MODELLING THE WEAR PROCESS IN COMPOSITE LINER BEARINGS

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KEYWORDS
Liner bearings, solid lubricant, wear modelling.

ABSTRACT

This paper presents a numerical method for modelling wear in spherical composite liner bearings. These are bearings that have two rings with spherical surfaces to form a ball joint. The concave spherical surface of the outer ring is lined with a thin composite liner material which is a made of a woven fabric impregnated with a thermoset resin. Two fibre types are used in the fabric, a structural fibre and a solid lubricant fibre. The liner is made up of two fabric layers impregnated with resin that are bonded together between heated pressure plates. The surface layer has a high proportion of the solid lubricant fibre in its weave whilst the backing layer has a predominance of structural fibres with fewer lubricant fibres. These bearings are used extensively in reciprocating motion applications in the aerospace industry.

The life of the composite liner bearing is determined by the wear occurring in the liner which leads to increasing backlash with use. The effective life limit is reached when the face layer is worn away and contact with the backing layer occurs with a significant resulting increase in friction and wear rate. The analysis method is based on a contact model where the liner is considered to develop a compressive stress in accordance with the normal strain caused by the eccentricity of the steel spherical rings. The eccentricity is calculated such that the liner pressure acting on the inner ring surface is in equilibrium with the load applied to the ring. An Archard wear model [1], $w' = kpu_s$ is used to calculate the liner wear, where w' is liner thickness loss rate, p is the liner pressure, u_s is the sliding velocity and k is the wear coefficient which is deduced from experimental tests. Figure 1 shows the development of the liner contact in a reciprocating test case where the load is fixed in magnitude and direction relative to the outer ring (and thus to the liner). The plots cover the load bearing area which extends over the full axial length, z_{max}, and different circumferential angle ranges as the wear develops. The initial contact has a diametric clearance of 50 μm which leads to a contact patch having a maximum pressure of 91 MPa. The evolution of wear in the liner is calculated in a sequence of wear steps where the pressure and wear rate is assumed to be unchanged so that the equation gives the change of liner thickness for the wear step. Wear steps with 1 μm maximum material removal were used for the analysis. Over 20 such steps the wear pattern redistributes the load to cover a larger contact area with a reduced maximum pressure of 40 MPa. This continues to the final calculated stage with maximum wear and pressure of 150 μm and 34.7 MPa, respectively.

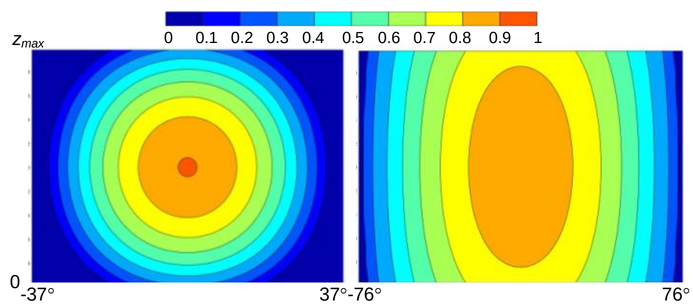

Figure 1. Contours of (a) p/100 MPa for first wear step and (b) p/40 MPa for last wear step.

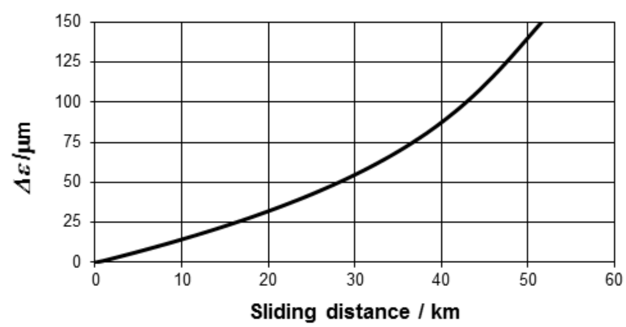

Figure 2. Best fit curve to the experimental test data.

The wear model was used to calculate the wear occurring in a bearing test where the change in dynamically measured eccentricity $\Delta\varepsilon$ with sliding distance takes the form shown in figure 2. It was found that the experimental results could be replicated in the model by adopting a wear factor that increases with wear depth in the form of a power law.

REFERENCES

[1] Archard, J F, Contact and rubbing of flat surfaces. Journal of Applied Physics, 24, 981-988, 1953.