Discussion on Paper IX (iii) by A. Ronen and S. Malkin : "Lubrication of journal bearings : the impact of oil contamination on wear and energy losses"

local energy density because of the transient nature of the phenomenon, the high energy gradients and the very localised nature of the effect.

Reply by Dr. A.N. Bramley, Dr. S.W. Dunning and Dr. C.M. Taylor (University of Leeds). Dr. Baker's comments are thought provoking. The paper describes preliminary experiments to investigate the effect of (i) gap between the probe and bearing surface (ii) vibrational amplitude (iii) bulk oil temperature. Our findings here were in accord with those of a number of previous workers. In particular, as Dr. Baker points out, a maximum erosion rate was obtained as the film thickness gap was increased, other parameters remaining fixed. The one parameter which was not at our disposal to vary was the vibrational frequency (fixed at 20 KHz). Thus we are unable to comment on the possible uniqueness of the optimum gap-vibration frequency relationship.

Mr. M.J. Neale (Michael Neale and Associates). One point relating to the practical application of the results is that the most severe form of bearing cavitation damage in terms of its effect on engine reliability is that which occurs around the groove in the lower half of main bearings. It is the side (or cross section) of the bearing lining that is exposed in that case and with some alloys such as Al-Sn etc., there is a a layer of bonding material beneath the lining which may be a lot softer. In the case of Al-Sn it is the cavitation erosion resistance of this pure aluminium layer which determines the resistance to erosion of the bearing in practice.

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Mr. D. Landheer (Eindhoven University of Technology, The Netherlands). The work presented here is most interesting both in giving an experimental analysis of the friction and wear behavior of hydrodynamic bearings containing abrasive contaminants and in working out a model for the partial abrasive embedding phenomenon. Closer examination of the first two references given in the synopsis turns out to be quite thought provoking. Let me restrict myself to one or two points:

1. What fraction (α) of the total bearing load F_n is transmitted by the action of the abrasive particles? It was reported that the bearing friction torque M increased from 0,1÷0.2 Nm to 0.7÷0.9 Nm on the switching from clean to contaminated oil (ref. 2). Assuming that the change in hydrodynamic friction is relatively small the increase $\Delta M/r = (0.5\div0.7)/0.027 = 18{,}5\div 27$ N (r= radius of the journal). Taking a coefficient of abrasive friction of about $f_a = 0.3$ (cf. sandpaper; loose rolling particles would tend to lower f_a) a normal force $F_{na} = F_{ta}/f_a = 62\div86$ N is estimated for the load transmitted by the particles. With the total bearing load of 250 N it follows that $\alpha = F_{na}/F_n = 0.25\div0.35$.

2. A "semi-classic" description of the volumetric wear V of a sliding member under conditions of dry 3-body abrasive contact is (4)

$$V = (1/3).\ k,\ \frac{F_a}{H_i}\ .\ s \qquad (1)$$

in which: k = wear coefficeint

F = normal load upon the system

H_i= indentation hardness of the surface considered.

s = sliding distance of the surfaces over each other

Typical values of k are quoted as $(2\div6).10^{-3}$ (ref. 4). If eq. (1) is to be used for an estimation of the abrasive wear in the present case of a lubricated bearing F should be replaced by the load F_{na} which is actually transmitted by the abrasive particles, as derived under 1. The sliding distance in the teststages 3 and 4, in which the abrasive is in operation (as indicated by the measured friction), is about 4000 m. For the hardened shafts (H_s = 6900 N/mm^2) eq. (1) thus becomes

$$V = (1/3).\ (2\div6).10^{-3}\ \frac{75}{6900}.\ 4.10^6 = 29\div87\ \text{mm}^3$$

In a corresponding way it is found for bearing liners with $H_1 \approx 1000$ N/mm^2: V = 200÷600 mm^3.

These figures are of the same order of magnitude as or somewhat higher than those reported in (2).

3. The combination of running surfaces with strongly differing hardness values is usually considered as a suitable measure to reduce abrasive damage to the hardest member. The present experiments show that a beneficial (that is counter surface protecting) embedding of the particles does not occur in a wide range of hardness ratios and is replaced by some kind of lapping action.

In (2) it was suggested that the deviation between "classically expected" and observed behaviour may be explained by the dominant contribution of the oil film in the load carrying action. Now, being shown that the load on the abrasive particles is appreciable in the present case (see under 1) and that that the wear rate is of the order of magnitude of that estimated by means of eq. (1), and considering that the minimum oil film thickness of 6 µm is rather small compared with the linear particle dimensions (40 µm). could it possibly be that the general concept of useful embeddability requires some revision?

4. The criterion for partial embedding of the abrasive particles was checked by the experimentally determined film-thickness at the places of embedding marks on the liner for one single operating condition. The model presented implies that in the same bearing, but with minimal film thickness values in excess of h_{cr} = 12 µm, no embedding would occur. Can direct evidence be given for this?

References

(1) Ronen, A., Malkin, S. and Loewy, K. "Wear of dynamically loaded hydrodynamic bearings by contaminant particles", Trans. ASME, J. Lub. Tech., 1980, 102. 452.

(2) Ronen, A., and Malkin, S., "Wear Mechanisms of statically loaded hydrodynamic bearings by contaminant abrasive particles",

Wear, 1981, 68, 371.

(4) Raninowicz, E., "Friction and wear of materials", Wiley, New York, 1965.

Reply by Dr. A. Ronen and Professor S. Malkin (Technion, Haifa, Israel). The authors appreciate Mr. Landheer's interest in this paper and having taken the time to carefully read our other papers on this subject.

In response to the first point, we followed the same procedure and arrived at a somewhat smaller fraction α due to our assumption of a larger friction coefficient $f_a = 0.5 - 1.0$. What was of particular concern to us is whether the presence of the contaminant particles affects the minimum oil film thickness. Experimentally, we found that switching while running from clean to contaminated oil, and vice versa, had virtually no effect on the oil film thickness.

One of the main points in our paper is that wear by contaminant abrasive particles does not follow the classic abrasive wear behaviour. While the wear results for shafts and liners may be consistent with the classic wear model in some cases, the influence of hardness is contrary to what might be expected. From the results in Fig. 4, it can be seen that increased hardness of either the shaft or liner results in more wear of that component. These anomolous results are explained by the relative tendency of an abrasive particle to roll or partially embed in the liner.

Regarding the remaining two points raised by Mr. Landheer, we can only reply that the abrasive model was derived in a general way based upon experimental results obtained over a certain range of test conditions. Additional experimental results would be necessary to check the validity of the model over a wider range of test conditions.

Dr. O.R. Lang (Daimler-Benz A.G.). It should be pointed out very clearly that the results are restricted to the materials of shaft, liner and particles under test. This must be especially borne in mind using the formula. Did you not feel for instance that the missing particle hardness in your proposed formula is a very important thing for practice.

Reply by Dr. A. Ronen and Professor S. Malkin (Technion, Haifa, Israel). While our wear model is derived in a general way, additional experimental results would be necessary to validate its applicability outside the range of our experimental test conditions. As in most descriptions of wear by abrasives, the abrasive hardness does not explicitly enter into the model. It is tacitly assumed that the abrasive is much harder than the wearing material, which is usually the case. When the abrasive is not more than about 20% harder than the wearing material, the abrasive hardness is found to have a significant effect, and we might expect analogous effects in our case. Such a situation could arise with silica sand abrasive and a hardened stell shaft material. Additional experiments are planned to explore this "soft abrasive" regime.