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Finite Element Analysis of Pin-on-Disc Tribology Test

Sumit Khot[1], Utpal Borah[2]

Manipal Institute of Technology Manipal-576104,Karnataka India

Scientific Officer-F, Metal Forming & Tribology Programme, Materials Technology Division, IGCAR, Kalpakkam-603102, Tamilnadu, India

Abstract: ***The main objective of this work was to evaluate the state of stress and strain in Pin-on-Disc (POD) tribology test setup with of 316LN austenitic stainless steel as the tribo elements under self-mated conditions using finite element method (FEM). Type 316LN stainless steel is a major core and structural material in the Prototype Fast Breeder Reactor (PFBR. In PFBR there are many in-core and out-of-core component involving contact pairs of 316LN stainless steel and undergoing sliding wear during operation as well as maintenance. Estimation of wear during operation of the machine would lead to developing appropriate wear mitigation approaches. However, measurement of in-situ wear in machine components is very difficult, if not impossible. Finite element method (FEM) based numerical modeling of machine operation with appropriate wear models would enable estimation of wear a-priori. As accuracy of calculated wear values strongly depends on the state of stress and strain in the components, accurate modelling of the state of stress and strain is essential.***

Keywords: FEA, Tribology, ABAQUS, Pin-On-Disc, WEAR, state of stress, state of strain, Elastoplastic Analysis.

1. Introduction

Type 316LN stainless steel is a major core and structural material in the Prototype Fast Breeder Reactor (PFBR) designed by Indira Gandhi Center for Atomic Research (IGCAR), Kalpakkam. Currently PFBR is being constructed in Kalpakkam. It consists of many in-core and out-of-core component made up of different types of materials. These component pairs undergo sliding wear during operation as well as maintenance. Therefore a study and analysis of mechanism of wear has to be understood and predicted for the optimum design and effective operation of components. There are many contact pairs involving 316LN stainless steel in PFBR and extensive experimental works are being carried out in Metal Forming and Tribology Programme (MFTP) in IGCAR. MFTP has embarked upon on numerical modeling of tribological phenomena in fast breeder reactor components.

The objective of the project was to evaluate the state of stress and strain in Pin-on-Disc (POD) tribology test setup with of 316LN austenitic stainless steel as the tribo elements under self-mated conditions using finite element method (FEM). Finite element analysis has been used in tribology and the study of wear to model the phenomena at widely different length scales. POD test covers the laboratory phenomena for determining wear during sliding. In the POD test, pin is held stationary under a specified load while the disc rotates beneath it at a constant velocity. According to ASTM G-99(05) standard specified that pin can have any shape to stimulate a specific contact, but the spherical tip are often used to ensure normal load transfer to the disc as well as to establish initial point contact.

2. Objective of this Work

The primary aim of this work is to evaluate the state of stress and strains and its evolution with the sliding cycles for the case of 316LN Stainless Steel in the form of a rounded pin in contact with a metallic spinning disc using FEA program. The simulation is expected to permit isolation of the individual effects of operating parameters such as load, sliding speed and sliding distance. Also, in terms of suggesting further improvements to the simulation process and widening its application, this type of conformal contact in which, the deflection of the pin arising from the frictional reaction created during sliding is non-negligible.

2.1 Motivation

The motivation is from the fact that many machine components fail due to wear and estimation of wear during operation of the machine would lead to developing appropriate wear mitigation approaches. However, measurement of in-situ wear in machine components is very difficult, if not impossible. Finite element method (FEM) based numerical modeling of machine operation with appropriate wear models would enable estimation of wear a-priori. However, accuracy of calculated wear values strongly depends on the state of stress and strain in the components. Hence, accurate modeling of the state of stress and strain isessential to estimate wear accurately. Hence, this work was taken up to establish the FEM modeling methodology to evaluate the stress-strain state accurately in the ball-on-disc contact configuration in the form of pin-on-disc tribology test. This FEM model would lead towards establishing a fully functional FEM model extendable to real world components having contact pairs to estimate wear a-priori.

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3. Tribology

Tribology is the science and engineering of interacting surfaces in relative motion. It includes the study and application of the principles of friction, lubrication and wear. Tribology is a branch of mechanical engineering. The tribological interactions of a solid surface's exposed face with interfacing materials and environment may result in loss of material from the surface. The process leading to loss of material is known as "wear". Major types of wear include abrasion, friction (adhesion and cohesion), erosion, and corrosion. Wear can be minimized by modifying the surface properties of solids by one or more of "surface engineering" processes (also called surface finishing) or by use of lubricants (for frictional and adhesive wear). Contacting elements of structures are very common in technology. Many mechanical devices and mechanisms are constructed with the aid of component parts contacting one with another. Contact regions occur between tools and work pieces in machining processes. Loads, motions and heat are transmitted through the contacts of structures. Friction and wear accompany any sliding contact. It has been agreed that wear cannot be totally prevented. In machine technology, wear is an equally important reason of damage of materials as fracture, fatigue and corrosion. The modeling of friction and wear is an important engineering problem. In the process of design of machine elements and tools operating in contact conditions, engineers need to know areas of contact, contact stresses, and they need to predict wear of rubbing elements. Friction, wear and contact problems are subjects of numerous experimental and theoretical studies. The very complex nature of tribological phenomena is a reason that many problems of contact mechanics are still not solved. The modeling of friction and wear can be carried out not only with the aid of laboratory tests but using also mathematical models and computer simulations. Due to computer simulation techniques, physical and mechanical phenomena in real objects can be reconstructed with a high degree of precision. There is still a need for efficient and reliable computational procedures of contact problems taking into account complex phenomena of friction and wear. On the one hand, the accuracy of numerical procedures should be improved. On the other hand, new models of friction and wear should be included in numerical calculations. Contemporary numerical codes do not discuss how to calculate and how to predict wear. Better understanding and control of wear in materials can be done with the aid of new models of wear.

4. Wear

According to ASTM G-40-04, Wear is defined as ―Damage to a solid surface, usually involving progressive loss of material, due to relative motion between that surface and contacting substances. Wear is related to interactions between surfaces and more specifically the removal and deformation of material on a surface as a result of mechanical action of the opposite surface. The need for relative motion between two surfaces and initial mechanical contact between asperities is an important distinction between mechanical wear compared to other processes with similar outcomes. The definition of wear may include loss of dimension from plastic deformation if it is originated at the interface between two sliding surfaces. However, plastic deformation such as yield stress is excluded from the wear definition if it doesn't incorporates a relative sliding motion and contact against another surface despite the possibility for material removal, because it then lacks the relative sliding action of another surface. The complex nature of wear has delayed its investigations and resulted in isolated studies towards specific wear mechanisms or processes. Some commonly referred to wear mechanisms (or processes) include Adhesive wear, Abrasive wear, Surface fatigue, Fretting wear, Erosive wear. In this project the wear mechanism was predominantly observed was adhesive wear. So the description of adhesive wear is presented here. Adhesion is major contributor to sliding resistance (friction) and it was inferred in mechanics at least to be operative in wear as well.Thus abrasive substance are not found, if the amplitude of sliding is greater than fretting , and if the rate of material loss is not governed by principle of oxidation, then the adhesive wear is said to occur.[10]

Adhesive wear can be found between surfaces during frictional contact and generally refers to unwanted displacement and attachment of wear debris and material compounds from one surface to another. Two separate mechanisms operate between the surfaces.

1) Adhesive wear are caused by relative motion, "direct contact" and plastic deformation which create wear debris and material transfer from one surface to another.
2) Cohesive adhesive forces, holds two surfaces together even though they are separated by a measurable distance, with or without any actual transfer of material. The most frequently used model is the linear wear equation

$$\tilde{Q} = K\tilde{p} \quad (1),$$

Where, the volume wear rate is proportional to the normal load. This model is often referred to as the Archard's wear law, though its basic form was first published by Holm [3]. The model was based on experimental observations and written in the form…….

$$\frac{V}{s} = k\frac{F_N}{H} \quad (2)$$

Where, V—Volume of material removed,s—sliding distance, F_N—applied normal load, k—Dimensionless wear coefficient, H—Hardness(ratio of load over projected area) is that of softer material(pin).

5. Pin on Disc Test

This test covers a laboratory procedure for determining the wear of materials during sliding using pin disc apparatus. Materials are tested in pairs under nominally non-abrasive condition. For the POD test two specimens are required. One, pin with radius tip is positioned perpendicular to a flat circular disc. The test machine causes either the disc specimen or the pin specimen to revolve about the disc center. In either case, the sliding path is a circle on the disc surface. The plate of the disc may be oriented horizontally or vertically. The pin Specimen is pressed against the disc at a specified load usually by means of an arm or lever and attached weights. Other loading methods have been used such as hydraulic or pneumatic. Wear results are reported as volume loss in cubic mm. Wear results are usually obtained by conducting test for a

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selected sliding distance and for selected values of load andspeed.[14]

Figure1: Pin on Disc Test

6. Hertz Contact Analysis

Contact mechanics deals with bulk properties that consider surface and geometrical constraints. It is in the nature of many rheological tools to probe the materials from "outside". For instance, a probe in the form of a pin in a pin-on-disk tester is brought into contact with the material of interest, measuring properties such as hardness, wear rates, etc. Geometrical effects on local elastic deformation properties have been considered as Hertz Theory of Elastic Deformation. This theory relates the circular contact area of a sphere with a plane (or more general between two spheres) to the elastic deformation properties of the materials. Finite element simulations of ball indentation tests will be perform and analyzed using the Hertz ball indentation method. The accuracy and reliability of this method will be assessed. It shows that ball indentation testing techniques can be used to evaluate stresses and strains between the ball and disc. This work is aim at assessing the accuracy and reliability of this method based on finite element analysis (FEA) simulations of the ball indentation process. [21]

Continuous indentation tests are now commonly used to measure the elastic moduli of materials in accordance with the unloading response of a plastic indentation. However, a previous technique utilizing a Hertz ball indentation test, applied to the initial loading behavior, is shown to be equally useful provided that sufficient sensitivity is available for the loading curve measurements. Continuous loading measurements for the indentation of lignin are used to demonstrate the possibility of using the loading curve for computing the elastic modulus, and the analysis is carried over to test results on 316LN stainless steel materials. The normal contact pressure distribution is calculated by both FEA and the Hertz formulaefor sphere on plane configuration……

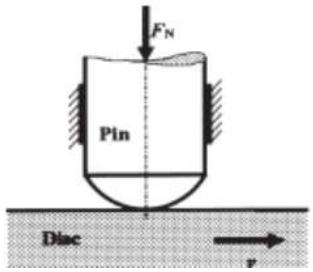

Figure 2: Pin on Disc Hertz Contact

$p = p_{max}\left(1-\frac{x^2}{y^2}\right)$ (3) with $r_0 = \sqrt[3]{\frac{3F_N R}{4E^*}}$

$$p_{max} = \frac{3F_N}{2\pi r_0^2} \quad (4)$$

Where,

$E^* = E/2(1-\mu^2)$ is the normalized elasticity modulus (Pa),

r_0 – contact area between pin and disc

p_{max}--contact pressure between pin and disc

F_N—applied normal load on the pin

The plastic deformation was disabled and the friction is neglected in the model. [24]

7. Finite Element Method

FEM consists of a computer model of a material or design that is stressed and analyzed for specific results. It is used in new product design, and existing product refinement. Modifyingan existing product or structure is utilized to qualify the product or structure for a new service condition. In case of structural failure, FEM may be used to help determine the design modifications to meet the new condition. There are generally two types of analysis that are used: 2-D modeling, and 3-D modeling. While 2-D modeling conserves simplicity and allows the analysis to be run on a relatively normal computer, it tends to yield less accurate results. 3-D modeling, however, produces more accurate results while sacrificing the ability to run on all but the fastest computers effectively. Within each of these modeling schemes, the programmer can insert numerous algorithms (functions) which may make the system behave linearly or non-linearly.[6] Linear systems are far less complex and generally do not take into account plastic deformation. Nonlinear systems do account for plastic deformation, and many also are capable of testing a material all the way to fracture. The main task of the Finite Element Analysis (FEA) is to identify the nonlinear behavior such as interaction of one part to another (Pin and Disc). Non- linear problems pose the difficulty of describing phenomena by realistic mathematical and numerical models and difficulty of solving nonlinear equations that result. Pin on Disc contact problem are categorized as nonlinear because of the stiffness, loads, deformation and contact boundary conditions. The FE wear calculations involve solving the general contact problem with the area of contact between the bodies not known in advance. The analysis is therefore non-linear. This problem was enforced using penalty function method and geometric description of slave and master surface. The FEA mesh is constructed using tri-linear isoperimetric 8-noded brick elements. The problem is modeled as being quasi-static and as such time-independent. In the structural analysis the degrees of freedom are defined as nodal displacements. The equations for every element are assembled into a set, expressed in the structural level as

$$[K]\{D\} = \{R\}, \quad (5)$$

Where, [K] is the structural or global stiffness (N/m) matrix, {D} is the structural nodal displacement or deformation (m) vector and {R} is the vector of structural nodal loads (N).[24]

Virtual motions are useful concepts in mechanics of material. They are used both in the analytical formulation of problems and also constitute the foundation of the finite element methodology. Virtual motions are imaginary movements of material points and the method of virtual power consists of determining the associated work or power involved. The principle of virtual power is given by,

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$$\int_\Omega \delta\varepsilon.\sigma d\Omega - \int_\Omega \delta d.f\, d\Omega - \sum_{i=1}^{n} \delta d_i.p_i = 0 \quad (6)$$

Where, Ω is the Finite Deformation Analysis volume, δε is the Virtual rate of deformation tensor, σ is the Cauchy stress tensor, δd is the Virtual displacement vector, f is the Body force vector, and p_i is an External force vector acting on points n. For the Continuum element (Pin on Disc), finite element formulation is given by,

$$\sum_{j=1}^{m}(\delta u_e^T . F_e)_j + \sum_{j=1}^{m}(\delta u_e^T . M_e . u_e)_j - \sum_{i=1}^{n} \delta d_i^T . p_i = 0 \quad (7)$$

Where, m is the number of elements, ue is the Element displacement vector, Fe the Element force vector, Me is the Element mass matrix.

Newton's method with updating of the tangential stiffness is used during each iteration. The linear equation system is solved using a preconditioned conjugate gradient solver and as convergence criteria, norms for residual forces and increment displacement fields. In an elastic-plastic problem, the concentrated loads and point supports are associated with the high local stresses. To solve this condition at these locations, it takes much iteration which is wasted. The system is preconditioned using a Crout element-by-element scheme.

The finite deformation analysis is formulated using a polar decomposition approach and the Jaumann objective stress rate. The three essential ingredients of elastic and plastic analysis are yield criterion, a flow rule and hardening rule. The yield criterion relates the state of stress to the onset of yielding. The flow rule relates the state of stress {σ} with the corresponding increment of plastic strain {dεp} when an increment of plastic flow occurs. The hardening rule describes how the yield criterion modified by straining beyond initial yield.

Elastoplasticity of the deformable material is specified according to a bilinear Von-Mises model using isotropic hardening. Since the deformable material has a very low hardening tendency, a bilinear model with a constant tangential modulus is a suitable fit for the material parameters. The plastic modulus for isotropic hardening isgiven by,

$$H' = \frac{E.E_T}{E - E_T} \quad (8)$$

Where E is the elastic modulus,
E_T isthe tangent modulus.
The evolution equation for the yield surface by isotropic hardening is given by,

$$k = k0 + \frac{\sqrt{2}}{3} H'.\lambda + \frac{1}{\sqrt{3}} H'\varepsilon_e^p \quad (9)$$

Where,
λ is proportionality coefficient

$$\varepsilon_e^p = \text{Strain in elastic-plastic material} = \sqrt{\frac{2}{3}\varepsilon^p \cdot \varepsilon^p} \quad (10)$$

The contact problem is kind of geometrically non-linear problem that arises when different structures, or different surfaces of a single structure comes into contact or slide on one another with friction. Contact forces either gained or lost must be determined in order to calculate the structural behavior. Contact is enforced using a penalty function method, i.e. the potential energy of the finite element model is added a so-called penalty term as

$$\Pi = \frac{1}{2} U^T . K.U - U^T . F + \frac{1}{2} P^T \tau . P \quad (11)$$

Where, U is the nodal displacement vector, K the stiffness matrix, F the nodal force vector, P the penetration displacement vector, and τ the penalty parameter vector.

Contact identification is carried according to three rules. First, a position vector is identified between the current location of a node, Φ, and a base point in the contact sphere, Φc

$$v = \Phi - \Phi c \quad (12)$$

If |v| > R, where R is the radius of the sphere, no contact occurs. The analyses were carried out using multiple values of the penalty parameter in order to verify the independency of results on the selected values.

Usually the friction between the contacting surfaces must be taken into the account and it must be represented by Coulomb friction model because it influences structural behavior and the problem may be quasi-static and time independent. So the equivalent frictional stress is defined by,
$\tau^e = (\tau_i^2)^{1/2}$, i=1,2,…,

where the frictional shear components are proportional to the coefficient of friction and the contact pressure, i.e.

$$\tau_i = \mu p . \frac{\gamma_i}{\gamma^e} \quad (13)$$

Where, γ_i is the slip rate in direction I,γ^e is the slip velocity,

μ is the coefficient of friction (given a constant results independent value), p is the contact pressure. [15]

The commercial finite element (FE) software ABAQUS can handle several material and structural non-linearties, such as plasticity, visco-elasticity, friction, etc. The coupled-field analyses, for instance thermal–structural, can be performed as well.

8. Methodology

Methodology includes the modeling and FEA of pin on disc model by ABAQUS v.6.10, which includes the pre-processing, analysis and post-processing stages. It is possible to make an exact copy of the configuration of the pin on disc test, but this will require a very long CPU-time (Central Processing Unit time). This is not necessary to obtain good results. A simplified model will be made. The model that is made contains as less elements as possible in order to make the simulation run very fast. Difficulties with the settings like materials, boundary conditions and loads can be solved easily. The main goal is to get the simulation running. After this step, the model will be refined in order to get more accurate results. A proper friction formulation will be chosen and the number of elements will be increased to be able to check the real pressure distribution between pin and disc, due to the different materials of the pin. In the property module the different material properties and body sections are assigned to specified regions of the pin and the disc. The structure with spherical ended pin with radius of R=5mm was thus represented in 3-dimensional (3-D) sphere on plane contact model. Both the pin and disc are made up of 316(LN) Stainless Steel. Young's modulus and Poisson's ratio of material are 197GPa and 0.29 respectively. Density of material is 7850 kg/m^3. To find plasticity, using Ludwingson Equation,

$$\sigma = K_1 . \acute{\varepsilon}^{n}_{1} + e^{(K_2 + n_2 . \acute{\varepsilon})} \quad (14)$$

Where,
K_1 = 1232.8, n_1=0.312, K_2 = 5.1, n_2 = -70.6
Initial yield stress = 320.27MPa at έ = 0

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Table 1: Plastic Properties (From Ludwingson Equation)

Yield Stress	Plastic Strain
320.27 MPa	0
488.94 MPa	0.05
601.6 MPa	0.1
682.08 MPa	0.15
746.129 MPa	0.2
799.92 MPa	0.25
846.7 MPa	0.3
888.46 MPa	0.35

9. Results and Discussion

In the beginning elastic FEA was carried out to evaluate the maximum contact pressure between the pin and the disc. The results from elastic FEA were compared with analytically calculated maximum contact pressure from Hertz contact theory and the optimum FEA mesh size was determined. Subsequently Elastoplastic FEA of pin on disc was done with constant load on the pin and various friction conditions. The state of stress and state of strain results obtained are discussed in the following. Normalized Elasticity Modulus is $1.07x10^{11}$ N/m^2.Contact area between pin and disc is 7.05×10^{-5} N/m^2. And maximum contact pressure between pin and disc is 9.6×10^8 N/m^2.

9.1 Comparison between various mesh sizes

The mesh sizes of pin on disc configuration were analyzed with the FEA approach. The plastic deformations and the influence of friction on the contact pressure distribution were considered to be negligible in this case.

The concentrated load was applied on pin was F=10N and the sizes of the elements used in mesh validation were 100 µm, 120 µm, 140 µm, 200 µm.

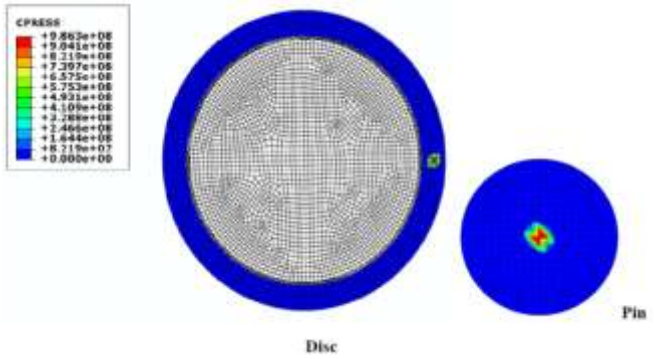

Figure 3: Contact Pressure between Pin and Disc of mesh size of 100µm

Table 2: Comparison between Analytical and FEA Results of different mesh sizes

Mesh Size, micron	Load	Maximum Von mises Stress (S,mises) N/m²	Maximum Contact Pressure FEA (CPRESS) N/m²	Maximum Contact Pressure Analytical Values (CPRESS) N/m²	% error
200	10N	2.019 X 10^8	2.486 X 10^8	9.6 X 10^8	28.6 %
140	10N	3.693 X 10^8	4.141 X 10^8	9.6 X 10^8	13.1%
120	10N	3.511 X 10^8	5.536 X 10^8	9.6 X 10^8	7.3%
100	**10N**	**8.043 X 10^8**	**9.863 X 10^8**	**9.6 X 10^8**	**2.7%**

From the above given table, the correct element size of contact between pin and disc is 100 micron and it will be used for further analysis of pin on disc test.

9.2Elastoplastic Analysis of Pin on Disc

In Elastoplastic analysis, both elastic and plastic properties [12] must be put in the FEA model and check the stress-strains properties at the load of 10N. A very fine mesh is employed in the contact zone of 100 micron element size of pin and disc where the deformation gradients are the highest. This fine mesh region would have to extend the full length of the unit cell to obtain an accurate solution. The normal load of 10N is applied through the axis of the rigid pin head of radius 5mm and the different coefficients of friction values such as 0, 0.1, and 0.3 are used in contact between pin and disc.

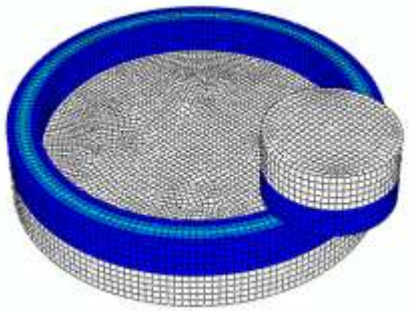

Figure 4: Elastoplastic Analysis of Pin on Disc Visualization

9.3 State of Stress and State of Strain in Pin on Disc

- Von Mises Stress(S,Mises):-These are the equivalent tensile stresses between Pin and Disc material when both materials are in sliding contact and material start to yield. The Von Mises stress satisfies the property that two stress states with equal distortion energy have equal Von Mises stress.

- Plastic Equivalent Strain (PEEQ): The equivalent plastic strain gives a measure of the amount of permanent strain in an engineering body. Most common engineering materials exhibit a linear stress-strain relationship up to a Stress level known as the proportional limit. Beyond this limit, the stress-strain relationship will become nonlinear, but will not necessarily become inelastic. Plastic behavior of Pin on Disc FEA model characterized by non-recoverable strain or plastic strain begins when stresses exceed the material's yield point.

- Logarithmic Strain(LE):- The logarithmic strain is true strain to provide the correct measure of the final strain when deformation takes place in a series of increments, taking into account the influence of the strain path. It is necessary to verify that the disc material experiences the plain strain conditions in the central region of the wear track. For this to be the case, the logarithmic strain components LE12, LE13 and LE22 must be equal or close to zero in the central region. The FE results from 3D non-linear contact calculations for a pin load of 10N and using coefficient of friction of 0, 0.1 and 0.3.

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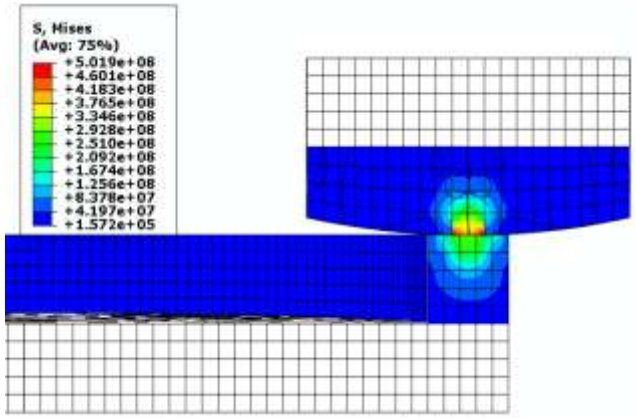

Figure 5: State of Stress

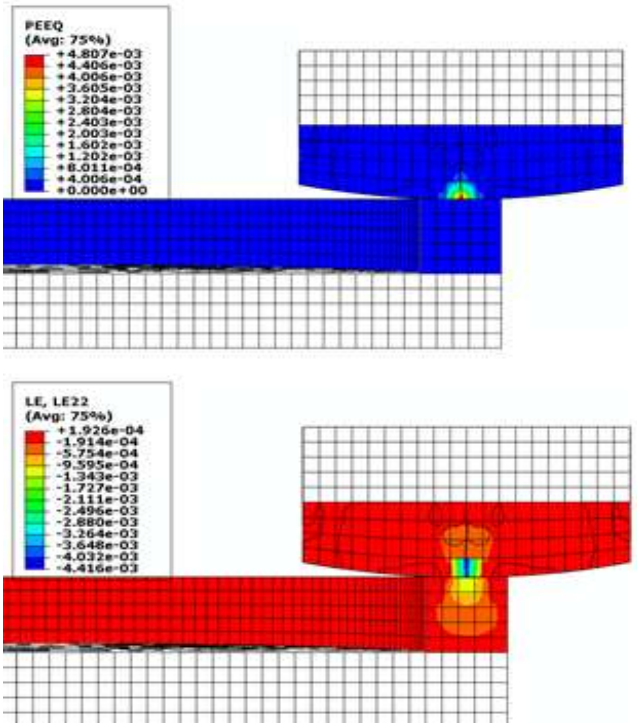

Figure 6: State of Strain

Table 3: State of Stress and State of Strain in Pin-on-Disc

Results	$\mu = 0$	$\mu = 0.1$	$\mu = 0.3$
Von Mises Stresses (S,Mises), N/m^2	5.019 X 10^8	5.137 X 10^8	5.227 X 10^8
CPRESS(Contact Pressure), N/m^2	9.858 X 10^8	9.858 X 10^8	9.857 X 10^8
PEEQ (Plastic Equivalent Strain)	0.004807	0.004915	0.005738
LE_{11}(Logarithmic Strain)	0.001516	0.001543	0.001738
LE_{22}	0.0001926	0.0001943	0.000487
LE_{33}	0.001516	0.001541	0.001727
LE_{12}	0.007282	0.007386	0.008160
LE_{13}	0.0004111	0.0007807	0.002524
LE_{23}	0.007282	0.008127	0.001059

Note: Given Table shows normalized logarithmic strain component of LE_{12}, LE_{13} and LE_{33} which is negligibly small in the whole studied region. Based on these results, it is clear that the plain strain region exists up to depth below the contacting pin. For the proposed methodology to be valid, the maximum depth of the wear track must lie within this region.

10. Conclusion

In this work the pin-on-disc (POD) tribology test setup was modeled using elastoplastic finite element method (FEM) for 316LN austenitic stainless steel with 10N applied load and 100 rpm speed using the commercial FEM code ABAQUS version 6.10. The FEM model was analytically validated and optimum mesh size for the contact elements were determined by comparing the maximum contact pressure from elastic analysis results with analytical results from Hertz contact theory applied to ball on flat surface. Using this optimum mesh size elastoplastic FEM analysis was carried out to determine the state of stress and strain in the pin and the disc for different friction conditions between the pin and the disc. For this type of contact problem (ball on flat surface) the FEM mesh size for accurate maximum contact pressure is strongly dependent on the applied load and material properties. The optimum mesh size for 316LN austenitic stainless steel with 10N applied load is 100 micron for which the error in maximum contact pressure is 2.7%.The von-Mises stress reduces for elastoplastic model as compared to elastic model. With increasing friction, both equivalent plastic strain and von-Mises stress increase. A state of plane strain condition is established along the mean diameter of the contact ring as depicted by negligible ε33 below the contact point.

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