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FRETTING WEAR OF COATED SURFACES UNDER GROSS SLIP CONDITIONS

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ABSTRACT

Blade-disk contact within high pressure compressor, is a high temperature fretting contact zone submitted to high stress levels. Therefore, it is important to estimate the various damages encountered in this area in order to correctly predict the lifetime of the parts. Coatings are used in order to improve the strength of the parts. In the framework of Innolub project, this study aims at developing a numerical model of coated contact accounting for wear and temperature. The objective is to estimate the kinetics of wear, to simulate worn surfaces and to allow the manufacturer to produce analyses of design with worn surfaces using a fast, efficient and reliable method of calculation.

Keyword: Coating, High temperatures, Fretting, Wear, Energy approach, Gross slip

INTRODUCTION

The aerospace is a sector of activities where many challenging problems arise. The area of contact between the blade and disk in turbo-machinery has been identified as a critical area for the nucleation of damage for the safety of the flying machine. To control damage, coatings are applied. There is a great need for innovative coatings given good lubricating properties under high temperatures conditions and limited wear of the parts. The prediction of wear is a challenging task. Many formalizations have been proposed for quantitative approach to wear. More than 300

have been identified by Meng and Ludema [1], resting on different applications [2–5]. The energy theory of wear, inspired by the Archard's theory, is selected as it is very simple to implement. This work aims at developing a numerical model of coated contact accounting for wear and temperature. The objective is to perform wear prediction using semi analytical computational methods. This research is based on previous works [6–8] developed at LaMCoS. Gallego et al. [9] proposed an elastic contact model under stick-slip conditions. This model has been extended to the phenomenon of fretting wear [10] and under partial slip conditions [11]. Analytical solutions to take into account the plasticity in the resolution of the tangential contact have been developed [12]. The presence of heterogeneities within the contacting bodies has been later introduced and the model applied to the elastic-plastic contact under fretting wear conditions [13]. Finally, Leroux [14] extended the formulation and proposed a model to account for the presence of multiple spherical inclusions based on the equivalent inclusion method originally proposed by Eshelby. The first part of this paper focuses on coating. The coating is modeled as a cluster of inclusions based on the Eshelby's theory. Validation is obtained by comparison with multigrid results. In the second part, wear simulation is carried out illustrating the model robustness and efficiency. Results are then presented and briefly discussed.

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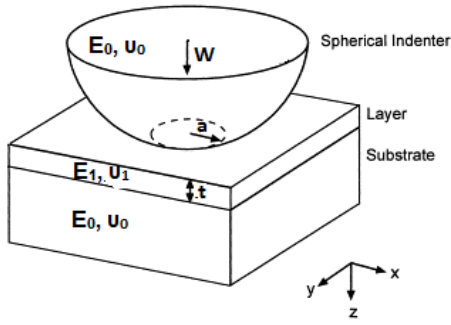


Figure 1: 3D model of sphere/coated plane elastic contact.

1 Coating: Modeling and Validation

1.1 Modeling

A rigid indenter is considered, with a spherical tip of radius R . It is pressed against an elastic flat of properties ($E_2 = 210\text{GPa}$, $\nu_2 = 0.3$) covered by a layer (E_1, ν_1) - more or less rigid than the substrate - and of thickness ' t ', see Fig. 1. ' W ' is the normal load and ' a ' the contact radius. The equivalent inclusion method in the sense of Eshelby allows to describe accurately the effect of inhomogeneities. This method is modified and improved in order to take into account the mutual influence between neighboring heterogeneous inclusions. A few analytical solutions have been obtained for inclusions of simple geometry embedded in an infinite isotropic medium, such as ellipsoidal [15], cuboidal [16] and cylindrical [17, 18] inclusions. The aim of this part is to model an isotropic heterogeneous coated contact, based on [14, 19], using an arrangement of cuboidal inclusions to describe the layer. The Eshelby's equivalent inclusion method is considered to quantify the surrounding material stresses. An iterative process is implemented to determine the displacements and stress fields caused by the eigenstrains of all inclusions, accounting for interactions between close inclusions. The proposed method can be seen as an enrichment technique for which the effect of heterogeneous inclusions is superimposed on the homogeneous solution. 3D and 2D Fast Fourier Transforms are utilized to improve the computational efficiency. Displacements and pressure relationships account for this contribution. For more details on the method and algorithm the reader may refer to [14].

1.2 Validation

The contact solution and internal stress field obtained for the 3D coated model are compared with results obtained using a 3D multigrid solver [20]. A sphere is pressed against a coated half space. The coating elastic constants are: $E_1 = 420\text{GPa} = 2 * E_2$, $\nu_1 = 0.3$ and $t=0.75\mu\text{m}=0.25*a$. The normal load is kept constant $W=0.04\text{N}$. Figure 2 presents the distribution of pressure obtained using a 3D multigrid solver and the semi-analytical solver.

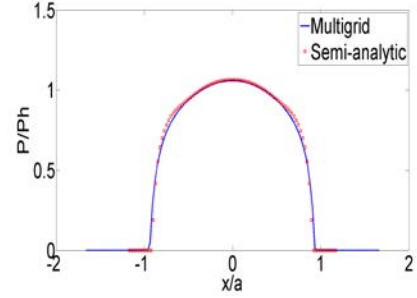


Figure 2: Distribution of pressure: fine mesh ($dx=0.15\mu\text{m}$)

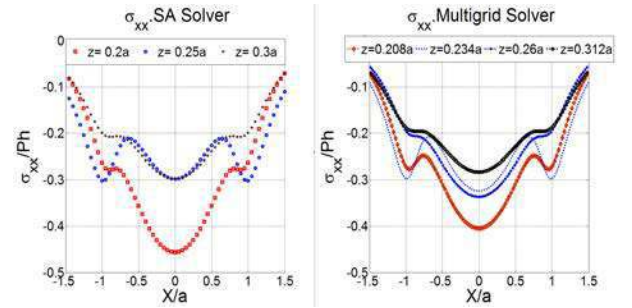


Figure 3: Distribution of stress σ_{xx} -for a thickness $t=0.25a$ - at different depths z : fine mesh ($dx=0.05\mu\text{m}=0.05a$)

σ_{xx} within the layer and the substrate are plotted in Fig. 3 for a mesh size of $0.15\mu\text{m}$. The depth at which the stress is calculated is different, which explains the difference here.

2 Wear: Methodology and simulation

2.1 Methodology

The tribological analysis of a contact can be understood through the energetic balance. Friction is a dissipated phenomenon, and a part of this frictional dissipated energy is consumed by different processes that conduct to wear [21]. To quantify the wear of a sliding contact, Archard [22] published in 1953 his famous equation: $V=\alpha_{\text{ARCHARD}} \cdot s \cdot \frac{P}{K}$, where α_{ARCHARD} is the wear coefficient. The worn volume V is a function of the sliding distance s , the normal applied load P , and the hardness of the softer material K . The friction coefficient does not appear in this equation. However it is often observed that the greater the friction coefficient the larger the wear. In addition, the friction dissipated energy is equal to the work of the tangential load. For these reasons Fouvry and co-workers [2-4, 23, 24] developed a modified Archard's wear law: $V=\alpha \cdot \sum_N E_d$, with E_d the frictional dissipated energy density. A method to simulate wear of contacting surfaces is proposed. A wear law based on the energy dissipated by friction, derived from the work of Fouvry, is used. The local sliding amplitude is assumed constant along the

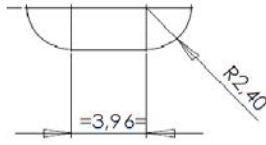


Figure 4: Profile of the punch [mm] in (x,z) plane

wear cycles. Wear can concern only one of the contacting bodies, or both. In addition, the worn material is assumed in the numerical procedure to be removed at the end of each cyclic movement only. The input data of the problem are:

- P^k , $k=1, \dots, N$: the normal load history of each cycle
- s^k , $k=1, \dots, N$: the corresponding local sliding amplitude
- δ_0 : the sliding amplitude
- ΔN : the number of cycles to update the worn geometry
- N_{max} : the maximum number of cycles
- H_{max} : the maximum value for the wear depth to end the computation

For each cycle, E_d is computed at each point. It corresponds to the product of the tangential stress with the sliding distance; thus, in gross slip regime: $E_d = \sum_{k=1}^N \mu x p^k x s^k$.

The wear depth Δh for one cycle is calculated at each point with an appropriate wear law $\Delta h = f(E_d) = \alpha \cdot S \cdot \sum_N \delta_0(N) E_d(N)$. The value is then multiplied by ΔN , and the geometry is updated before next contact solution and wear calculation. The corresponding wear of each surface is taken into account according to a wear partitioning coefficient WP, equal to 0.5 in the case of an equal wear repartition for each body. The simulation can be stopped through two ways, one when the number of cycles N exceeds the maximum number of cycles N_{max} , the other when a maximum value of a wear depth is reached. Finally the computation gives the final geometry and the wear distribution.

2.2 Simulation of a Fretting Wear Test

The numerical simulations presented here correspond to experimental fretting tests [25]. It consists of fretting tests with a Ti6242 punch/Ti17 plan. The indenter (Ti6242 punch) is considered as a plan with crown radius. The geometry of this punch is shown in Fig. 4. Materials properties are ($E_0 = 114GPa$, $\nu_0 = 0.36$) and ($E_2 = 110GPa$, $\nu_2 = 0.34$). A slip amplitude $\delta_0 = 100\mu m$ is considered. The normal load W and the friction coefficient μ are assumed constant all over the cycles and equal to 1150N and 0.68, respectively.

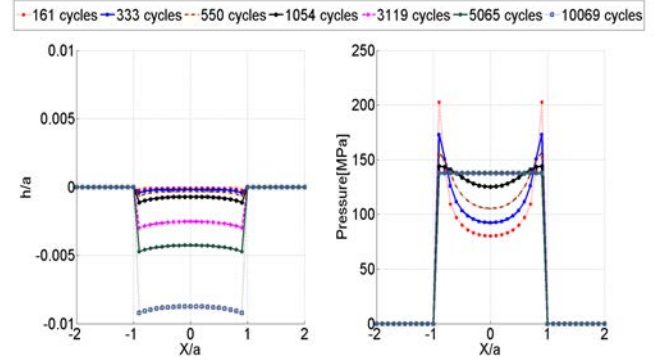


Figure 5: Evolution of wear: pressure and depth

3 Results and discussions

For industrial application, the most important element is the lifetime of the disk, which can not be repaired, and whose failure is the most critical. In contrast, the blade root can be repaired, and breaking of the blade can be contained within the engine. Thus, the main objective is to develop a sacrificial deposit which wears out preferentially but which will avoid disk wear (Ti17). Before studying the response of deposit, we aim to characterize wear for uncoated contact to quantify the gain provided by the deposition.

3.1 Uncoated contact

A first simulation was carried out to analyze the wear response of uncoated contact Ti6242g/Ti17g. The simulation were conducted by imposing the sliding amplitude $\delta_0 = 100\mu m$. The normal load was chosen equal to 1150N, the friction coefficient to 0.68 and the wear coefficient α to $3e^{-5} mm^3/J$. 10 000 cycles were realized. Wear is assumed to affect only the plan (WP=0). The evolution of wear is detailed in the graphs of Fig. 5 in terms of contact pressure history and wear depth history. It can be observed that the pressure distribution has a profile similar to the surface of the punch for the first cycles and after it evolves naturally through a flat profile, i.e., with a uniform increment of wear inside the contact area as the wear law used in our case is proportional to the contact pressure. The area where the wear is maximum is located at the edge of the punch because of the peaks of pressure produced by the geometrical discontinuity. The resulting worn geometry of the plane after $N_{max}(=10\ 000)$ wear cycles is given in Fig. 6. These data characterize the uncoated interface, they will allow us to quantify the gain provided by the coating studied in the next section.

3.2 Coated contact

The coating was supposed two times harder than the matrix (the plan): $E_1 = 2 * E_2 = 220GPa$ with a Poisson's ratio $\nu_1=0.3$.

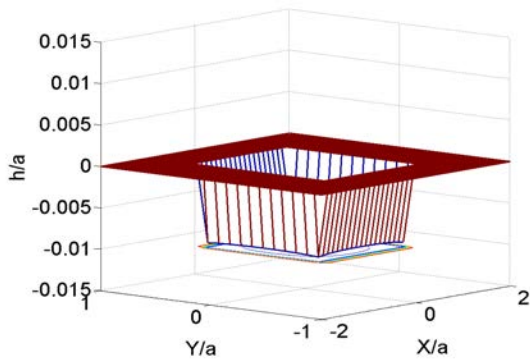


Figure 6: Worn geometry after $N_{MAX} = 10000$ cycles

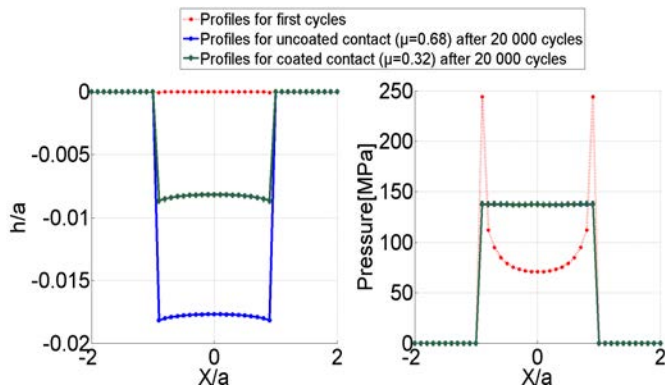


Figure 7: Comparison of coated and uncoated contacts after $N_{MAX} = 20000$ cycles

It is characterized by a thickness $t = 400 \mu\text{m}$. The coefficient of wear is assumed identical. Simulation loading conditions considered are $W=1150\text{N}$ and $\mu = 0.32$. Wear is assumed to affect the plan and the punch ($WP=0.5$). The simulation is carried out for 20,000 cycles. Comparison between coated and uncoated results are shown in Fig. 7. It is observed that the wear depth for the coated case is reduced by more than half compared to the uncoated case. This result tends to confirm the role of sacrificial coating. The distribution of the shear stress component σ_{xz} is plotted in the plane (x,z) with consideration of wear. It is shown in Fig. 8. The asymmetry is due to the effect of the tangential loading.

CONCLUSION AND PERSPECTIVES

A semi-analytical contact method has been employed for the simulation of fretting wear. The contact solver is based on the CG method and uses optimized FFT techniques to reduce

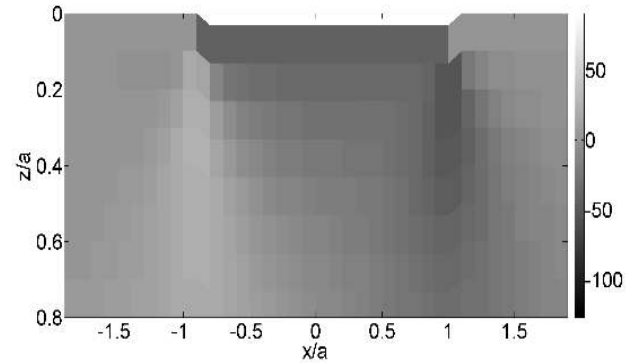


Figure 8: Distribution of the shear stress component σ_{xz} [MPa] in (x,z) plane

CPU time for contact resolution. A contact model permitting us to simulate coating, based on the equivalent inclusion method in the sense of Eshelby, has been first presented and validated through comparison with Multigrid solutions. The contact solver has been coupled to a wear law proposed by Fouvry and co-workers. The model then enables one to account for surface geometry changes due to wear. Several academic simulations have been performed thanks to the efficiency of the contact solver. Results obtained through this analysis show that wear has a significant impact on the contact pressure distribution and on the geometry of the contacting surfaces, leading to changes even for limited wear. First, the simulations for uncoated contact show that the pressure distribution tends to be flattened, erasing the peaks of pressure at the contact edges. Secondly, for coated surfaces, the numerical method permits to obtain a contact solution for a coating of non-uniform thickness, with is not possible with other analytical techniques such as integral transforms.

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