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Analysis of sliding behavior of coated surfaces under shear traction

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***Abstract*---**The tribological behavior of non conformal coated contacting surface i.e. rigid sphere on the coated flat material surface normal force and shear traction is studied through numerical results. The non-conforming contact is a fundamental problem in the contact mechanics, which has a wide range of applications. Contact is the science behind tribology, the interdisciplinary study of friction, wear and lubrication with major applications in particle handling or friction between contacting rough surfaces, brakes, Bearings and so on. In the present investigation, the contact between the coated flat and the rigid sphere under the combination of normal and tangential loading is analyzed using finite element software ABAQUS. The substrate material is chosen for the analysis High speed steel, Titanium alloy and coating material is TiN (E=305GPa) of 5µm thick with a bonding material is Chromium (379GPa) of 1µm thick at the interface between the coating and substrate is modeled by three dimensional finite element method (3DFEM) in order to optimize coating coated surface and bonding layer to coating fracture. The elastic-perfectly plastic contact is assumed on the flat model. The results obtained from the FEA are used in the understanding of the tribological behavior of the contact pair. This finite element model facilitates in evaluating the surface phenomena variables like contact stress distribution and at the surface level and sub-surface level, contact pressure, contact width, shear stress and displacement. The contacting surface is subjected to both normal load and shear traction. The tensile and compressive stress results obtained after sliding are normalized with the corresponding yield strength to determine the failure of surface or

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coating. It is found from the FEA result that the bonding layer between the coating and substrate has considerable influence on the stress distribution on the loaded coating surface. The surface seems to fail primarily due to tensile stress induced during sliding.

***Keywords*---**contact analysis, non-conformal contacts, coating, shear traction, hertz contact.

Introduction

Contact is necessary in any engineering applications to transfer force and power and hence it is almost indispensable field of study. It is the science behind the tribolgy, the interdisciplinary study of friction, wear and lubrication, with major applications in bearings and brakes, involving such issues as microscopic surface geometry, chemical conditions and thermal conditions. The mechanics contact between two bodies is very complicated. Also the force displacement is nonlinear and high stresses are induced. Because of the maximum stresses, the failure of machine parts to do with the properties near their surface. The important approach of developing new materials is therefore to improve the surface contact stress distribution by depositing thin coatings on components of industrial machinery to expect to meet stringent requirement and gained tremendous popularity over the past several decades. However, in spite of their increased use, the selections of coating materials and composition are often problematic for designer, because of development of various coating techniques.

In the case of sliding contact, the failures can be classified into two categories i.e. surface wear and surface delamination. Both of these failures are made are correlative the maximum contact stresses (maximum tensile stress, shear stress or equivalent stress). The maximum stresses are the important factors for the design of hard coatings in tribological applications. Therefore a qualitative analysis of these is important. Up to now the theoretical approach is fail predict successfully contact stress and wear behavior in the layered media. The analytical techniques using integral equations also are still closed form solutions available to designers for evaluating the contact characteristics of coated surfaces. In the present paper, the research is focused on the stress distributions within coatings under spherical Hertzian load with the finite element method.

Contact analysis

The theory of contact mechanics is concerned with the stresses and deformation which arise when two solid surfaces are brought into contact. Any tribological phenomenon involves the contact of surfaces and hence an understanding of the mechanics of surface contact is essential. When two bodies contact each other, the development of the surface stress is based on the contact geometry. If the contact is conformal i.e. a body making contact with the counter body with large apparent area of contact, the stress induced is equally distributed on the real of contact owing to the surface asperities. An example for conformal contact is the contact between the two flat plates. Contact of non-conformal takes place in an extremely small concentrated area known as Hertzian conjunction. If the contact is non- conformal, i.e. a body with lesser apparent area of contact, the induced will be very high in the contact region. An example for non-conformal contact is the contact between the sphere and the flat surface.

Elastic Deformation

We know, even institutively, that whenever two surfaces come in contact a stress form at that point. This even true, when you have just point contact of curved surfaces since the load deforms the two bodies turning the point into a contact area. This situation regularly shows up in such things as bearings, wheels and mating parts like gears. For given set of conditions met these stresses and related parameters can be calculated by using Hertz's formulation, hence the name Hertzian contact stress. In order to apply the Hertz theory and subsequent equations to analyzing the stresses in contact points we have to make sure our system meets a set of conditions.

1. The load force is normal and the induced contact area is small compared to radii of the two bodies.
2. The radii of curvature of the contacting bodies are large compared with the radius of the contact.
3. The dimensions of the each body are compared to the radius of the circle of contact.
4. The contacting bodies are in frictionless contact.
5. The surfaces in contact are continuous and nonconforming.
6. The contact between two bodies is perfectly elastic.

Consider the figure 1 in which rigid

Theoretical background

The maximum compressive stress at the contact between the sphere and the flat plate is given by sphere deforms the flat material by a displacement of magnitude of w under the application of normal

σ_{max}

$= 0.918 (4 P E^{*2} / R^2)^{1/3}$

load W. The contact radius a corresponding to the normal load W.

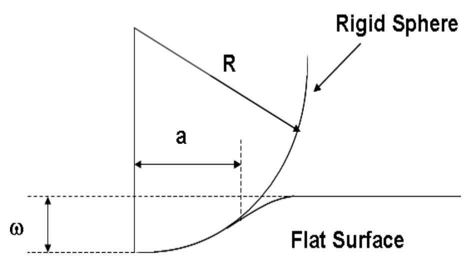

Fig.1 Deformation of flat surface pressed by rigid sphere

Plastic Deformation

When the normal load on the flat surface increased, the contact region which is initially elastic slowly moves to elastic-plastic regime and when the load is further increased the transition occurs from elastic-plastic to fully plastic regime. The critical displacement, w_c that makes the transition the elastic to the elastic-plastic deformation regime i.e. yielding inception is given by Chang et al (CEB). For $w \geq w_c$ the contact is elastic-plastic. Transition of material from elastic- plastic state is based on the yield criterion, wherein the plastic deformation initiates at a mean pressure of 2.8Y of the softer material. If the maximum shear stress, τ_{max} and principal stress, σ_x, σ_y and σ_z are plotted as a function of maximum pressure, max p, below the surface contact point, the plot of fig 2 is generated. This plot reveals that a critical section on the load axis, approximately 0.4a below the sphere surface. Many authorities theorize that this maximum shear stress is responsible for the surface fatigue failure of such contacting elements; a crack originating at the maximum shear, progress to the surface.

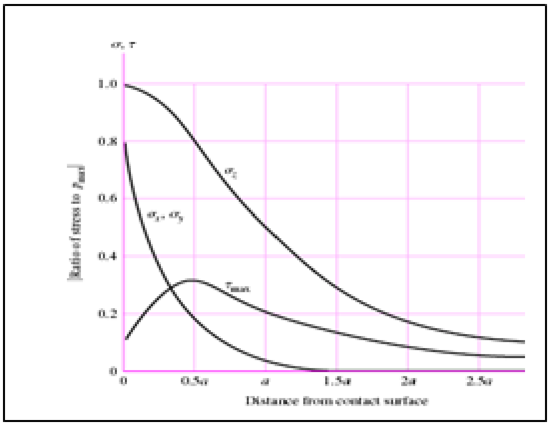

Fig 2 magnitude of the stress components below the surface as a function of maximum pressure.

Where the P is the applied force on the sphere in N, E^* is the equivalent young's modulus of the two materials or Hertzian elastic modulus, R is equivalent or relative radius of curvature in mm.

The contact radius between the non conformal point contact i.e. the sphere and the flat plate is given by

Contact Radius (a) = $(3PR/4E^*)^{1/3}$ Deformation (ω) = a^2/R
Hertz elastic modulus between the two bodies in contact is given by

$E^* = E_1 E_2 / [E_2 (1-\upsilon_1^2) * E_1 (1-\upsilon^2)]$ $R = R_1 R_2 / (R_1 + R_2)$ 2
Where E1 is elastic modulus of the rigid sphere (N/mm2), E_2 the elastic modulus of flat material (N/mm2).

Finite Element Model

A commercial ABAQUS 6.10 package was used to solve the contact problem. The rigid sphere is modeled as analytical surface. A rectangular block is drawn and extruded to represent the underneath the rigid sphere. The size of the elements influences the convergence of the solution and hence it has to be chosen with care. If the size of the elements is small, the final solution to be more accurate. However, we have to remember that the use of smaller size will also mean more computational time.

The flat surface, bonding surface and coating surface model are meshed using hexahedron element. The rigid sphere is modeled as analytical surface, so, it is not necessary to mesh the rigid sphere. The material properties are assigned to flat plate material, bonding material and coating material. The modeled surfaces are assigned as homogeneous section. The parts flat plate, bonding surface and rigid sphere created are assembled, in such a way that fully constrained. For analyzing the assembled parts, the surface interaction between the flat plate surfaces, bonding surface and coating surface as cohesive interaction property. The surface interaction of the sphere and the coating surface is defined using master and slave option. The more stiff body is treated as the master surface and less stiff body as slave surface. This type of contact is defined as surface to surface contact. The co-efficient of friction value is defined between the contacting pair. The boundary condition is applied in first step wherein bottom of the plate is fixed and the symmetric boundary condition is applied to represent the full model. The load is applied on the sphere. The commercial ABAQUS 6.10 package is used to solve the nonlinear contact analysis. During the solution phase the stress value is computed at the surface of the coating, bonding surface, flat plate surface and underneath surface of flat plate. The other parameters computed are contact pressure and deformation.

Verification

Although the FEM is widely accepted to analyze the compliance and the behavior of complex physical phenomena, the determination of a method of validating results should be needed. Up to now, there are no analytical or empirical solutions available for the Hertz contact of coated surface to the author's knowledge.

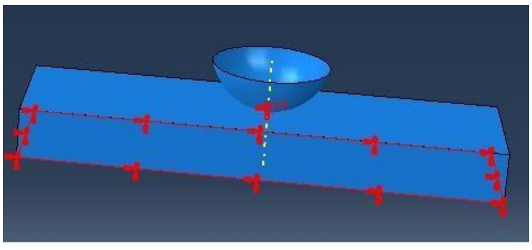

Fig.3 Boundary conditions applied to the model

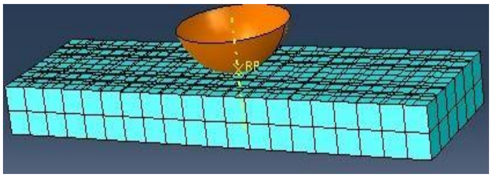

Fig.4 FEM model with mesh

Material Properties

Material	Young's Modulus(GPa)	Poisson ratio	Yield strength
Alumina (Rigid Sphere)	300	0.34	
High Speed Steel	210	0.30	415
Titanium Alloy	113.8	0.34	880
Chromium Bonding	300	0.25	

Stress distribution in the coated contact

Commonly surface layers are referred to with respect to their hardness. However, earlier studies have shown that the elastic properties are more relevant when evaluating the tribological behavior of coated surface. Bond layer materials used for TiN coatings are typically Chromium. The thickness and young's modulus values used in the stress simulations were chosen to be to these values representing common practical applications. The young's modulus values were chosen so that they represent coated surface systems with bond layer more complaint than the coating.

Bond layer thickness

The stresses and strains in the surface were with 3D FEM model. The first principal stresses in bond layer at a load of 225N and sliding distance 2mm from left to right, for 1μm and 2μm shown in fig.5 and 6. It can be observed that the stresses the same young's modulus value very close value. This indicates that the influence of thickness in the studied parameter range is very small. For this reason, we will concentrate on the effects of bond layer stiffness only for one bond layer thickness i.e. the bond layer thickness 1μm.

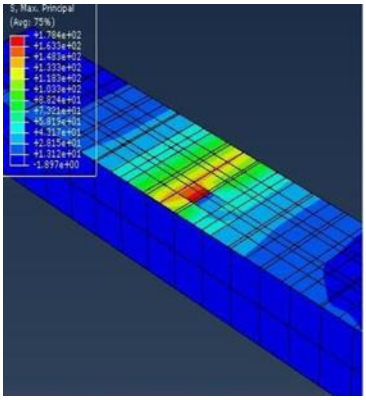

Fig.5 Max. Principal stress

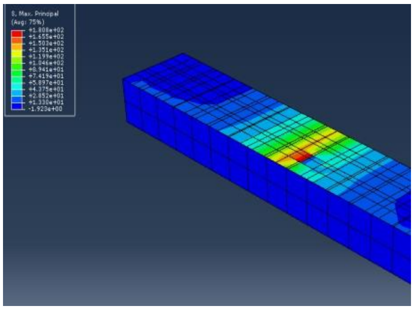

Fig. 6 Max. Principal Stress

Bond layer stiffness

The simulated stress distributions shown in figs. 7-9 are topographical stress field maps showing the first principal stresses on the coating with 5µm thick TiN coating with a bond layer (E=200GPa) and 1µm thick at the load 225, 250 and 275N and sliding distance 2mm. High tensile stress are found on the coating top behind the contact zone.

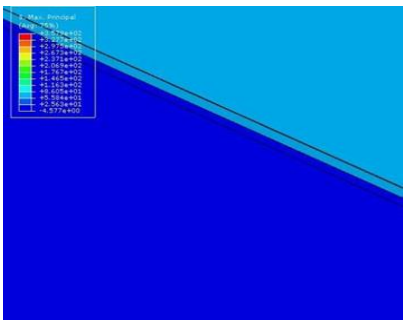

Fig. 7 Max. Principal stress

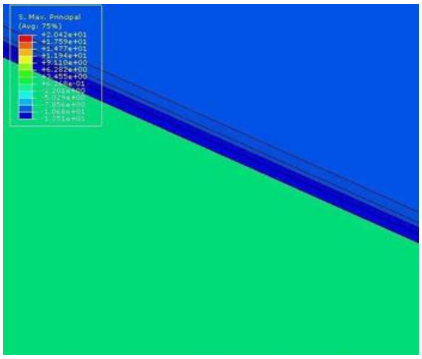

Fig. 8 Max. Principal stress

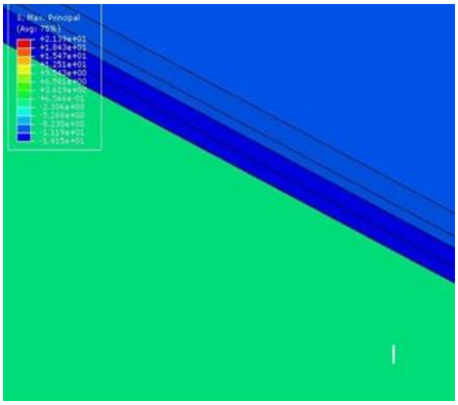

Fig.9 Max. Principal Stress

The simulated stress distribution are shown in figs. 10-12 are topographical stress field maps showing the first principal stresses on the coating with 5μm thick of TiN with bond layer(E=300GPa) and 1μm thick for the load of 225, 250 and 275N.

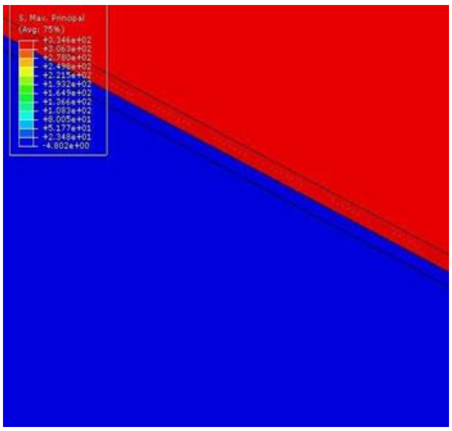

Fig.10 Max. Principal stress

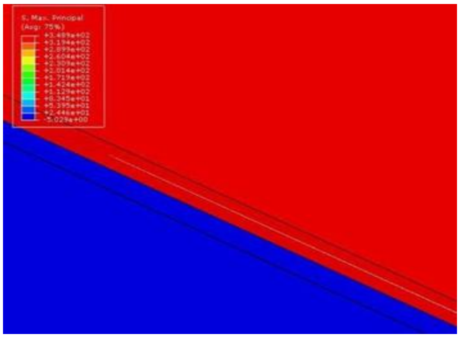

Fig.11 Max. Principal stress

Results and Discussion

A bond layer considerable influence on the stress and strain conditions of a loaded coated surface. The stress calculation showed that the influence of bond layer thickness is very small in the studied thickness range.

1. Cracking of the coating is strongly influenced by the tensile stress on the top surface ofthe coating.
2. Cracking at the coating/substrate interface or within the bond layer is influenced by the tensile and shear stresses in the interface region.
3. The coated surface load-carrying capacity is reflected by the level of compression at the coating top surface under sliding condition

Conclusion

A rigid sphere is sliding with the different load condition over a steel surface coated with ceramic coating and having a bond layer between the coating and the substrate were studied by 3D FEM modeling and give the following conclusions:

1. A bond layer between the coating and the substrate has a considerable influence on the stress and strain condition of a loaded coated surface.

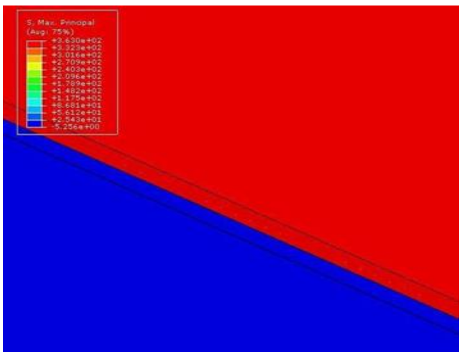

Fig.12 Max. Principal stress

2. The influence of bond layer thickness isvery small.
3. There is a high tensile stress behind contact zone and compressive stress in

front of the contact zone during sliding.

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