data presented in this paper concerning the properties of misaligned oil-lubricated journal bearings with length-to-diameter ratios of 1.0 and more. With the increasing use of short journal bearings (with length-to-diameter ratios less than 1.0), properties of misaligned short journal bearings are very important. Fortunately, the authors list a source of this information[9] based upon their research investigations.

Their conception of a pivoting point at the bearing center is very useful in the computation of eccentricity ratios and corresponding minimum oil-film thicknesses at the ends of journal bearings. From the smallnesses of the applied central unit bearing load indicated in the examples, it is assumed that the experimental misalignment data presented were taken when hydrodynamic-film lubrication procedures were in evidence. Experimental findings of this investigator[10] for short journal bearings (length-to-diameter ratios of 1/53 to 1/2) without any misalignment indicate that marginal lubrication procedures initially begin to be substituted for hydrodynamic-film lubrication when the minimum oil-film thickness equals the sum of the predominant-peak surface roughnesses of the mating journal and bearing surfaces measured in the circumferential direction after run-in of the bearing-journal pair at a point axially centered between the ends of the bearing and for the bearing at a point corresponding to the location of the minimum oil-film thickness under operating conditions. This investigator would like to ask the authors if they have observed any similar criterion for the initial appearance of marginal lubrication procedures of misaligned journal bearings. If so, where should the minimum oil-film thickness be measured, at the center of the bearing, at the end where the minimum oil-film thickness is smaller, or at some other location?

The authors' photographs of oil-film pressure-distribution models of journal bearings are most enlightening. It is wondered if such a pressure distribution was employed to determine an average effective viscosity of the oil in plotting the Sommerfeld number for the curves or whether in most cases of oil-lubricated journal bearings the variation of viscosity with oil-film pressure is negligible in comparison with other causes, such as temperature variation.

The design curves of misaligning couple variable C_m versus Sommerfeld number S for various misaligned attitudes constitute a useful manner in which to present the results of what must have been many years of careful experimental investigations and painstaking correlation of data.

E. A. Ryder.[11] In this and other papers, the Cornell group has given us practical means of calculating bearing oil-film thickness, which is a more useful and more easily visualized parameter than the eccentricity. Misalignment is generally ignored, but the

[8] Associate Professor of Mechanical Engineering, The University of Texas, Austin, Texas. Mem. ASME.

[9] Footnote 5 of the paper.

[10] "Predominat-Peak Surface Roughness, a Criterion for Minimum Hydrodynamic Oil-Film Thickness of Short Journal Bearings," by L. F. Kreisle, published in this issue, pp. 1235–1241.

[11] Consulting Engineer, Pratt & Whitney Aircraft, Inc., Hartford, Conn.

examples show how easily it may be a troublesome factor in bearing performance.

One of the most useful things in the paper is the observation that misalignment does not change the eccentricity at the bearing center. In the case of applied misaligning couples such as gear loads (mentioned in the paper but not discussed) this enables one to calculate the position of the shaft axis and the minimum oil-film thickness at the ends of the bearings.

A good reason for the value of lead coatings on heavy-duty bearings is that if metallic contact occurs at the ends of the bearing, the lead can move out of the way without doing any harm.

Although not always stated thus, the easiest way to get at the minimum film thickness is:

n is the fraction of the radial clearance used up by eccentricity

m_a is the fraction of the remaining clearance used up by tilt

In the loose-pulley example, $h_{\min}$ at the center is 0.00082 in. and at the end, 0.00018 in. If the bearing is cut off to be symmetrical and $1^1/_2$ in. long, $h_{\min}$ will be 0.00039 in. or twice as much. It will certainly pay us to take a sharp look at any lopsided bearings.

Authors' Closure

The authors would like to express their appreciation to E. A. Ryder and L. F. Kreisle for their very kind remarks about the practical value of the methods shown for estimating the effects of misalignment. Mr. Ryder's summary of the corresponding meanings of misaligned attitude m_a and eccentricity ratio n in two brief sentences is an outstanding bit of clear writing that will be remembered. His comment that the film thickness in the eccentrically loaded example 2 would be doubled by cutting off the long end of the bearing is also pertinent, as this change is easily incorporated in existing machinery.

In regard to Mr. Kreisle's questions, while the main interest in the investigation is in the hydrodynamic range of misaligned attitude m_a from 0 to 1.0, some of the data gave values of m_a equal or slightly exceeding 1.0, indicating metallic contact and wear in a small area at one end of the bearing. At the start of the tests, it was found that the small component of the misaligning couple which could not be eliminated in the plane of rotation was of the order of magnitude of the friction couple. It was therefore decided not to attempt to report friction effects. However, there were indications that the increase of friction due to point contact at the end of a bearing is much less than the friction increase due to line contact in an aligned bearing. The viscosity used in preparing the data was the usual engineering average value obtained from a viscosity-temperature calibration line at atmospheric pressure using a temperature representing the average of several thermocouples imbedded near the bearing surface in the loaded area. The extent to which the pressure effect on viscosity offsets the temperature effect within the bearing is an area of current analytical interest, but the experimental data include these effects, and the simple parameters used on the charts facilitate use of the experimental data.