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for journal bearings, where $\bar{h} = (h/C)$, and by

$$(T_{\max} - T_1) = \frac{1}{\alpha} ln\left\{1 + E_T\left(\frac{R_1 + R_2}{2}\right)^2 \int_0^\beta \frac{d\theta}{\bar{h}_T^2}\right\} \quad (18)$$

for thrust bearings where, $\bar{h}_T = (h/\delta)$

In the above equation α is the temperature viscosity coefficient in the viscosity formula.

$$\mu = \mu_1 e^{-\alpha(T - T_1)}$$

Finally, Table 3 gives a comparison of journal bearing performance resulting from the use of three different inlet temperatures; cold inlet oil T_0, T_{1a} obtained from equation 2 ($\lambda = 0$), and T_1 as postulated here. With the latter considered the correct solution, Fig. 32 gives the deviation of the T_0 and T_{1a} solutions from the correct T_1 = 53°C (127.5°F) results. As seen while from the standpoint of performance, (ϵ, HP, etc.) both solutions yield relatively modest errors of the order of 5 percent–15 percent, from the standpoint of stiffness and damping the usage of incorrect values of T_1 can yield errors as high as 30 percent–50 percent.

Acknowledgment

Acknowledgment is due to Reliance Electric Company for its sponsorship of the present program and in particular to Mr. Paul Gorski for his sustained interest and support of this research.

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DISCUSSION

J. H. Vohr[1]

This paper addresses what I consider to be a relatively neglected but very important subject with respect to analytical prediction of the performance of journal bearings; namely, the problem of predicting the bulk lubricant temperature at the entrance to each bearing pad or arc. Many analysts have developed computer codes for calculating the temperature rise in the lubricant film along a bearing arc, but these have no usefulness for the important task of predicting the operating temperatures of a bearing if the inlet temperature to the arc is not known.

The authors are to be congratulated for the clarity and thoroughness with which they discuss the physical mechanisms involved in the carryover of hot oil from one arc to another. However, I do have a concern with the experimental results that are presented. This concern is that the authors appear to have assumed that the temperatures measured by thermocouples placed along the surface of a bearing arc at its leading edge provide an accurate measure of the bulk temperature, T_1, of the lubricant entering the arc. As I shall try to show in the following calculations, there can be a substantial difference between T_1 and the bearing surface temperature near the pad inlet. As a consequence, I would submit that the empirical factor λ determined by the authors serves only to correlate bearing arc inlet *surface* temperature, and not the lubricant inlet *bulk* temperature.

To demonstrate how pad arc surface temperature can differ from T_1, let us consider one of the typical operating conditions for the 305 mm (12 in.) experimental bearing, i.e., T_0 = 48.9°C (120°F), P = 690 kPa (100 psi), and 60 Hz. Applying a bearing computer program to solve for the operating flows and temperatures of this bearing under these operating conditions and assuming that λ = 0, i.e., that equation (2) holds, we obtain the following results:

$Q_2 = 1.56 \cdot 10^{-3}$ m³/s	(21.3 GPM)	
$T_2 = 79.9$°C	(175.8°F)	
$Q_0 = 0.51 \cdot 10^{-3}$ m³/s	(7.0 GPM)	
$T_0 = 48.9$°C	(120°F)	
$T_r = 77.2$°C	(171.1°F):	Journal Temperature
$Q_1 = 2.07 \cdot 10^{-3}$ m³/s	(28.3 GPM)	

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Referring to Fig. 12(b) of the paper under discussion, we ask the question: to what extent has the hot stream of oil Q_2T_2 mixed with the cold make-up stream Q_0T_0 by the time the flows arrive at the pad inlet θ_1. If the flow Q_2 leaving the trailing edge θ_E of the upstream pad is laminar, then little or no physical mixing will occur, although some *conduction* of heat will occur from the hot stream to the cold stream as discussed in this paper and in reference [5].

Neglecting any physical mixing of Q_0 and Q_2, the heat transferred from Q_2 to Q_0 as the two streams cross the oil groove can be calculated in a reasonably accurate fashion by assuming that the lubricant oil film starting at θ_1 extends back to θ_E in a stream tube depicted by the dashed lines in Fig. 33. For simplicity, the velocity profile across this stream tube will be assumed to be the same as the fully developed profile attained in the bearing pad just downstream of θ_1. Temperature of the stream Q_2 entering this stream tube at θ_E is taken to be uniform at T_2 while the temperature of the flow Q_0 is taken to be uniform at T_0.

In the pad downstream of θ_1, physically reasonable temperature boundary conditions for the lubricant flow are that the shaft surface stays at a constant temperature T_r while heat transfer at the bearing surface is assumed to be negligible. For simplicity, these boundary conditions will be assumed to hold along the hypothetical stream tube in the groove.

With these temperature boundary conditions, the temperature profile throughout the groove stream tube and into the lubricant film downstream of θ_1 may readily be computed by a finite difference solution to the two-dimensional convective heat transfer equation applicable to this region. Conduction in the θ direction is neglected. Dissipation due to viscous shear in the groove stream tube and in the bearing film is included assuming constant viscosity evaluated at $T = T_1$. Using the computed data for Q_2, T_2, Q_0, T_0, and T_r cited above, the oil temperatures along the boundaries of the stream tube-bearing film central volume as calcualted for laminar flow conditions are shown in Fig. 34. For the 305 mm (12 in.) test bearing, the first line of thermocouples used to measure arc temperature are located 5° downstream of θ_1, the arc inlet. At this location, the calculated bulk fluid film temperature is T_1 = 72.9°C (163.3°F) while the calculated wall temperature is 53.8°C (128.9°F), substantially lower than the bulk film temperature.

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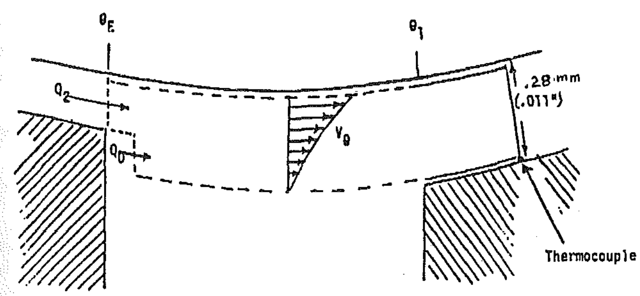

Fig. 33 Control volume for laminar flow across groove

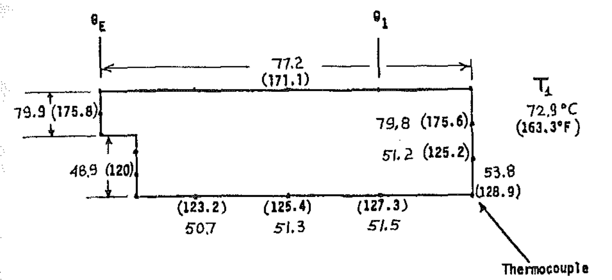

Fig. 34 Temperature distribution around laminar flow control volume

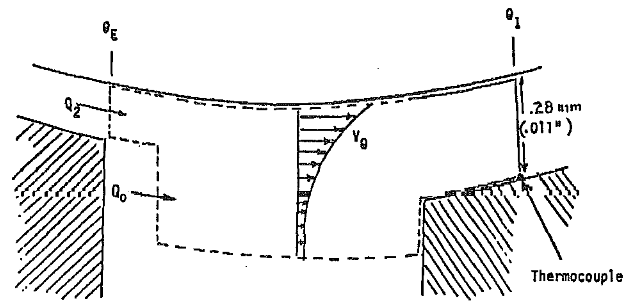

Fig. 35 Control volume for turbulent flow across groove

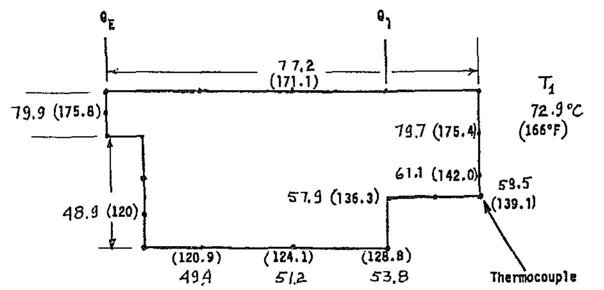

Fig. 36 Temperature distribution around turbulent flow control volume

For the case in question, the calculated Reynolds number at the entrance to the arc is 887 while the calculated Reynolds number for the flow exiting from the previous arc is 1303. Thus, the flow in the bearing films is close to the turbulent transition point, so that the assumption of laminar flow prevailing in the boundary layer crossing the lubricant groove may not be valid. A second calculation of temperature distribution was therefore made assuming fully developed turbulent flow in the groove and using an eddy viscosity formulation for turbulent heat transfer described by Keys.[2] A somewhat enlarged control volume was employed as shown schematically in Fig. 35. The resulting temperature distribution around the control volume is shown in Fig. 36. At the aforementioned thermocouple location, the bulk lubricant film temperature was calculated to be 72.9°C (166°F) while the wall temperature was determined to be only 59.5°C (139.1°F).

If the calculated wall temperatures cited above are interpreted as being the bulk inlet temperature T_1, corresponding values for the correlation parameter λ may be calculated. These are shown plotted along with the experimental values for λ shown in Fig. 37 (Fig. 19 of the paper). As can be seen, these theoretically obtained values for λ bracket the experimental values presented in the paper. This supports the contention that the expressions for λ presented in the paper correlate inlet *wall* temperature rather than inlet *bulk* temperature, and that a more accurate evaluation of T_1 may very well be obtained by using equation (2) directly than by using the correlation values presented for λ.

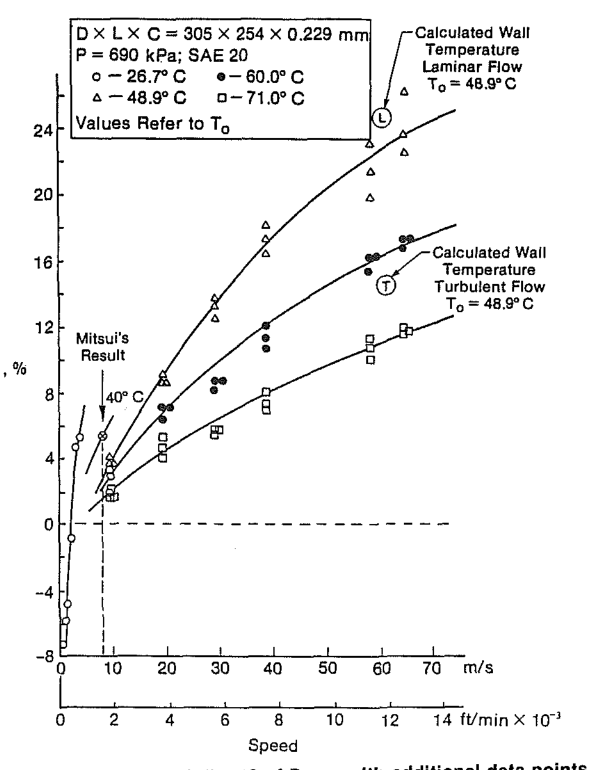

Fig. 37 Reproduction of Fig. 19 of Paper with additional data points

C. M. McC. Ettles[3]

This paper is concerned with one of the most neglected aspects in bearing analysis, which is that the actual entry temperature at the start of a bearing film is considerably greater than the temperature of the supply being fed to a groove. The authors have used a flow balance to give the leading edge entry temperature T_1 as

$$T_1 = T_0\left(1 - \frac{Q_2}{Q_1}\right)\frac{1}{1+\lambda} + T_2\left(\frac{Q_2}{Q_1}\right)\frac{1}{1+\lambda} \qquad [1A]$$

Equation [1A] is in the author's notation and is in the form in which the results of the paper would be used. The quantity λ accounts for heat lost from the groove, and could represent (for example) the proportion of the thermal boundary layer which is deflected by the leading edge of the bearing shoe and not captured by the film. The mixing of inlet flows is a logical basis for determining an effective inlet temperature and this method has been used by Robinson [1a],[4] Stokes [2a], the

[2] Keys, W. M., *Convective Heat and Mass Transfer*, McGraw-Hill, 1966.

[3] Rensselaer Polytechnic Institute, Troy, NY 12180-3590.

[4] Additional references are given as [1a], [2a] etc.

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discusser [3a] and probably several others. In the cases quoted above the imbalance parameter λ was taken at zero.

A more detailed approach is to attempt to solve the heat and mass transfer equations in the groove (4a, and reference [3] in paper). The results show that mixing occurs in the boundary layer that forms on exit from the film. The boundary layer increases in thickness across the groove. The entry flow to the film is contained in a streamtube actually adjacent to the moving surface. Figure 38 shows an isothermal solution from [4a] of the streamlines and pressure contours in the transverse section of a feed groove. The length to depth ratio $L/B = 10$ and the Reynolds number $\rho\ UB/\eta = 1000$. A small amount of makeup flow is admitted through the top of the domain. The kink in the streamline $\psi = 2$ shows how the makeup flow joins the recirculating flow. The dashed lines and point values around the domain show proportions of the nondimensional pressure p^*, where $p = p^*\rho U^2$. The lower diagram (39) shows the variation of pressure along the rotor surface. The direct entry of a make-up stream Q_0 at the cold (supply) oil temperature T_0 (as shown in Fig. 38 and Figs. 11, 12 of the paper) appears from the numerical solutions to be unlikely. From a pragmatic point of view it could be said that this makes little difference, since mixing does occur but within the entry streamtube which extends across the whole groove, rather than at a discrete point at film entry.

Considering the entire transverse section of the groove has advantages and also drawbacks. The stream function distribution (Fig. 38 and Fig. 4 in paper) shows the whole flow pattern and allows the pressure field to be determined. Pressure ram effects appear to be greatest at the leading edge and occur from inertial as well as viscous mechanisms. The occurrence of ram pressures has been verified in [5a] when piezo electric pressure transducers were installed immediately below the rotor surface. It is also possible to determine the actual temperature distribution at entry to the film, and such details as surface heat transfer coefficients on a local basis.

Concerning the validity of such solutions, Vohr [5] derived a numerical value from [3] of 340 w/m^2 for convection losses from the rotor surface in his experimental configuration. This value is an average of the coefficient on the three walls and the rotor surface. On the rotor-surface alone the results of [3] give a value of about 1000 w/m°C which compares more favorably with the experimental results but is still too low by a factor of about 3.

This discrepancy brings out the drawbacks of considering the entire groove cross-section. In particular the solution method used in [3] is satisfactory only for laminar flow whereas the conditions in Vohr's experiments were probably turbulent. Also it is necessary to use a very fine grid to reduce truncation errors. A coarse grid of only 11×11 was used in [3] due mainly to the limitations of available computers when these solutions were obtained twenty years ago. An alternate is to consider only the entry streamtube which reduces the size of the domain (normal to the moving surface) by about two orders of magnitude. This allows far better resolution of the temperature in the area where it is actually important. The streamtube configuration is used by Vohr in his discussion to this paper. This discusser has had the benefit of seeing Dr. Vohr's discussion before publication. The method used by Vohr is (in the discussers view) the best presently available, since it considers mixing as it actually occurs and gives better resolution than modeling the whole groove. The use of the thermal boundary layer equations as in [2] are an alternate to solving the propagation problem described by Vohr, although the results will clearly be less accurate.

Within the streamtube or whole groove methods the factor which most directly affects the result (in terms of the film entry temperature) is the surface temperature of the rotor. Vohr [5] has used a heat balance of the whole rotor to determine this value. In the early paper [3], the discusser assumed that the

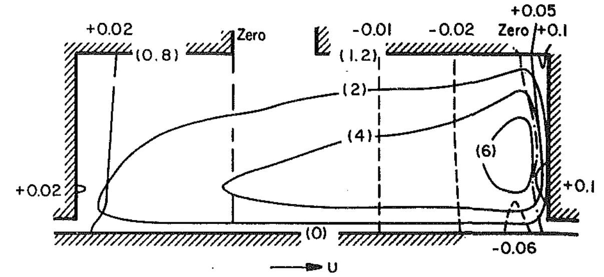

Fig. 38 An isothermal solution for flow in a transverse groove of depth B (normal to sliding direction) and length *L*. The vertical scale is expanded by a factor of about 3.9 compared to the horizontal scale. Values of stream function are shown as ( ). Unbraced values are of nondimensional pressure.

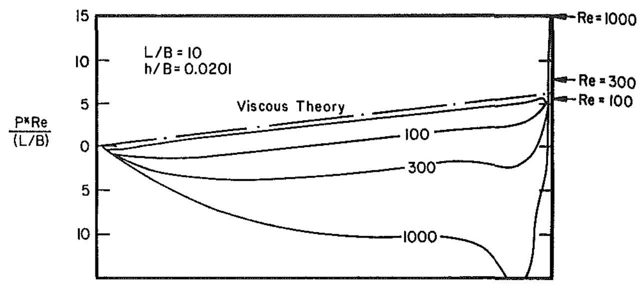

Fig. 39 The variation of pressure along the rotor surface for various Reynolds numbers $\rho UB/\eta$

rotor surface temperature was the average of the pad temperature. Data concerning the hot oil carry-over coefficient k have been considerably refined since the 1970 paper [4] quoted by the authors. The data contained in [6a, 7a] are based on many hundreds of test results of thrust bearing assemblies varying in size from 75 mm to 3000 mm diameter. This discusser believes that the hot oil carry-over method is fairly satisfactory in spite of criticism that has appeared in the literature. In fact, all methods used to solve the entry temperature problem resort to empiricism. Perhaps in all cases there is too much concern with getting the "right answer."

As a point of detail the final result of the hot oil carry-over formulation for the leading edge temperature has similarities with the authors' formulation [1a]. The hot oil carry-over result is (in the notation of the paper)

$$T_1 = T_0\left(\frac{2-2k}{2-k}\right) + T_2\left(\frac{k}{2-k}\right) \tag{2A}$$

the similarity between this and [1A] is that the entry temperature T_1 is expressed as weighted values of the supply temperature T_0 and the trailing edge temperature T_2 in the form

$$T_1 = T_0 \bullet C_1 + T_2 \bullet C_2 \tag{3A}$$

In [2A] the weighting coefficients sum to unity and this is also the case for [1A] if $\lambda = 0$.

In conclusion, this discusser feels that the flow mix method as described in the paper is robust and applicable to a wide range of configurations. The streamtube or whole groove models deal more directly with the actual flow in the entry region to the film. The earlier whole groove solutions were obtained with a coarse grid. The boundary conditions for those solutions should be re-examined. The streamtube method gives much improved resolution. In both methods the rotor surface temperature strongly affects the result. Methods for adequately dealing with turbulence need to be evaluated.

The whole groove or streamtube methods give the required information at inlet for three dimensional thermohydrodynamic solutions. Such solutions will probably become more commonplace as the cost of numerical computation continues to decrease. Furthermore, use of these con-

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figurations will allow the spray bar and leading edge groove configurations to be analyzed. These devices give extremely worthwhile improvements on bearing performance.

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Authors' Closure

Dr. Vohr is convinced that what we measured was the surface temperature of the bearing, and not something else. We shall try to show that whatever it was we measured it was certainly not the surface temperature. Figure 40 shows the topography of thermocouple location and mounting vis-a-vis the fluid film and adjacent surfaces. The picture is drawn to scale, with the actual dimensions as indicated. Note should be taken of the fact that thermal insulation separates the thermocouple from the bearing metal; that the thermocouple head is closer to the runner surface than it is to the babbitt; and that the only thing it is in contact with is a pool of fluid. The fluid film that is carried into the indentation housing the thermocouple certainly does not behave as if it were not there, so that fluid circulation is set up in the quasi-spherical well about the thermocouple. What then is the temperature registered by the thermocouple? We do not know exactly, except that it is not the singular babbitt temperature at the lower end of the hydrodynamic film but more likely some bulk temperature of the surrounding fluid.

That circulation and mixing occur in a geometry of this sort is supported by the experimental data of Fig. 41. This shows an oil groove at the entrance to the pad which has a similar cross-section, though the groove is much larger than the thermocouple well; but this merely reinforces our argument. In Fig. 41 we have recorded three temperatures $T_0 = 50.6°C$ (123°F), $T_G = 55.6°C$ (132°F) and $T_1 = 60.6°C$ (141°F), equidistant from each other. Now according to the discusser's argument, T_0 should read close to the incoming cold oil i.e., 50.6°C, because the point is way below the hot oil and the adjacent mixing zone. One could, of course, argue that the 5°C rise, (55.6 − 50.6), was due to heat transfer from the wall but this is difficult to support because over the next interval of 2 cm, the same as the distance between T_G and T_0, the rise in temperature was also no more than 5°C (60.6 − 55.6) and that region included the hot oil carry- over Q_2T_2 and the shearing losses in the narrow gap of the oil film, in addition to any possible heat transfer. It is thus not heat transfer that is responsible for the 5°C rise from T_0 to T_G but more likely circulation and mixing of some of the upper layers of the hot oil. This would support our argument about the thermocouple measuring not a surface temperature but some mixing bulk temperature of the surrounding fluid.

More serious than the lack of faith in our thermocouple is Dr. Vohr's conclusion that the value of λ must be zero. Such a conclusion, of course, says that all the perturbations cited in our paper, the complex flow in the groove, backup of cold oil, loss of hot oil, etc. do not take place and that Q_2T_2 mixes with

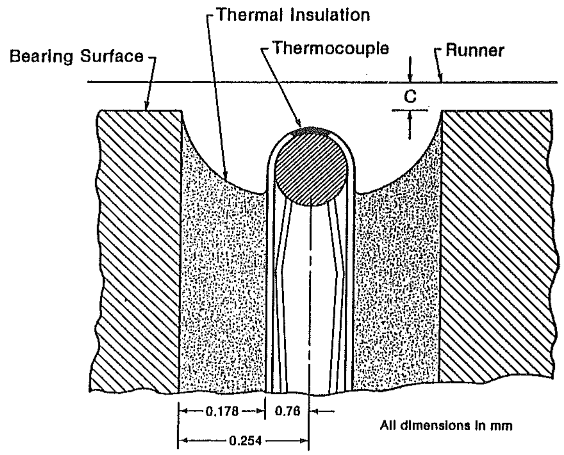

Fig. 40 Construction and location of thermocouple for measuring T_1

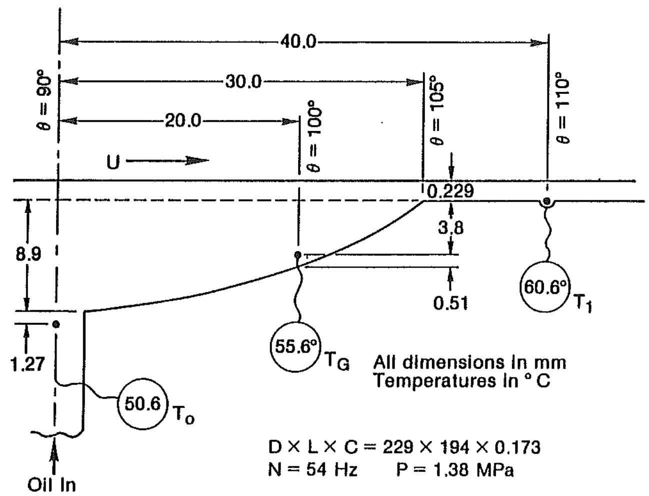

Fig. 41 Geometry of oil groove and temperature measurements

Q_0T_0 in the most ideal of fashions. It is here worth noting that the use of the simple relation

$$T_1 = \frac{Q_2T_2 + Q_0T_0}{Q_1} \tag{C-1}$$

yields, for the example cited, a $T_1 = 72.2°C$ (162°F) whereas Dr. Vohr's complex thermal analysis yielded $T_1 = 72.8°C$ (163°F), that is all the theoretical refinements added little to the simple mixing equation. It is this thermal analysis and the imputation to our thermocouple of having measured surface instead of a mixing temperature that led to the conclusion that $\lambda = 0$. But a $\lambda = 0$ is it variance not only with our effort, but, unfortunately, also with other experimental results on mixing temperatures. This is the work of Mitsui et al. (1983), [C-1], which, regrettably, we were unaware of when we wrote our paper. In the experimental setup of this paper, the authors measured separately surface temperature by embedding thermocouples a few mils below the babbitt, and separately shaft temperatures via a slip ring. They then obtained their mixing temperature T_1 by fitting a temperature profile across the film. In this case, therefore, there is not much doubt about the validity of T_1. The Japanese paper uses a mixing function η_m to represent the various perturbations and it can be related to our λ as follows

$$\lambda = (1 - \eta_m)\left(\frac{\bar{\bar{Q}}_2}{1 + \bar{Q}_2}\right)\left[\frac{T_2 - T_0}{T_1 + \delta\,\dfrac{160}{9}}\right] \tag{C-2}$$

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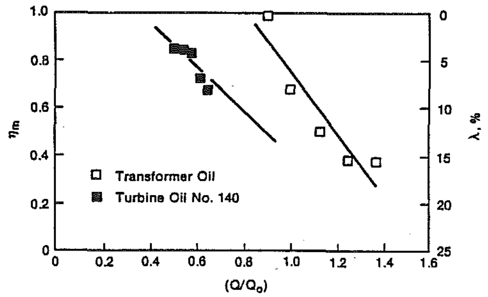

Fig. 42 Experimental values of mixing functions, reference [C1]

with $0 \le \eta_m \le 1$. When $\eta_m = 1$, $\lambda = 0$; when $\eta_m \to 0$, λ assumes various positive, often fairly large, values. In other words, the range of η_m corresponds to our range of λ. Now, if we put in the flows and temperatures from the discussor's example into the expression for η_m in reference [C-1], it very consistently yields $\eta_m = 1$ (i.e., our $\lambda = 0$). Unfortunately, this is not the η_m that the authors obtained from their experiments. Figure 42 shows the values of η_m and the corresponding λ's that the experiments yielded using two different kinds of oil. They are nowhere near the η_m demanded by Dr. Vohr's analysis. They not only support our $\lambda \ge 0$ mapping, but more importantly the experiments of Mitsui et al. corroborate our functional correlation of λ with $\bar{U}$ and $\bar{T}_0$ which, to our mind, is one of the main contributions of the paper, irrespective of what the exact values of λ may turn out to be. Using Mitsui's data for the heavier oil, which yields about the same $\epsilon = 0.2$ for their 100mm (4 in.) diameter bearing as for our 300mm (12 in.) bearing, we have for the Mitsui bearing the following flows

$$\bar{Q}_2 = 0.83 \quad \bar{Q}_s = 0.384 \quad \bar{\bar{Q}}_2 = 2.16$$

whereas from their experiments we have

$$T_0 = 40°\text{C} \quad T_2 = 57°\text{C} \quad T_1 = 46°\text{C}$$

$$0.6 \le \eta_m \le 0.8$$

yielding from equation (C-2)

$$\text{for } \eta_m = 0.6 \quad \lambda = 7.3\%$$

$$\text{for } \eta_m = 0.8 \quad \lambda = 3.65\%$$

Now the linear velocity for the Mitsui bearing is 9.56 m/s (1883 ft/min) and $T_0 = 40°\text{C}$ (104°F). If we take the average of the above values, i.e, $\lambda = 5.5$ percent and place it on the chart of Fig. 37 (Fig. 19 of the paper), it fits perfectly in relation to its functional dependence on $\bar{U}$ and $\bar{T}_0$.

With regard to Prof. Ettles' comments, first the authors would like to say that their approach was not meant in any way to substitute, or compete with a detailed groove analysis, be it the total groove or the streamtube method. Their simple approach was motivated by similarly simple motivations. One is that in actual bearing work, the goings-on in the groove per se are of little interest to the designer or the analyst. The other point is that, as stressed in the paper, what is aimed for here is a method of estimating T_1 without resort to thermohydrodynamic analyses which are always complex and rarely satisfactory. We join, in particular, Prof. Ettles in his pointing out the unresolved issue of journal or runner temperature, among many others. It is gratifying to read Prof. Ettles' summary that in the end the more elaborate approaches boil down, as does our approach, to a set of weighting functions attached to the cold and hot temperatures, T_0 and T_2.

In a recent paper dealing explicitly with the subject of thermal research in tribology, reference [C-2], one of the authors tried to highlight the disarray prevailing in this field. Specifically it was felt that it "seems to be the peculiar nature of research on thermal effects that each technical paper and each set of new results, while certainly shedding more light, at the same time adds new complexities to the problem. As a result, in place of a steady if slow resolution of the difficulties, we face a discipline which is continually expanding and becoming ever more formidable." This discussion sadly corroborates this pessimistic outlook. Dr. Vohr feels that a thermal analysis can give "reasonably accurate" solutions for the almost intractable flow conditions in the oil groove of a multipad hydrodynamic bearing. We do not share this confidence and have tried to cut through the unreliability of much of the theoretical thermohydrodynamics at least with respect to T_1 which, in essence, is no more than a boundary condition. A complete thermal analysis of the fluid film and surroundings is still much more complex and is, at the present stage, far from having been adequately resolved.

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