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FINITE ELEMENT STUDY OF EFFECT OF ORIENTATION OF BODIES ON CONTACT STRESS

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ABSTRACT

Engineering surfaces can be characterized as more or less randomly rough. Contact between engineering surfaces is thus discontinuous and the real area of contact is a small fraction of the nominal contact area. The contact is necessary for any engineering applications to transfer the force and power hence it is indispensible field of study. A contact model is developed for determining the effect of orientation on contact stress. A contact model that takes the properties of engineering surfaces into account has been developed and implemented using finite element software. The results obtained from the Ansys software are compared with Greenwood and Hertz theoretical results. The results show that the contact stress is changing with orientations. The work shows that the orientation effect on the contact stress is significant.

Key words: *Finite Element Method, Surface Roughness and Contact Stress*

1. Introduction

Engineering surfaces can be characterized as more or less randomly rough. Contact between engineering surfaces is thus discontinuous and the real area of contact is a small fraction of the nominal contact area [1]. Microscopic and macroscopic irregularities are present in all practical solid surfaces. Surface roughness is a measure of the microscopic irregularity, whereas the macroscopic errors of form include flatness deviations, waviness and for cylindrical surfaces, out of roundness. Two solid surfaces apparently in contact, therefore, touch each other only at a few individual spots as in figure 1

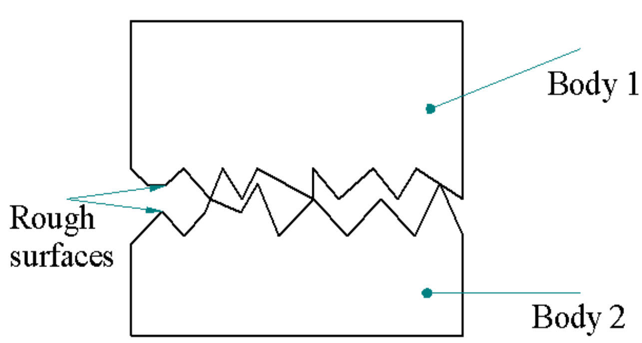

Fig.1 Two Surfaces in Contact

It is now well recognized that bearing surfaces are microscopically rough. Thus high spots on the contacting surfaces can directly contact each other, deforming elastically and plastically with critical consequences to the fatigue life, friction and wear behavior of the bearings of which they are a part. When real surfaces are pressed together they touch at a large number of high spots which deform elastically or plastically to form micro contact areas.

The sum of the micro contact areas is ordinarily a small fraction of the nominal or apparent area over which the two bodies contact and therefore the pressure at these micro contacts is high. It is natural, therefore, to expect that the severity of wear and surface fatigue are related to what happens at these micro contacts. In early studies of real surfaces it was assumed that the deformation at micro contacts must be plastic because of the very high stresses they support [2].

Archard showed, however that if a surface was made up of large number of irregularities approximated by spheres, onto which were superimposed a smaller set of spheres which in turn supported yet a smaller set, the aggregate relationship between load and area approached linearity, even though individual spheres deformed non-linearly. Greenwood and Hertz theoretical methods are used to study the relations between different surface parameters and to compare and verify results obtained with the FE-based surface model. In the present study, Greenwood and Williamson model is used for finding the real area of contact. The model is applied appropriately to consider the effect orientation on contact stress.

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1.1 Literature review

John I. McCool [3] studied the contact area and contact stress and compared with different models. Like GW and BGT models. Greenwood and Williamson [6] developed one of the first models for the contact of real surfaces that specifically accounted for the random nature of contact of two nominally flat surfaces. The model applies to the contact of two nominally flat elastic surfaces, one of which is rough and the other of which is smooth. It is readily adapted to the case of two rough surfaces. In the GW model, the rough surface is presumed to be covered with local high spots or asperities whose summits are all spherical in shape. All the summits are presumed to have the same radius *R*, but the summit heights are randomly variable. Summits are presumed to be uniformly distributed over the rough surface with a known density Dsum of summits per unit area. The mean height of summits lies above the mean height of the surface as a whole by the amount Z_{SMEAN}. The ratio A_C/A_O of the real contact area to the apparent area. The first satisfactory analysis of the stresses at the contact of two elastic solids is due to Hertz (1882) [7]. He was studying Newton's optical interference fringes in the gap between two glass lenses and concerned at the possible influence of elastic deformation of the surface of the lenses due to the contact pressure between them. In addition to static loading he also investigated quasi static impacts of spheres. Results in this field have since been extended to all branches of engineering, but are most essential in the study of tribology and indentation hardness. Hertzian contact stress refers to the localized stresses that develop as two curved surfaces come in contact and deform slightly under the imposed loads. This amount of deformation is dependent on the modulus of elasticity of the material in contact.

J.F. Archard [3] pointed out that plastic deformation could not be universal rule, and introduced a model which showed contrary to earlier ideas. The area of contact could be proportional to the load even with purely elastic contact. The theory developed leads to a set of relations which gives the total real area of contact, the number of micro contacts, the load, and the conductance between two surfaces in terms of the separation of their mean planes. Whereas the separation depends on the nominal pressure (that is, the load divided by the nominal area of contact), the number of micro contacts and the total area of contact depend on the load only. The separation is not very sensitive to the pressure: in fact the mean planes of two similar surfaces in contact are usually separated by 1 to 2 times the standard deviation, or roughly by the center line average. This means that the average gap between 20 microns. Surface is, for a wide range of loads, approximately 20microns. This explains the difficulty of making metal to metal gasket seals.

1.2 Objectives and methodology

The main aim of present research work is to study and suggest a method to enhance contact area and in turn contact stress between a given set of work pieces at low loads. The objectives include the study of effect of orientation, distance between planes on contact area and contact stress. The objectives of the present research work are listed as

i. To study and quantify the effect of orientation on contact area and in turn on contact stress.

The objective is elaborated in the following sections

1.2.1 Effect of orientation on contact stress

The most important parameter influencing real area of contact between any two surfaces is surface roughness of two contacting surfaces. Many techniques are being employed to measure and express surface roughness. Number of contact mechanics theories such as GW, BGT, or modified GW models available in literature can be used to determine the real area of contact between two bodies. GW model which is widely used in contact area calculations by consider the entire surface either as isotropic or anisotropic. It is very difficult to produce entire surface of anybody with the same surface roughness parameters (average roughness, root mean square roughness, standard deviation of peak heights, density of peaks etc). Surfaces are produced as combination of patches with different roughness value as shown in figure 2 and 3.

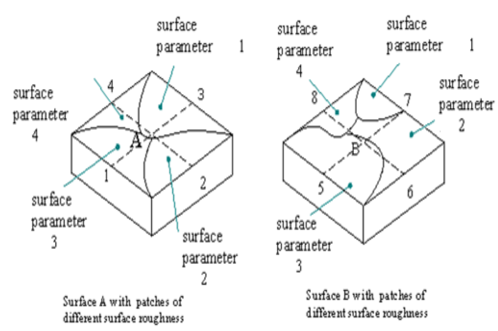

Fig. 2 Two Surfaces with Patches of Differing Surface Roughness and Division of Surfaces into Quadrants

With reference to figure 2 when surface A is required to be in contact with surface B, the contact can be realized in 4 different ways. These 4 different alignments are termed in the present work as 4 orientations.

i. Line 1-3 of surface A is aligned with line 5-7 of surface B. Point 1 is in contact with point 5 and point 3 in contact with point 7; correspondingly in this orientation line 4-2 is aligned with line 6-8.
ii. Line l-3 aligned with line 7-5 and correspondingly line 4-2 is aligned with line 8-6.
iii. Line 1-3 aligned with line 6-8
iv. Line 1-3 aligned with 8-6

In all these 4 orientations different patches of surface a come in contact with different patches of surface B. These results are slightly different equivalent surface parameters for each Orientation which leads to variation in real area of contact for different orientations. In the present work the effect of orientation on contact area and in turn on contact stress is studied. But intuitively it can be seen that when two surfaces are in contact depending upon which patch of one surface is in contact with the other patch on second surface the equivalent surface parameters change and corresponding contact area changes. In other words for same contacting surfaces, area of contact changes depending upon how the two surfaces are aligned or oriented with respect to each other.

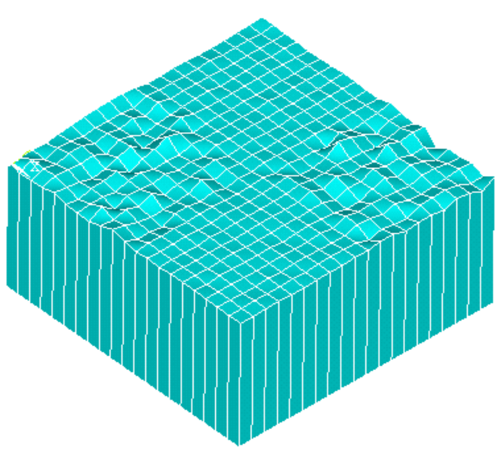

Fig. 3 A Rough Surface Having Different Patches of Roughness

1.3 Methodology

The methodology of project can be summarized in the flow chart as given below. Finite element study of effect of orientation and distance between planes are studied. In the present work APDL commands are used to create the program in the Ansys, which will generate solid model, meshing, create the contact pair and also solve for given boundary conditions.

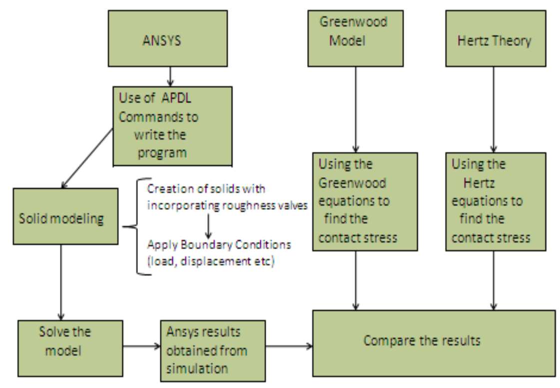

Fig. 4 Methodology

Greenwood model is used to find the contact stress for different orientations and different distance between mean planes. The finite element analysis, Greenwood and Hertz theoretical results are compared. The Greenwood and Hertz theoretical results are used to validation of finite element analysis results. The main aim of present research work is to study the effect of orientation on contact stress. It is important to note that, the objective is not to find the exact values of contact stress but to observe the variation of stress with orientation.

2. Theoretical Calculations

2.1 The greenwood-williamson model

In the present study, Greenwood and Williamson model is used to find the real area of contact and contact stress. The model is applied appropriately to consider the orientation effect.

Load, P = Pressure*Total area …… (1)

$$\frac{Ac}{Ao} = \Pi\ R\ \sigma s\ Dsum\ F1\ (\frac{d}{\sigma s})$$

$$Ac = (\ Ao)\ *(Avg)$$

$$Po = \frac{P}{Ac} \quad …….. (2)$$

2.2 Hertz theory

The Hertz equations used to find contact area and contact stress.

Load, P = Pressure*Total area

n = no of peaks in contact

$$n = D_{SUM}\ F_O\ (d/\sigma_S) \quad ……. (3)$$

$$P1 = \frac{P}{n}$$

$$a = \left(\frac{3PR}{4E*}\right)^{1/3} \quad …….. (4)$$

$$Po = \left(\frac{3P}{2\pi a2}\right)^{1/3} \quad …… (5)$$

3. Finite Element Analysis

3.1 Simulation procedure

Generation of rough surface is the most difficult part in the modeling. The surface roughness parameter is in microns whereas the geometric nominal dimensions are in millimeter. The different scales pose a great challenge in modeling a volume with nominal dimensions as well as incorporating surface roughness. Two solid bodies are created with different surface parameters on same surface. This variation in surface parameter is included in the modeling at the generation of key point level. Steps involved are

i. Creation of random locations in 3 dimensional space
ii. Creation of key points with the help of created random locations**.**
iii. Creation of areas by using the key points.
iv. Extruding areas to get the volumes and creation of blocks. Further in this step only subtraction of block from extruded volumes is done to get one side flat and another side rough surface.
v. Creation of spheres on the key points: This is also one of the important steps in model generation. In this step 441 spheres are created on 441 key points of the volume.

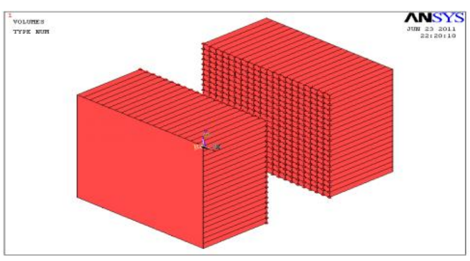

Fig. 5 Creation of Spheres on the Key Points

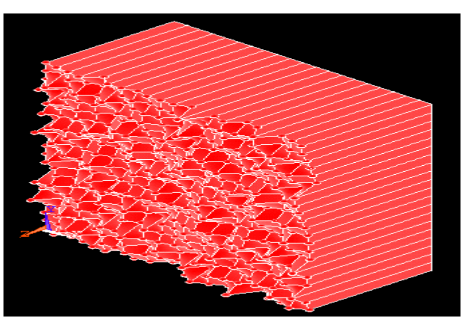

Fig. 6 Spheres on the Summit

For representation purpose the above figure 6 is shown, in this all the spheres are created on the summits. The above five steps are of main importance and these are done with the help of Ansys Parametric Design Language (APDL) programming, which are listed in further chapters.

vi. Assigning the material properties: In this step various material properties such as young's modulus (E) and Poisson's ratio (P) along with material numbers are assigned for solid model with the help of APDL programme.
vii. Meshing the model with structural solid element: In this step solid models are divided into number of elements. A three dimensional solid element Tet 10 node 187 is applied to mesh the models, 331053 elements and 488279 nodes are generated.

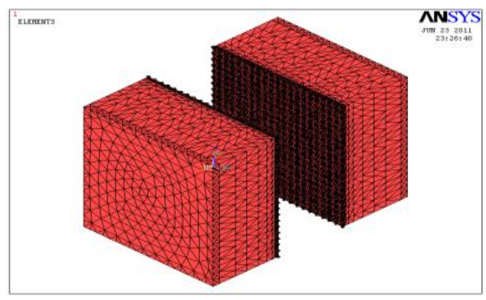

Fig.7 Meshing of volumes

viii. Contact elements are created at both the surfaces of bodies where contact is going to occur.
ix. One body is moved towards the other to achieve the contact pair between two bodies with different mean plane separation values.

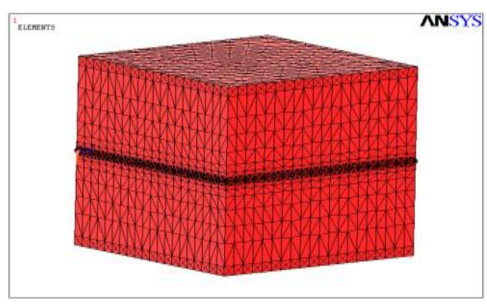

Fig. 8 Two bodies in contact with each other

x. Apply various boundary conditions: In this step various boundary conditions such as pressure and displacements etc. are applied.
xi. Solve for stress distribution in the contacting bodies.
xii. The process is repeated for contact pairs with different orientations

3.2 Material and modeling data

Specifications (all dimensions in mm)

Body………20x20x10

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Summit radius…….. 0.2, 0.15

Material ………………. Steel 0.2% CHR

Young's modulus……… 200x103 N/mm^2

Passions ratio……………0.3

Pressure applied………... 5x10^{-3} N/mm^2

Distance between mean plane separations

D1 = -9.9995 and D2 = -9.9996

Table 1: Roughness Values for All Four Quadrants of Both the Sets

Roughness	Set1	Set2
σ_1	0.000204	0.0001785
σ_2	0.000187	0.0001751
σ_3	0.000153	0.0001649
σ_4	0.000136	0.0001615

3.3 Software related data

Element: Solid 187, Contact Element: CONTA 174, Target Element: TARGE 170, Solver: PCG, Analysis type: Steady state

4. Results and Discussion

The theoretical and finite element analyses are carried out. Whenever one body is in contact with other body with different orientations, the variations in stress distribution are calculated. The results obtained from the orientation and distance between mean planes is discussed in this chapter.

4.1 The effect of orientation on contact stress

Finite element analysis of effect of orientation of two nominally flat surfaces of contact area and contact stress is studied. It is observed that even though same set of work pieces are in contact, the contact area changes for different orientations, once the contact area changes, the contact stress is also changes and this result in the variation of stress distribution in work pieces. A stress distribution plot of same set of work pieces in contact with orientation is shown in figure 10. The results for surface parameter set 1 and surface parameter set 2 are discussed here.

Set 1:

Orientation (1_5)

The above body has quadrants 5,6,7,8 and below body has quadrants 1,2,3,4. In this orientation, the quadrant number 1 come in contact with quadrant number 5 which means 1 contact with 5 and remaining quadrants will contact with their pairs. The distance between planes 0.0005 mm and the summit radius 0.2mm are used for this orientation.

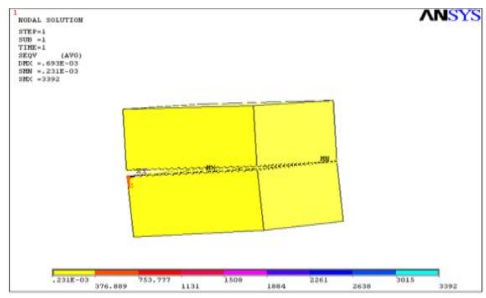
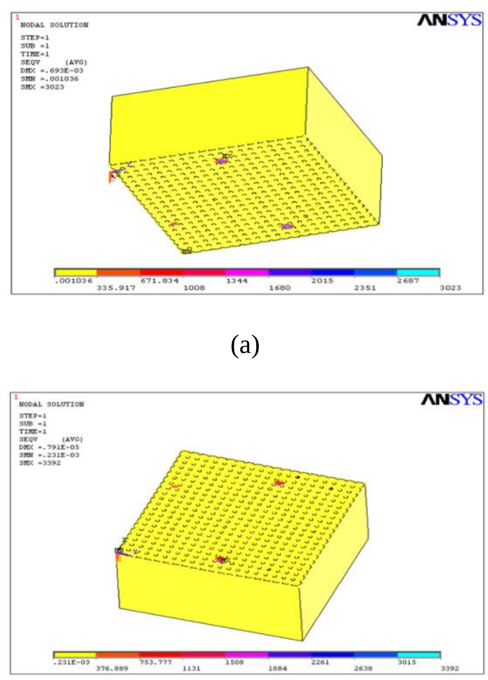


Fig. 9 Stress Distribution for Orientation 1_5, set1 and Distance between Planes 0.0005 mm and Summit Radius 0.2mm



(a)

(b)

Fig. 10 The upper work piece (a) and lower work piece (b) of 1_5 set

Orientation (1_7)

In this orientation, the quadrant number 1 contact with quadrant number 7 which means 1 contact with 7 and remaining quadrants will contact with their pairs. The distance between planes 0.0005 mm and the summit radius 0.2mm are used for this orientation.

Figure 11 shows the variation of contact stress with orientation. Essentially this figure indicates the variation of stress with orientation for Ansys, Greenwood, and Hertz results. All the Contact stresses are in decreasing order for the distance between planes 0.0005 mm and summit radius 0.2mm.

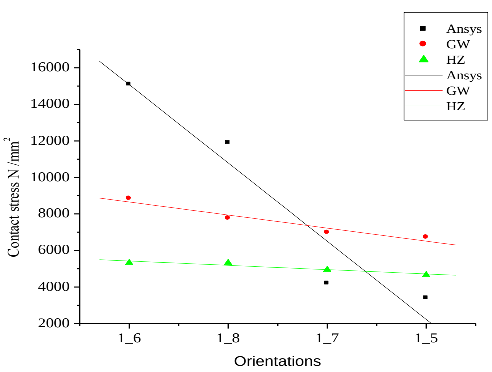


Fig. 11 Variation of Contact Stress for Orientations and Distance between Planes 0.0005mm and Radius 0.2mm

Orientations (1_5, 1_7 with summit radius 0.15)

The surface parameter of set 1 is studied with different summit radius. In these orientations the summit radius is reduced to 0.2 mm to 0.15 mm and then simulation to find the contact stress Stress distribution for orientations 1_5,1_7,1_8 and1_6, set 1 and distance between planes 0.0005 mm and summit radius 0.15mm.

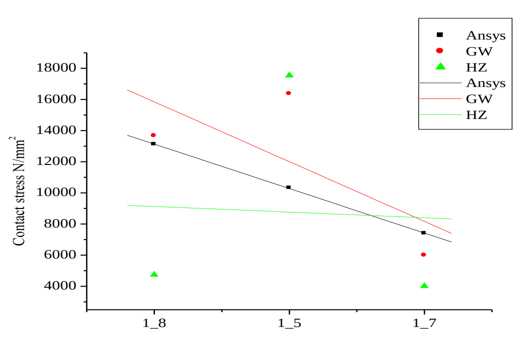


Fig. 12 Variation of Contact Stress for Orientations and Distance between Planes 0.0005 mm and Summit Radius 0.15 mm

In these orientations, the surface parameter of set 1 is studied with different summit radius. The result of both the orientations shows that, even though the radius of the summits is changes, the contact stress is also changes. Figure 12 shows the variation of contact stress with orientation. This figure indicates the variation of stress with orientation for Ansys, Greenwood, and Hertz results. Here also all the contact stresses are in decreasing order for distance between planes 0.0005 mm and summit radius 0.15mm.

Set 2:

Orientation (1_5)

In this orientation, the quadrant number 1 contact with quadrant number 5 means 1 contact with 5 and the remaining all quadrants contact with their pairs. The distance between planes 0.0005 mm and summit radius 0.2mm are used for this orientation.

Orientation (1_7)

In this orientation, the quadrant number 1 contact with quadrant number 7 means 1 contact with 7 and the remaining all quadrants contact with their pairs. The distance between planes 0.0005 mm and the summit radius 0.2 mm is used for this orientation.

Figure 13 shows that, the variation of contact stress with orientation. This figure indicates the variation of stress with orientation for Ansys, Greenwood, and Hertz results. All the contact stresses are in decreasing order and distance between planes 0.0005mm and summit radius 0.2mm.

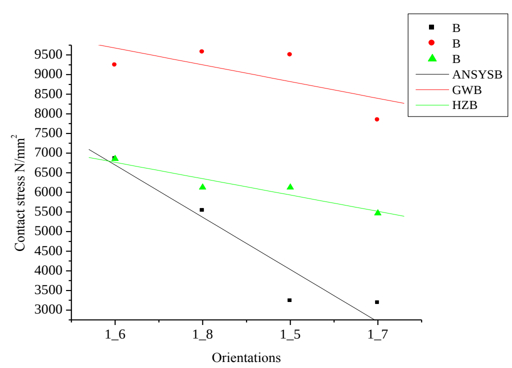


Fig. 13 Variation of Contact Stress Orientations and Distance between Planes 0.0005mm and Summit Radius0.2mm

Orientations (1-5, 1-7, with summit radius 0.15)

In these orientations, the surface parameter of set 2 is studied with different summit radius. The result of both the orientations shows that, even though the radius of the summits changes, the contact stresses also changes. Figure 14 show the variation of contact stress with orientation. This figure indicates the variation of stress with orientation for Ansys, Greenwood, and Hertz results. All the contact stresses decreasing order for distance between planes 0.0005mm and radius 0.15mm

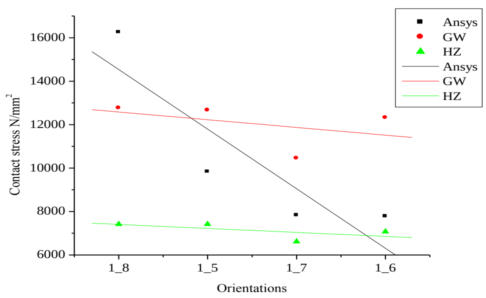


Fig. 14 Variation of Contact Area for Orientations and Distance between Planes 0.0005mm and Summit Radius 0.15mm

5. Conclusion

The main objective of present research work has been to study the effect of orientation and the distance between surfaces on contact stress. The inclusion of surface roughness in the finite element method was the most difficult part of the project. Substantial effort has been put to consider the realistic surface model. The finite element study of effect of orientation and distance between planes on contact stress is carried out. Different sets of surface parameters are considered for the analysis. The result of finite element analysis shows that, the orientation has the significant effect on contact stress. The theoretical study of the research objectives is carried out using Greenwood and Hertz models. This analysis indicates that, the contact stresses between a pair of bodies changes with orientations. The results of Greenwood and Hertz model are compared with those of finite element analysis. The trends of theoretical and finite element analysis results are same. The effect of orientation can plays an important role in optimizing contact stresses in many applications like electronic chip assembly. The practical significance of the present research outcomes would be effectively applicable where contact stress plays an important role as in aircraft structural, electronic chip assembly, bolted and riveted joints, structural joints of machine tools, joints and other applications.

Scope of Future Work

i. Better surface model can be produce.
ii. Change density of peaks can incorporate in the model.
iii. Mesh refinement can be checked.
iv. Simulation can be extended by considering the effect of self weight of bodies and the application of external loads. Coupled field analysis can be carried out with structural and thermal analysis.

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Nomenclature

Symbol	Meaning	Unit
A_a	Apparent contact area	m^2
A_r	Real contact area	m^2
E	Elastic modulus	MPa
P	Load	N

