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DISCUSSION

A. Mukherjee[2]

The authors have attempted to analyze a problem of considerable significance. Since early fifties attempts were made to evaluate the influence of solid particle contaminated lubricants on wear rate, thermal performance and dynamics of rotor bearing systems. So far only two categories of contaminations were mainly studied. The first category dealt with lubricants loaded with extremely small size dirt particles and as first approximation micro-polar fluid models were used. The second category included dispersions with somewhat larger size particles at lower concentrations, and the analyses were approached through liquid-solid biphase models. Lubricants with solid contaminations specially synthesized to control bearing performance often belong to this category. The authors, however, did not refer to any previous work related to this category.

The situation considered by the authors differs from both aforementioned categories. The concentration here is extremely low and particles are not only large but also highly aspected. This makes a third category of problems. No continuum model is adequate for this class as only a few particles can occupy the clearance space at a time. The authors deserve compliments for initiating an organized course of studies on this category.

However, the use of over-simplified model by the authors for analyzing the problem and further simplifications made in the illustrative examples render their conclusions somewhat restricted in significance and validity. Following are a few observation which the discusser would like to make in this regard.

The phenomenon of large and abrupt fall in fluid pressure is a consequence of flow blockage due to a large size particle covering the entire bearing width. In a real situation a particle covers only a part of the bearing width. There are margins on either side of such a particle which allow fluid-film to negotiate both upstream and downstream. The discusser believes that due to these films the jump in the fluid pressure will be considerably reduced. In this regard, the discusser feels a different scheme of modeling would have been closer to real situation. The fluid-film spanned by the particle and the margins could have been analysed using short bearing approach. This would have made the influence of relative width discernible. The discusser would like to know, whether any attempt on these lines was at all made by the authors or whether they dropped it altogether due to model inadequacy and/or analytical complexities.

The authors have presented the pressure distribution for arbitrarily prescribed particle coordinates and velocities. The discusser feels in a real situation a particle will be convected by the fluid (in a broader sense of the term). The authors should have outlined a procedure to establish the equilibrium of a particle at a given position and the velocity it might attain there. This can be done, even in the framework of present analysis by integrating the fluid stresses on both the faces of a particle. The angular velocity of a particle about its centroidal axis though appears in the general analysis, no reference of it has been made in the illustrative examples (it is presumed to be zero!) The discusser feels that this angular velocity is an important dynamic variable, specially in case of a high aspect ratio particle. Instability in this dynamic coordinate may lead to nearly total blockage of flow or a coughing phenomenon. In the discusser's opinion an estimation of angular stiffness and damping coefficient, even about an aligned configuration, could have provided a significant clue to the question of rotational instability.

A. E. Yousif[3]

Dr. Languirand and Professor Tichy are to be congratulated on their contribution to the fundamental understanding of the effect of presence of an elongated particle in full hydrodynamic lubricating films. Their study is not only important to the understanding of the behavior of lubricating oils containing additivies and contaminants but it may also help to understand the behavior of worked greases in mechanical bearings. Although their contribution, at present, might seem merely an academic excerise but it will certainly aid to the future development of lubricating oils where the best type, size, shape and distribution of solid additives may be achieved in the ultimate aim of improving the mechanical bearing performance.

In some previous rheological work [25] on synthetic greases we showed that non-newtonian viscoelastic effects take place even at very low percentage concentration of solid particles in a newtonian oil. We have also shown that soap base greases [26, 27, 28] suffer mechanical degradation in heavily loaded contacts so that their virgin three dimensional fiberous structure broke down to almost spheriodal particles in the medium of the carrier oil. The mechancial degradation was mainly responsible for the initial reduction and increase in

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grease film thickness and traction respectively with time from the instant of feeding a fresh charge of grease to the contact zone.

I do agree with Dr. Languirand and Professor Tichy that the micro-polar fluid theory as used for lubricating oils with additives and contaminants is not the best mathematical model to fully appreciate the behavior of lubricating oils containing additives and contaminants although we have recently shown (29) that the bearing performance is generally improved with such fluids. In some recent work (30) we considered the lubrication of plain slider bearing taking into account the fact that the volume concentration of particles is governed by the equation of mass transfer we found that both the load capacity and friction force increase with increasing concentration additive and containinant particles.

In the past we tried to insert a sheet of nylon in the gap of a plain slider bearing apparatus and recorded an increase in the pressure distribution but the sheet wrinkled. Therefore, I would like to ask the authors how did they make sure that their metalic sheets were carried in the oil film and not by the belt of their apparatus, and if it was possible for them at all to work out the effect of presence of their particle in the oil on the overall performance of the bearing (31) as this should be the ultimate aim of the excersise? I would also like to question the shape of their pressure distribution curves which seem rather interesting since they deviate from the usual curves in a slider bearing. I would have thought that smooth continuous bulges in the pressure curves would result as the particle travels along the bearing in the absence of cavitation off-course. Could this be due to the pertubation series or perhaps due to the simplifying assumptions employed in their analysis?

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Authors' Closure

The authors would like to express their appreciation to Professor Mukherjee and Professor Yousif for their comments and interest in this paper.

The authors agree with Professor Mukherjee that the magnitude of the pressure jump would decrease for a particle covering only a part of the bearing width. Also that the particle angular velocity is an important dynamic variable. Possibly future work in this area will demonstrate the significance of these effects. Pertaining to the question of a short bearing approach, the method was not considered.

In reference to Professor Yousif's question on partaicle stiffness, the "particle" used for the experiment was a 1/32 inch thick stainless steel sheet. Based on the magnitude of the pressure field the particle deflection would be small. However, it was not possible to observe the position of the particle when traveling through the bearing gap. Professor Yousif also comments on the sudden changes in the pressure curves. It is anticipated by the authors that a smoothing of these curves would result if the perturbation series were carried out further. This was attempted however analytical complexities prevented the solution of higher order terms.

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