NUMERICAL MODELLING OF A GREASE LUBRICATED PNEUMATIC SEAL

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ABSTRACT

This work addresses the numerical modeling of grease-lubricated pneumatic sealing systems. An EHL model have been developed by coupling a non-linear commercial software (Abaqus) with an unsteady Reynolds equation. Mixed lubrication and non-Newtonian rheological behavior of the lubricant are taken into account. Comparison with experimental measurements are made for two different piston velocities and pneumatic pressures. Starved lubrication conditions seems to be a key aspect in obtaining good friction force predictions.

MODEL DESCRIPTION

The numerical model used in this study has already been presented in reference [1]. It consists in creating a new element in Abaqus, called Reynolds User Element (RUE) that deals with the interface between the seal and the cylinder. The RUE solve the Reynolds equation by taking into account mixed lubrication conditions, mass conserving and grease rheology. A U-Cup 100 mm in diameter classical piston seal is investigated. The seal is tested experimentally on a specific test bench and the friction force has been determined as a function of piston velocity and pneumatic pressure [2]. It has been observed that the friction increases with both the velocity and the pneumatic pressure. In addition, the friction force values are of the same order in both directions of movement (outstroke and instroke).

NUMERICAL RESULTS

Visualizations made with a transparent cylinder show that, after one outstroke/instroke cycle, the contact zone is only supplied by a thin lubricant layer deposed on the cylinder. Therefore, it seems incorrect to impose, as a boundary conditions, a fully supplied lubricated contact. Indeed, the first simulations showed that if starvation lubrication conditions are not taken into account, the numerically predicted friction force is much lower than that measured experimentally. More importantly, great differences are predicted between the instroke and the outstroke movements. Also, the overall friction decreases with increasing piston velocity, which contradicts the experimental measurements.

A second series of simulations was performed by imposing a given quantity of lubricant present in the contact inlet. In this case, the numerical simulations predict an increase in friction force as a function of piston velocity and similar friction values during instroke and outstroke movements (Fig. 1). The absolute values of the friction forces are directly influenced by three numerical parameters: the effective film thickness applied as a boundary condition in the inlet zone, the lubricant apparent viscosity and the friction coefficient used to compute the tangential shear stress generated by the asperity contact pressure. Unfortunately, these parameters have not been measured experimentally. The friction coefficient was considered to be 0.2, a value already used in previous work for elastomer/steel contacts. Different sub-micrometer values for the effective film thickness have been tested and their influence on the friction force seems to be very important. Also, even if the rheological behavior of the grease has been measured on a conventional rheometer, the film thickness predicted numerically in the contact zone is two orders of magnitude smaller than the film thickness used to identify the grease rheological behavior. It is than assumed that the apparent viscosity of the lubricating fluid is close to the base oil of the grease.

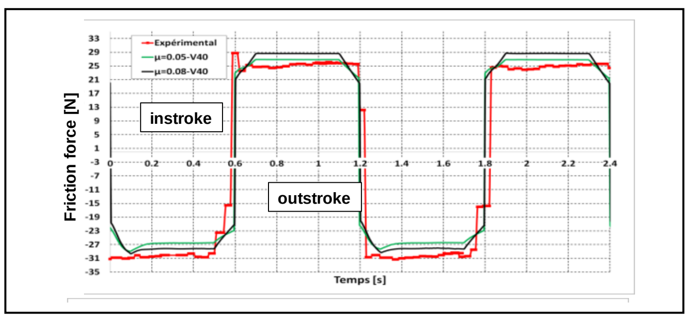

Fig.1 Variation in friction during two instroke/outstroke cycles at 40 mm/s

REFERENCES

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