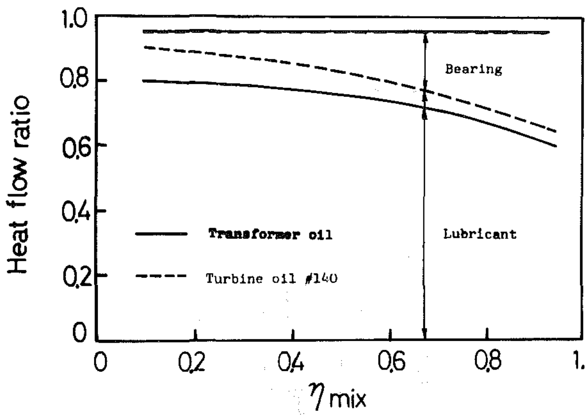

Fig. 9 Theoretical heat balance against η_{mix}

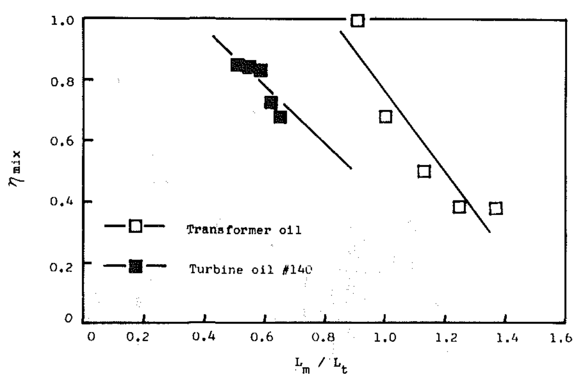

Fig. 10 The relation between the mixing coefficient and supply oil flow rate

Study of Results

From Figs. 3 and 4, the correlation between the supply oil flow rate and the mixing coefficient η_{mix} is conjectured. Two groups of temperature distribution of the bearing surface, obtained theoretically for one group and experimentally for the other, are found to be in good accordance with each other for a certain variation range of the mixing coefficient and the supply oil flow rate. As the mixing coefficient increases, the film temperatures at respective depths in thickness tend to rise after the mixing process throughout the whole thickness range, and the journal surface temperature becomes influenced thereby, as Fig. 5 indicates.

The pictures appearing in Fig 7(*b*) and Fig. 7(*c*), which give flow patterns before and behind the inlet oil groove, may contribute to an explication of the cooling effect and cooling mechanism due to the supply oil. In the case of a small amount of supply oil, the backward flow tends to exist mainly in the bearing midwidth area, while it is spread over the whole bearing width in the case of a large amount of the same oil. The temperatures go down after the mixing process by virtue of the cooling effect of the supply oil, though they are still higher than the supply oil temperature itself. The isotherms after the mixing process gradually turn into uniform lines running in the axial direction at certain angular distances of 40-50 degrees from the inlet oil groove, while they exhibit considerable convexity just behind the inlet oil groove.

Figure 8 illustrates the heat balance of the test bearing in relation to the nondimensional supply flow rates. Two separate cases, one using transformer oil and the other #140 turbine oil, are represented by solid and dotted lines respectively. The heat balance obtained by the thermohydrodynamic calculation is shown against the mixing coefficients as variables in Fig. 9. Likewise in this case, solid and dotted lines are used to follows the same representing system as mentioned above.

On the basis of these figures, the relation between the supply oil flow rate and the mixing coefficient is obtained by making a comparison between the theoretical heat flow rates and the experimental ones of the lubricant, the result being shown in Fig. 10. The mixing coefficient ranges roughly from 0.4 to 0.8, while the supply oil pressure varies between 19.6 and 177 KPa.

Conclusion

The cooling effect of the supply oil in the circular journal bearing was investigated theoretically and experimentally based on THD analysis. The mixing coefficient η_{mix} was introduced in the course of the theoretical development to investigate the cooling effect as influenced by the supply oil flow rate.

The theoretical results were in good agreement with the experimental ones. The cooling mechanism affected by the supply oil was also investigated by observing the oil flow behavior through the transparent plastic bearing. The existence of the backward flow into spaces left by the contracted streamers in the cavitation region was recognized, and it certainly seemed contributive to the cooling effect in the cavitation region to a substantial extent.

The relation between the mixing coefficient and the supply oil flow rate was obtained from the comparison between the theoretical and the experimental heat balances. It was found that the mixing coefficient remained within the range roughly from 0.4 to 0.8.

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DISCUSSION

A. Z. Szeri[1]

The authors are congratulated for attacking an important, and hitherto neglected problem in the design of large journal bearings. Conditions at oil inlet are far too complicated for rigorous analysis and we are forced to put our faith into the type of energy balance employed by the authors of this paper. This discusser would like to point out the importance of the inlet pressure; this, of course, might have been considered by the authors in their analysis and discarded subsequently.

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It is not clear at what angular position is the full hydrodynamic film assumed to begin in the THD analysis. Only if the inlet pressure is larger than the pressure required to uniformly distribute the lubricant at the "leading edge," and thus create a continuous film, will the effective bearing arc equal the theoretical one. This condition might serve as a vehicle for the introduction of the inlet pressure into the analysis.

The inlet pressure of the lubricant might also have another role to play. It is well known that at large eccentricity backflow will occur near the bearing surface at inlet [9]. This is especially true for full journal bearings [10]. How much will such back flow affect the authors energy balance in and around the grove will depend, to a certain extent, on the supply pressure.

Lastly, the discusser is puzzled by the fact that the energy equation is employed in the parabolic form. This, of course, limits the applicability of the numerical analysis to, say, $\epsilon<0\cdot5$. Does this mean that experiments were performed only with this condition?

Additional References

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Authors' Closure

The authors would like to express their gratitude to Professor Szeri for his valuable discussions and comments.

It seems that Professor Szeri presumed the inlet pressures the authors used to be pretty high. On the contrary, they were low enough to be neglected in the experiments and were therefore discarded in the analysis.

In the THD analysis, the full-width hydrodynamic film was assumed to begin at the angular position of the oil groove and this condition was achieved at very low inlet pressures in the experiments, because the oil groove of the bearing was very long (the groove length to bearing width ratio was 6/7). Therefore it was assumed that the effect of inlet pressure was negligible in the analysis. Furthermore, the oil film shrinkage after the oil groove was assumed to be negligible in accordance with the experimental observations. The oil film pressure was also assumed in the analysis to develop from the position of maximum film thickness.

Since the authors considered only the cases of low inlet pressures, it was not necessary to deal with the backflow in the oil film. The sign of the velocity gradient $\partial \bar{u}/\partial \eta \mid_{\eta=0}$ was checked at every grid point in the calculation and was found to be positive up to $\epsilon = 0.9$. This means that no backflow appeared in the film until $\epsilon = 0.9$. Experiments were carried out at eccentricities less than 0.85.