Discussion on Paper by O.R. Lang: "Oil film rupture under dynamic load? Reynolds statement and modern science"

squeeze effect will be much larger, and consequently, the damping will be higher (the film thickness is much slower in changing to another value).

The question of vibrations in a set of involvute gear wheels has been studied before. I believe that in 1981 Wang and Cheng published two papers on this subject in ASME's Journal of Applied Mechanics. Their analysis includes Vichard's approach for the fluid film behaviour, and it was combined with a Finite Element analysis for the bending and torsion in the gears.

Reply by Dr A.K. Tieu and Mr J. Worden (University of Wollongong, Australia).

We thank Dr van Leeuwen for his comments. In his paper, Vichard correctly assumed that at the centre of contact, the reduced pressure Q approaches its asymptotic value 1/G, the inverse of pressure-viscosity coefficient. From an inspection of the expression for Q under transient condition, Q is not constant in the area of contact, but it changes from inlet to the centre. For steady-state case, an analysis in 'Principles of Lubrication' (A Cameron, Longmans 1966) shows that the difference of the reduced pressures Q at the inlet and at some distance inside the contact zone is small, but not zero as implied by the discusser. If the integration is carried out at inlet, then appropriate boundary condition of Q (different from 1/G) should be used.

The ASME paper by Wang and Cheng came from Ref (11) in the paper. In their paper, the dimensionless load includes the gear root radius, whereas here the relative radius of curvature of the gear pair was used as per Ref (1), which results in higher applied loads, and more significant squeeze film damping. This paper considers in isolation the effects of different dynamic loads on the gear film thickness, and in particularly it can apply to those loads resulting from gear tooth chattering with excessive backlash. The frequency and level of impact loads were not considered in Ref (11). An aim here is to arrive at a simple criteria based on the parameters X_p, X_e, which indicates readily without an involved analysis, whether arbitrary applied loads would have any significant influence on the film thickness.

SESSION XII - BIO-TRIBOLOGY

'An Analysis of Micro-elastohydrodynamic Lubrication in Synovial Joints under Conditions of Cyclic Loading and Entraining Velocities'

D. DOWSON and Z.M. JIN.

Professor T Murakami (Kyushu University, Japan). Your application of micro-elastohydrodynamic lubrication theory to natural synovial joints provides strong support for fluid film lubrication in natural joints under walking condition. (1) Could you show some data on the effective film parameter which is defined as the ratio of minimum film thickness to effective roughness after deformation? (2) Have you investigated the influence of highly concentrated gel formed on articular cartilages in the concave area on the effective film parameter under thin film conditions?

Reply by Professor D. Dowson and Mr Z.M. Jin (Institute of Tribology, Department of Mechanical Engineering, The University of Leeds).

We are grateful to Professor Murakami for his interest in our paper on the micro-elasto-hydrodynamic lubrication of synovial joints. The effective film parameter (Λ) is normally defined as the ratio of the minimum film thickness to the composite roughness average (Σ Ra) for the two bearing solids. In micro-elasto-hydrodynamic lubrication, the effective composite roughness average varies throughout the conjunction and, in the case of the dynamic conditions considered in the present paper, with time. However, in general terms we found the effective film parameter (Λ) to be in excess of 10 during the periods of each cycle governed by entraining motion. When squeeze-film action dominated the situation the surface roughness partially recovered its initial form and the effective film parameter fell to about 2. We would stress, however, that it is the absolute value of minimum film thickness, rather than the film thickness ratio alone, that is important under these circumstances.

Professor Murakami's second point refers to the possible role of enhanced viscosity and hence film thickness associated with the gel formation on cartilage surfaces. He no doubt has in mind the earlier indications of the mechanism of boosted lubrication (D1) and this is incorporated in some of the analysis presented in the paper. We find that it is necessary to invoke the 'surface viscosity' concept in order to predict realistic values of the coefficient of friction.

The powerful role of boosted lubrication is evident in the preliminary results presented in the paper. We are now undertaking a more detailed study of the combined role of micro-elastohydrodynamic and boosted lubrication actions in synovial joints. Our findings indicate that boosted lubrication leads to substantial improvements in the predicted film thickness and theoretical coefficients of friction which are still small, but just within the range of experimental results recorded in the literature.

References

D[1] Dowson, D., Unsworth, A. and Wright, V. (1970), "Analysis of 'Boosted Lubrication' in Human Joints", J. Mech. Engrg. Sci., Vol. 12, No. 5, pp 364-369.

SESSION XV - BEARING DYNAMICS (2)

'Oil Film Rupture Under Dynamic Load? Reynolds Statement and Modern Experience'

O.R. LANG

Mr H.J. van Leeuwen (Eindhoven University of Technology, The Netherlands).

The experiments carried out at Karlsruhe University on dynamically loaded bearings can be very helpful in testing different approaches for calculations of journal orbits, film pressures, etc. I understand that these experiments confirm the results of the Holland-Lang method.

I would like to know whether the Holland-Lang method makes use of different boundary conditions compared to the Holland method, as it was published in 1959. Are there any other differences?

If the boundary conditions are Reynolds type, viz. $\nabla p=o$, $p=o$ at the outlet, for both pressure components p_d (pure wedging action) and p_v (central pure squeezing action), a simple superposition is no longer allowed because this boundary condition is not linear. The Reynolds equation is linear in p, however, and therefore methods like the mobility method or the impulse-whirl angle method are correct from a mathematical viewpoint: they fulfil the boundary condition at the outlet because only one free boundary exists (for the pure squeezing action). Two free boundaries, as arise in the Holland approach, will never coincide, and hence the problem is overspecified.

For example, at the outlet, where $\nabla p=p=o$, it is only valid to state that $\nabla p_d = -\nabla p_v$ and $p_d = -p_v$. Requiring $\nabla p_d = p_d = o$ and $\nabla p_v = p_v = o$ will lead to a smooth pressure profile, but also to results which are in error, expecially at high loads. Therefore, the Holland approach of 1959 will lead to erroneous results.

Reply by Dr.-Ing. O.R. Lang (Daimler-Benz AG, West Germany).

The 1959 solution of Holland used an analytical solution for pure wedge and pure squeeze for an infinitely long bearing, approximated to finite length by the assumption of a parabolic pressure distribution. The Holland-Lang solution uses numerical solutions for pure wedge and pure squeeze for finite length with Reynolds' boundary conditions for both independently with high accuracy by a refined mesh, especially in the region of high pressures and the outlet. This is presented in the doctoral thesis of Butenschon, University of Karlsruhe 1976.

The formulation of Reynolds' boundary conditions in terms of pressures $p = o$ and $\Delta p = o$ is not the primary physical one, this is continuity within the total pressure development. Circumferential flow and the two different types of velocity profiles develop independently as long as the gap is filled. Dynamically loaded bearings are overfeed and the oil inlet is chosen according to the state of the art. The point is not a pure mathematical correct solution, but a physical one, concerned with pressure development, which is given in Holland-Lang as well as in the mobility method or impulse-whirl angle method.

An alternative physical theorem is that of minimum potential energy, which Swift used to assure Reynolds' boundary conditions for wedge action. Minimum potential energy means highest load capacity and minimum friction. Theoretical superposition of combined wedge and squeeze with Reynolds' boundary conditions yields significantly smaller pressures, lower load capacity and higher friction than the Holland-Lang superposition. The experiments at Karlsruhe University confirm this in displacement and pressure extent. Moreover, nearly twenty years of practical experience in the field of dynamic loading gives much more security than a one-sided mathematical viewpoint, which is not the unique one. There is no mathematical rule against superimposing independent solutions of a linear differential equation, as long as the physical boundary conditions are fulfilled. This is common in vibration, thermodynamics or stress-strain calculations.

SESSION XIX - MACHINE ELEMENTS (1) - OIL RING BEARINGS

'Performance Characteristics of the Oil Ring Lubricator. An Experimental Study'

K.R. BROCKWELL and D. KLEINBUB

Emeritus Professor H. Blok (The Netherlands)

The authors' Figure 10 shows their success in deriving a fairly strict non-dimensionalized correlation in terms of only two dimensionless groups, i.e. the two Reynolds numbers, Re_{ring} and Re_{shaft}. However, judging from other figures that have not been non-dimensionalized so far, the use of merely two such dimensionless groups cannot very well yield an equally small correlational scatter. The reason may well be sought in the fact that in ring-oiling at least a few more physically influential quantities have in general to be accounted for than the five represented in the two Reynolds numbers. These five quantities are: the circumferential speeds of the ring and the shaft, V_r and V_s, the viscosity and density of the oil, η and ρ, and some "scaling" length, L_{sc}, for which the discusser suggests the diameter of the shaft.

So, as to the dynamics of the ring the authors have considered only the viscous and the inertial forces. However, in ring-oiling three more kinds of forces will have to be accounted for, that is, the gravitational and the surface-tensional ones as they occur in the various oil flows, and the load imposed by the weight of the ring on its entraining oil film. These forces can be represented by introducing three more physically influential quantities, that is, the gravitational acceleration, g, the surface tension, σ, and the unit load, W, as referred to the unit width of the rubbing surface.

Now, the eight physically influential qualtities thus introduced in total, will yield a complete set of three more, that is five dimensionless groups, N_1 through N_5. True, the authors' two Reynolds numbers could be included therein but the discusser prefers the following set from which, after all, their two Reynolds numbers can be reconstructed, if so desired.

$$N_1=(\rho/\sigma g)^{1/4}.V_r;\ N_2=(\rho/\sigma g)^{1/4}.V_s;\ N_3=(g/\sigma^3\rho)^{1/4}.\eta; \quad (1a,b,c)$$

$$N_4=\sigma^{-1}.W, \text{ and } N_5=(\rho g/\sigma)^{1/2}.L_{sc} \quad (1d,e)$$