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Numerical Analysis of Hemispherical Pin on disk Tribometer under Lubricated Condition

Shubrajit Bhaumik*, Arnab Basak
Department of Mechanical Engineering, SRM University, Chennai
***Corresponding author: Email: shubrajit.research@gmail.com, Ph: 9840439097**

ABSTRACT

Wear component is a crucial factor influencing a product's service life which is based on material's property and behavior under certain conditions. Therefore, it is an important part of engineering to predict the wear rate of the components. In this work a lubricated sliding wear has been simulated using commercial FEA package (Abaqus 6.13) between hemispherical pin and flat disk. The contact pressure is implemented by moving the boundary nodes. The proposed framework relies on understanding how the pin slides on the disk with a lubricant film between them. In order to have a combined effect of metal-metal contact and lubrication, a mixed regime was considered in this present work. Simulation is done on sphere on flat geometry type of surface contact involving effect of three elements together. The output has been compared with analytical result. It was observed that the contact pressure measured through simulation is 50-60% accurate to the analytical value. In later stage, a conclusion on physical aspect of such contacts has been made.

KEY WORDS: finite element analysis, pin on disc, lubricated condition.

1. INTRODUCTION

Numerical analysis of any physical phenomenon is based on approximation done by mathematical formulation derived from established physical theory of natural fact and some presumed assumptions to narrow down the objective of any research. A pin-on-disk configuration is the simplest and most widely used experimental method for wear measurement. The acumen of this study is on sliding friction between two metallic surfaces and viscous erosion committed by lubricant oil.

In present work a hemispherical pin and a flat disk was considered. The analytical formulae come from Hertz's work on various types of surface contact. Few works have been reported on dry sliding wear. In a work on dry simulation by Podra & Andersson (1999) FEA simulation results showed measuring wear coefficient and sliding distance change equivalence can give 60-40% accuracy of wear simulation. Benabdallah and Olender (2006) proposed that the wear simulation can be done by two-dimensional FE model of a tribosystem. The work on numerical simulation by Shen (2010) showed that nonlinear wear simulation gave the closest result to experimental result and can be solved through a series of analysis with quasi-static model. Though some work has been reported on dry wear but no systematic work has been found in the case of lubricated wear. In the present work, a lubricant film has been considered between the hemispherical pin and flat disk. The effect of viscosity has been taken into consideration by Stokes' equation. The computer model has been made in commercial FEA package Abaqus 6.13. The contact pressure is the key parameter and has been compared with the value calculated from Hertzian contact pressure equation.

Analytical formulation by Hertz: The shape of the surface in contact depends on the shape of the contacting body or curvature of the surfaces. The convex and concave surfaces are categorized as positive and negative surface respectively. If the center of curvature lies within the solid then the curvature is positive, if it lies outside the solid then the curvature is negative. The contact geometry between flat and convex curvature is shown in Figure1.

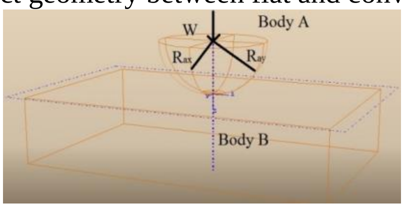


Figure.1. Schematic diagram of sphere on plane

The Hertzian approach to calculate reduced curvature R' is,

$$\frac{1}{R'} = \frac{1}{R_x} + \frac{1}{R_y} \qquad (1)$$

Now,

$R_{bx} = R_{by} = \infty$, as it is a flat surface.

Therefore,

$$\frac{1}{R'} = \frac{1}{R_{ax}} + \frac{1}{R_{ay}} \qquad (2)$$

The reduced Young's modulus E' is,

$$\frac{1}{\mathrm{E}'} = \frac{1}{2}\left[\frac{1-\vartheta_a^2}{E_a} + \frac{1-\vartheta_b^2}{E_b}\right] \qquad (3)$$

Where, ϑ_a , ϑ_b are the Poisson ratios.

Contact area dimension,

$$a = \left(\frac{3WR'}{E'}\right)^{1/3} \quad (4)$$

Where W is normal load.

Maximum contact pressure,

$$P_{max} = 3W/2\pi a^2 \quad (5)$$

The maximum contact pressure from simulation output has been compared with the value obtained from the above equations.

Calculating Lubricant's effect: The primary wear mechanisms which take place in the presence of a lubricant are adhesive wear and fatigue wear. The effect of fatigue wear has been neglected in this present work. The properties of the liquid are responsible for erosive wear by lubricant. Among them, surface tension is responsible for oil film formation and viscosity for the liquid friction or viscous drag. In the present work, the viscous effect of lubricant has been taken into consideration. In the conceptual model, it has been assumed that the oil is not continuous throughout the experiment, but film thickness is uniform where ever oil film is persistent. From the above assumption, it can be assumed that the type of flow of the oil is laminar. In case of laminar flow past a sphere Stokes has calculated the viscous drag neglecting the inertia terms of Navier-Stokes equation. The Stokes' equation for drag force is given below,

$$F_d = 6\pi\mu RV_T \quad (6)$$

Where, F_d = Drag force, μ= Coefficient of viscosity, R= Radius of sphere, V_T = Terminal velocity.

As the lubricant was assumed continuous and uniform it can be assumed that there is no contact between pin and the disk. It can further be assumed as a falling pin with constant speed. Thus, the sum of buoyant force and drag force is equal to the weight of the body which implicates that drag force is in the opposite direction to the normal load.

Finite Element Modelling: The 3D model has been created by using GUI of commercial FEA package Abaqus 6.13. A hemispherical pin (φ10mm) of EN 24 was modelled. EN 24 has been chosen as it is used in manufacturing of gears. The disk(EN 31) in the experiment generally has an inner diameter of 10 mm and an outer diameter of 80 mm. Castor oil has been chosen as the lubricant because of its high viscosity (0.650 Ns/m^2) and its evolving trend of being used as an industrial lubricant In order to reduce the number of elements of the model while meshing the modelled disk in the present work is a portion of whole disk this is directly relevant to the FEA analysis. So, only the portion of 50 mm to 80 mm diameter of the disk has been modelled. In assembly module (Figure 2) the pin on disk model was created assembling an instance with two parts the pin and the disk. The wear scar diameter was 120 mm.

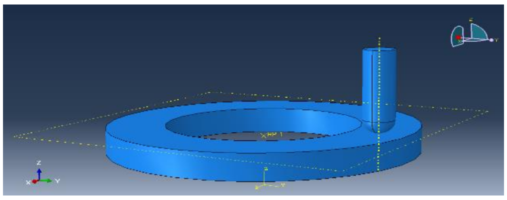
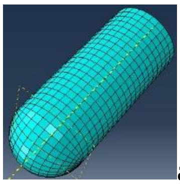
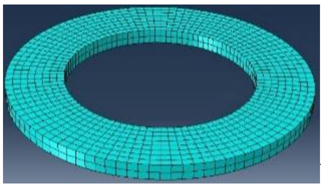


Figure.2. Pin-on-Disk assembly

The finite sliding motion is assumed to be frictionless. The interaction is between hemi-spherical surface of pin and flat surface of disk and is a surface to surface explicit type. Only frictionless tangential behavior has been assumed to define interaction property. In this model frictional sliding motion can also be created using penalty contact method. In step module apart from initial step other steps were created as dynamic explicit. The step time was set as 1 sec and the nonlinear property has been taken into account. In load module the load (1500N) has been given at step 1. The type of load is pressure and it is given on flat surface of the pin. The viscous drag is given along the opposite direction to the pressure and its magnitude is calculated from the property of castor oil. Absolute viscosity of castor oil is 0.650 Ns/m^2. Disk rotation has been modelled as 200 rpm rotation. In mesh module both pin and disk have been meshed with quad elements (C3D8R) in Abaqus 6.13. The pin element size is 1 mm and disk element is 2 mm. The hemispherical face of pin is partitioned separately to get better uniform mesh. The mesh control for pin is opted as advanced front and for disk it is median axis.



a



b

Figure.3. Meshed (a) pin (b) disc

2. SIMULATION RESULT

Any wear on surface happens due to the local plastic deformation of the surface. As in this simulation no plasticity property has been defined the value of maximum contact pressure came less than that of analytical value. In this simulation with the increase of simulation period the maximum contact pressure increased and appeared to get saturated after certain duration of simulation.

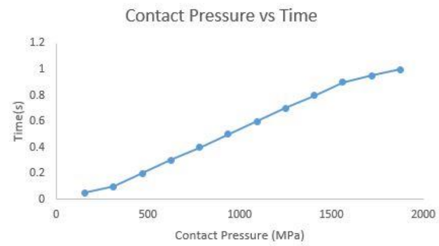


Figure.4.Contact pressure distribution from Abaqus

The maximum contact pressure from the simulation came 1874 MPa (Figure 4).

Model Validation: Validation was done through comparison with analytical contact pressure. As the lubrication regime is taken as mixed lubrication the analytical contact pressure is the subtraction of viscous drag from the Hertzian maximum contact pressure. The maximum contact pressure from simulation has been found to be:

$$P_{max} = 3380.87 \text{ MPa}$$

And the analytical maximum contact pressure is

$$P_{max} - F_d = 3380.87 - .07698 = 3380.79 \text{ MPa}. \quad [7]$$

The error percentage between simulation result and analytical value is,

$$Err\% = 1 - (1874 \div 3380.79) = .44;\ 44\% \quad [8]$$

3. CONCLUSION

a) In this present work contact pressure measured by simulation is much less than the analytical value. The Hertzian contact theory does not include the effect of relative motion between the sphere and the flat surface. In case of sliding friction, the contact between pin and disk may not be continuous as at microscopic level the asperities of hemispherical pin and flat disk frequently getting in contact and also remain intact instantaneously. So the maximum contact pressure in dynamic condition is not as high as static.

b) The viscous drag force's effect has been applied to the pin according to the assumptions of Stokes' theory of lubrication. As viscous drag does not depend upon the load on the pin and the magnitude of viscous force is very low in case of significant mechanical load condition the lubricant's effect on the contact pressure as well as wear of the pin is very less in mixed lubrication regime. Further, the scope of improvement in fluid mechanics theory of lubrication to understand the viscosity's effect in better way.

c) As very simple model has been used and the element size taken is relatively bigger the simulation result is more applicable for comparative study of different designs instead of predicting absolute wear life.

d) Further improvement in such simulation is very much expected as finer mashing and increased simulation time always remit better result. While defining material property if elasticity can be defined properly might increase the accuracy of such finite element analysis.

REFERENCES

Benabdallah H, Olender D, Finite element simulation of wear of polyoxymethylene in pin-on-disk configuration, Wear, 261, 2006, 1213-1224.

Podra P, Andersson S, Simulating sliding wear with finite element method, Tribology International, 32, 1999, 71–81

Shen X, Cao L, Li R, Numerical Simulation of Sliding Wear Based on Archard Model, Proceedings of International Conference on Mechanic Automation and Control Engineering (MACE), Wuhan, 2010, 325-329.

Som SK, and Biswas G, Introduction to fluid mechanics and fluid machines, TATA McGrawhill Education (India) private limited, 3rd edition, 2011.

Stachowiak GW, Batchelor AW, Engineering Tribology, Butterworth-Heinemann, 4th Edition, 2013.