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Arunachalam, Adhithya Plato Sidharth; Idapalapati, Sridhar

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Material removal analysis for compliant polishing tool using adaptive meshing technique and Archard wear model

Adhithya Plato Sidharth Arunachalam and Sridhar Idapalapati*

*School of Mechanical & Aerospace Engineering, Nanyang Technological University,
50 Nanyang Avenue, Singapore 639798, Singapore*

Abstract

In this paper, a simulation technique to predict the material removal profile is developed for a disc-shaped compliant polishing tool which is commonly used in robotic polishing. The methodology is based on the Archard wear model implemented with adaptive meshing technique in the commercial finite element ABAQUS® software. Initially, the effect of tool compliance on the static contact pressure distribution is investigated experimentally using pressure films. Numerical 3D **finite element** model is developed for the same in order to predict the contact pressure distribution which in turn influences the material removal profile prediction. The material removal study is carried out with a robotic arm, and the polished surface is later scanned for the material removal profile. In order to predict the material removal profile, the **finite element** simulation in ABAQUS® is carried out using ‘dynamic-implicit’, followed by executing the umeshmotion FORTRAN subroutine in the ‘general-static’, where the nodes are displaced based on the wear model and using Arbitrary Lagrangian-Eulerian (ALE). The results are again imported in dynamic implicit and the simulation is restarted. The cycle continues till the experimental polishing time is reached. The experimental and simulation results of contact pressure are in good agreement with each other and bring out the effect of tool compliance on dynamic pressure which in turns affects the overall three-dimensional material removal profile.

Keywords: Material removal; compliant polishing tool; Archard wear; Contact pressure; Adaptive-Lagrange Euler meshing (ALE); ABAQUS® umeshmotion subroutine

1 Introduction

In recent days, the manual metal polishing is being replaced by robots for better quality control, reduced process time and cost, and reduced rework with process control [1]. Further, the compliant tools are preferred in the robot polishing process for controlled material removal, better conformability to the polishing surface and helps in blending of surfaces [2]. These tools are used widely for a long time, not only in polishing dies and spare parts in the aerospace and automobile industries but also in polishing knee implants in bio-medical industries too [3]. The coated abrasive disc is a type of compliant tool which is used widely in the **above-mentioned** industries, apart from abrasive belts and flap wheels.

* Corresponding author: Sridhar Idapalapati (msridhar@ntu.edu.sg)

These abrasive discs are usually supported by a rubber backing pad and which in turn is attached to the spindle. The spindle is held to the robot arm, and the force-controlled robot is programmed to trace the contour of the workpiece. According to the amount of material removal required, the force and the spindle speed values are adjusted. The selection of the crucial machining factors like force and the tool compliance is based on the operator's experience and the previous experimental trials data. Hence to reduce the ambiguity in tools and parameter selection, it is necessary to investigate the effect of these parameters on the material removal distribution both experimentally and numerically for optimization purposes.

Researchers proposed different modelling techniques to predict the material removal in fixed abrasive polishing processes. In the macro-scale approach, the contact pressure distribution between the tool and workpiece was obtained either through analytical methods like Hertzian contact model [4,5], or experimentally [6]. Later, implementing Archard wear law, the material removal profile is obtained. The advantage of this method is that it does not consider the micro-level interactions between the grains and the workpiece, therefore, the computational time required is less. The next type of modelling technique is a micro-scale technique, in which the abrasive grains are modelled statistically using the distribution values of protrusion height and inter-grain spacing. Then, by using the contact models (elastic and plastic), the penetration depth of each grain and contact pressure are correlated. Then the material removal profile at a particular section is computed by integrating the material removed per unit contact length [7-9]. This kind of simulation technique allows investigating in detail, the grains workpiece interactions and the effect of machining parameters. The other type of modelling technique is by using the Johnson-Cook material damage model in commercially available FEA software. Most of the studies in this research area were done to predict the material removal profile in grinding. But because of high computational time, the analyses were restricted to study single grain scratch [10-12].

Zhang et al. [4] investigated the material removal in spherical and cylindrical shaped compliant polishing tool on concave and convex shaped geometries. The Hertzian contact model was used where the contact area was considered to be elliptical in shape. The material removal profile was derived by using Archard wear equation based on the computed contact pressure distribution. This model was helpful to understand the effect of tool and workpiece curvature along with the machining parameters (force, spindle speed, feed etc.) on the material removal profile. A similar study was carried out by Wang et al. [5] in belt grinding study, where the contact area between the tool and surface was also modelled as elliptical, and the material removal rate was derived using factors like the pressure distribution and the tool dwell time (which is, in turn, a function of feed). Another method, where the experimental dynamic pressure data was used along with Archard wear to predict the material removal profile in belt grinding process by Sun et al. [6]. Different wheel types such as flat and serrated contact wheels were taken to demonstrate the effect of dynamic pressure on the material removal profiles in both flat and convex work coupons at different load ranges.

Archard wear model with UMESHMOTION in commercial FE software ABAQUS® was used to predict wear in tribosystems. Bortoletto et al. [13] implemented **Augmented Lagrangian Eulerian** (ALE) to model wear in unlubricated sliding wear occurring in pin on

disc tribotests. A global wear coefficient was used in the numerical model to predict wear for different load ranges. The model predicted wear loss for different lower load ranges. Rezai et al. [14] implemented a similar approach in predicting wear in radial sliding bearings by a two-dimensional plane strain FE model. For a fixed sliding velocity, the wear depth for different normal pressure is compared between experimental and simulation. Similarly, a wear model in ABAQUS® to simulate wear using the subroutine in polymer–metal contact pair was proposed by Martinez et al. [15]. In order to simulate the entire sliding distance, an acceleration factor was used where the existing equation is multiplied with this factor at regular intervals to replicate the entire sliding distance.

Apart from the existing works as discussed above, a macro-scale modelling technique to compute the material removal using compliant polishing tool with abrasive discs is discussed in this paper. In order to implement Archard wear, umeshmotion subroutine is coded using FORTRAN in ABAQUS®. A similar approach was carried out to predict wear in bearings, polymers, brake pad etc., In these approaches, based on the numerical contact pressure and slip values at the interface, the geometry of interest is modified continuously by Adaptive-Lagrange Euler (ALE) meshing. Hegadekatte et al. [16] implemented FE based Archard wear model using UMESHMOTION to predict wear in tribosystems.

In this paper, **Adaptive-Lagrange Euler (ALE)** is implemented to investigate the three-dimensional material removal profile in compliant polishing tools by using UMESHMOTION subroutine as described in the flowchart (Figure 1). In the initial study to justify the model and to understand the tool behaviour, the contact pressure between the tool and workpiece is found experimentally and compared with the developed FE model. The load-displacement curves were matched to validate the models. Later, in the FE analysis, the dynamic contact pressure from the workpiece surface is computed, and by implementing ALE, the nodes are displaced according to the Archard abrasive wear model. Then the displaced nodes are re-meshed, and the process continues till the end of machining time is reached. The material removed profile from FEA result is compared with the experimental profile measured from profilometer.

Figure 1 Overall methodology for material removal profile computation

2 Experiment procedure

2.1 Static pressure distribution study

In this study, the key focus is to investigate the effect of contact pressure by the compliant tool on the material removal profile. There are two kinds of static experiments conducted as mentioned in Figure 1.

- (i) Initially, the pressure distribution using only the two different rubber backing tools (hard and soft) are carried out. No abrasive disc is attached to the rubber backing pads.
- (ii) Further, to understand the effect of an abrasive disc on the static pressure distribution, abrasive pad without grains is considered. Since the pressure films are highly sensitive and prone to get damaged by sharp and hard objects, the grains are removed from the abrasive disc.

In order to study the contact pressure distribution, pressure films are used as shown in Figure 2. The pressure films are placed in the interface between the tool and the workpiece. Once the tool is pressed against the films, color of different intensities is formed in correlation to the contact pressure. Using the image processing software, the color intensities are later translated to pressure values. The films are suitable to measure static pressure and highly sensitive to heat and rough environment. In addition, the load-displacement curves of both the hard and soft rubber tools are obtained in order to validate the numerical model.

Figure 2 (a) Robotic polishing setup with pressure films and backing pad (b) Color formation on pressure films when hard rubber pad pressed against the aluminium plate.

2.2 Material removal study

In this study, 3M™ ROLOC™ rubber pads are used to hold the abrasive discs. Two types of tool compliance such as hard and soft rubber are taken. The abrasive disc used is #60 grit size. These discs are attached to the bottom of the rubber tool which is in-turn held in the spindle which is attached to the robot arm. 6-axis ABB IRB 140 robot is used in this study as shown in Figure 3(a). Aluminium (Al6061-T6) workpiece is a 3 mm thick rectangular plate of dimensions 152 mm x 90 mm.

Figure 3 (a) Material removal tests with abrasive discs (b) 3D surface profile measurement using profilometer

Although usually titanium and nickel alloys are preferred to be polished, aluminium is considered because the key focus of the study is to develop a simulation technique to predict the material removal profile for the disc-shaped compliant tool. The spindle speed is 2500 RPM and the force applied is 20N. A tilt angle of 15° is considered to facilitate tool flexing and to avoid zero velocity at the tool centre [17,18]. The rotating tool is pressed against the workpiece for 3 seconds without feed motion. Later the tool is retracted, and the machined surface is scanned using Talyscan 150® profilometer which has both inductive and laser gauge. In this study, laser gauge (non-contact probe with laser triangulation technique) is used to measure the machined surfaces with the spacing of $100\text{ }\mu\text{m}$ (in both lateral and vertical directions) as shown in Figure 3(b). The vertical accuracy of the laser probe is $1\text{ }\mu\text{m}$.

3 Numerical simulation of contact pressure distribution

3.1 Tool and workpiece modelling

The beforementioned two experimental static pressure distribution experimental scenarios (i) rubber backing pad- aluminium plate interaction and (ii) rubber backing pad along with grainless disc – aluminium plate interactions are investigated using FE simulations. For this scenario, only the rubber pad along with the shaft is modelled and kept at a tilt angle of 15° with respect to the aluminium workpiece as shown in Figure 4(a) (say Model A). The internal surface of the rubber pad is tied to the shaft. The bottom surface of the rubber pad and the top surface of the workpiece is made as slave and master contact surfaces respectively in Model A. The shaft is modelled as a rigid body with a reference point attached to it for applying the load and boundary conditions. Once the initial validation study is performed for Model A, it is further extended by modelling an additional disc of 1.1 mm thickness and 50.8 mm diameter to represent the grainless abrasive pad. The top surface of the grainless abrasive disc is tied to the bottom surface of the rubber pad. This simulation case is referred to as Model B. The bottom surface of the disc is made to contact the top surface of the inclined workpiece as shown in Figure 4(b). Additional factors like grains and surface undulations on the workpiece are ignored in this study, as the main focus lies in understanding the effect of compliance on pressure distribution and material removal.

3.2 Material models

Hyperelastic constitutive model is used for the hard and soft rubber pads as they undergo instantaneous elastic response up to large strains. In this material model, the mechanical response of a material is defined by choosing strain energy potential to fit the particular rubber material. In ABAQUS®, the material coefficients can be either directly given as input or provide the experimental tensile data and the software computes the coefficients values by itself. Out of all the models available such as Mooney Rivlin, Ogden, Yeoh, Neo-Hookean etc., the Yeoh model is selected by fitting the experimental uniaxial tensile stress-strain responses. The Yeoh model is equivalent to using the reduced polynomial model with $N=3$, where N is the value assigned for strain energy potential order. The details regarding the Yeoh hyperelastic constitutive model can be found elsewhere [19]. The Al6061-T6 workpiece is modelled as a linear elastic solid with Young's modulus E of 68.9 GPa and Poisson's ratio ν of 0.3. Since in this study, the contact pressure difference is the key parameter of interest, and there are no grains involved in the simulation, the stress values at the surface will be far below the yield value of Al6061-T6, and hence the workpiece can be assumed to be linearly elastic without considering the plastic material properties. The grainless abrasive pad is made of polyester/cotton fabric with phenolic resin as matrix. The woven fabric is modelled as orthotropic lamina [20]. The Young's modulus values (E_1 and E_2) are obtained by performing three-point tests on the flexural samples cut in 0° and 90° directions of the disc respectively as depicted in Appendix Figure A.1 (a) and (b). The span length of these specimen is 40 mm and thickness of the samples 1.1mm. The measured material properties from the three-point bending test are as listed in Appendix Table A.1.

Figure 4 Rubber pad and aluminium block (Model A), (b) Composite abrasive disc attached to rubber pad (Model B) and ply stack orientations

3.3 Mesh and boundary conditions

The rubber backing pads are hyperelastic incompressible in nature. Therefore, they are modelled with 10-node modified hybrid, quadratic tetrahedron elements with hourglass control (C3D10MH). In order to capture the incompressible and highly non-linear deformation of the rubber material and also to model contact more accurately, C3D10MH elements are used. Mesh convergence study is done for different mesh sizes (0.5mm, 1mm,

2mm) for the rubber pad and the aluminium block. The contact pressure values at a single node are obtained for the different mesh sizes from the simulation results. It is found that the 1mm mesh size is more suitable for both rubber pad and aluminium block based on the computational time, the pressure value and the stress singularities. The boundary condition is applied to the created reference point on the rigid shaft which is tied concentrically to the internal surface of the rubber pad. For the boundary condition, the spatial displacement along the loading direction (U_2) is left unconstrained and constraining all other degrees of freedom. The aluminium block is meshed using 3D linear brick element, with reduced integration (C3D8R) elements and the base of the aluminium block is fully constrained (made 6 dofs=0). An 8-node quadrilateral in-plane general-purpose continuum shell elements (SC8R) is used to mesh the grainless abrasive pad to capture the membrane stretching behaviour. Only one layer of mesh is used through the thickness, as each element in the through-thickness direction represents the stack of plies (0° and 90°).

4 Wear model implementation using adaptive FE modelling

The flowchart in Figure 5 summarizes the simulation of material removal using Archard's wear approach. For implementing the **Adaptive-Lagrange Euler (ALE)** procedure, the aluminium part is selected as the ALE domain, and the aluminium surface nodes are selected as ALE node set. Movement of nodes replicating wear phenomena is implemented in these nodes by wear equation. In the present work, the Archard abrasive wear model is used to estimate the material removal rate at each node. The volume (V) of the material removed under a normal force, ' F ', over a sliding distance ' s ' is given by Eq.1[21]

$$V = k \frac{F}{H} s \quad (1)$$

where ' H ' is the hardness of the material (N/mm^2), and ' k ' is the wear coefficient. Expressing the Eq 1 in an infinitesimal form with respect to the sliding distance at a node ' i ', Eq 2 is obtained [21] .

$$dh_i = \frac{k}{H} \cdot \bar{p}_i \cdot ds_i \quad (2)$$

where $\bar{p}_i$ is the averaged contact pressure, ' ds ' is the incremental sliding distance and ' dh ' is defined as the incremental change in the surface node position due to wear in the surface during the time period ' Δt ' (0.3 seconds).

In order to reduce the computational time, the MRR results are scaled such that the total machining time and number of revolutions can be reached using the Eq 3. Similar acceleration factors were employed in earlier studies [22,15,23] governed by the justification that the wear coefficient considered being a constant

$$h_{ij} = \sum_{j=1}^{j=itr} \frac{fk}{H} \bar{p}_{ij} (\Delta s_{ij}) \quad (3)$$

where, ' f ' is the scaling factor ($f=5$, hence, simulation is restricted to 2 cycles)

‘ k ’ is the wear coefficient (0.0753 and 0.0636 for hard and soft respectively) calculated from the experimental measurements. The average material removed for hard and soft polishing pads are 67.342 mg and 51.735 mg respectively, when machined with 20N polishing load and 2500 RPM, and converted to volume (V) by multiplying with Al material density of 2700 kg/m³. The average sliding distance ($s_H=17.373$ m, $s_S=15.802$ m) is calculated from the product of scratch average radius ($r_H=22.12$ mm, $r_S=20.12$ mm), rotational speed ($\omega=261.8$ rad/s) and time ($t=3$ s). The average radius is obtained by taking the mean of outer radius (25mm) and inner radius (25mm-scratch width, which can be obtained from Figure 10 (a) and Figure 12(a)). The wear coefficient value is obtained by substituting these values in Eq 1.

h_{ij} is the material removal depth computed at node ‘ i ’ during the iteration ‘ j ’

$\overline{p_{ij}}$ is the averaged pressure at particular node ‘ i ’ during the iteration ‘ j ’

Δs_{ij} is the sliding distance difference obtained from the final frame and initial frame within the same iteration ‘ j ’.

$$itr = \frac{T}{f * \Delta t} \quad (4)$$

where ‘ T ’ (Total machining time) =3 seconds, $\Delta t=0.3$ seconds. Substituting above values in Eq.4, $itr=2$ is obtained, where each iteration ‘ j ’ is 0.3 seconds (Δt).

Figure 5 Algorithm for implementing UMESHMOTION subroutine to calculate 3D material removal profile

The nodes in the contact surface between the tool and the workpiece is created as an adaptive mesh node set. Later, by using the FORTRAN user subroutine, the nodes on the contact surface are modified based on the Eq. (3) as shown in Figure 5. UMESHMOTION subroutine in ABAQUS® is only available in 'General-Static' simulation and not in explicit. Since the model involves contact and highly non-linear deformations and inertial effects, dynamic-implicit is more suitable, but as mentioned, **Adaptive-Lagrange Euler (ALE)** is not presently supported in dynamic-implicit. To overcome this hindrance, the simulation is run in dynamic-implicit until step time ($\Delta t = 0.3s$) in which the tool makes ' Δn ' revolutions ($\Delta n = 12.5$). The dynamic pressure is averaged within the revolution time of ' Δt '. The average pressure and sliding distance range is calculated using python script and fed as input for the

contact pressure in the subroutine for the respective nodes. In order to simulate the complete machining time ' T ', the results are linearly extrapolated such that the wear depth (dh) is multiplied using scaling factor ' f ' according to Eq.4.4. The simulation is again restarted in 'General-Static', where UMESHMOTION subroutine is applied, and the mesh is displaced according to the Archard wear depth calculation. The purpose of these steps in General-Static is only to update the mesh. The nodal displacement values are added with the nodal displacement ($U3$) from the previous step. This cycle is repeated until the set ' itr ' number is reached, and the finally displaced ALE nodes can be compared with the experimental material removal profile.

5 Results and discussion

5.1 Static pressure distribution test without abrasive disc

The simulated contact pressure distribution is shown in Figure 6 (b) and (d), in comparison with the experimental measurements in Figure 6 (a) and (c). The simulated pressure distribution is obtained by considering the maximum pressure values from all the time frames (from 0 to 1 second), thus replicating the pressure film behaviour. Three parameters were considered to compare the contact pressure distribution viz., (i) maximum contact length is taken as the distance between the two extreme points of contact along the length of the aluminium plate, (ii) maximum contact width is considered as the distance between initial point of contact and the point where contact pressure drops to minimum value along the width of the plate and (iii) contact area which is calculated by multiplying the number of pixels with pressure values, multiplied by the pixel area (1 mm x1 mm in simulation and 0.125 mm x 0.125mm in experimental). The values of the above-discussed parameters for both hard and soft backing pads are listed in Table 1. The contact length of the soft and hard backing pad is almost the same (1.59% increase), but the contact width increased drastically by 108.7% for the soft backing pad. This can be related to the larger displacement observed in the load-displacement responses for soft backing pad compared to the hard backing pad, proving once again that compliance plays a crucial role in determining the contact area. The overall experimental contact area increased by 18.69% for the soft backing pad compared to the hard backing pad. The overall simulated contact pressure distribution values show good agreement with the experimental data as shown in Table 1. The deviation in the overall simulated contact area for hard rubber pad (186 mm²) and experimental contact area (185.32 mm²) is 0.366%. Similarly, the deviation in the soft rubber pad is 9.98%. Although the simulated and experimental contact area pattern matches, the deviations observed in the simulated and experimental contact area values can be attributed to the complex construction of the tool such as inbuilt cotton mesh at the bottom layer which is not considered in the simulation in order to simplify the model. Hence the load-displacement responses and the contact pressure distribution results prove that using the relevant material properties and contact conditions, the behaviour of the rubber backing pad (any backing material in a compliant tool) can be predicted and it helps to understand the overall behavior of the tool compliance on the contact pressure distribution. Since the results are agreeing, this FE model can be used for simulating case 'B' where an additional grainless abrasive disc will be added.

Figure 6 Experimental pressure distribution for (a) hard (c) soft rubber tool and FE pressure distribution for (b) hard and (d) soft tool

The simulated load-displacement response results as shown in Figures 7 (a) and (b) are compared with the experimental values, and there is a good agreement for both the considered scenarios within the material properties scatter. As observed from Figure 7(b), the response is initially linear for soft pad with disc till 2mm where as for the hard pad the entire behaviour is non-linear.

Figure 7 Load versus displacement response for (a) different pad compliance (rubber pad only) (b) rubber pad with grainless disc.

5.2 Static pressure distribution test with grainless abrasive disc

The experimental and simulated contact pressure distribution between two types of rubber pad with the disc are shown in Figure 8. The contact pressure follows a similar trend to that of the backing pad without the disc with increased value in response to the additional pad stiffness. The contact pressure in the case of the soft pad is more evenly distributed, where pressure values are scattered around the tool centre and near the rim of the rubber tool. Whereas, in the case of the hard pad with the disc, the concentrated pressure regions are observed around the outer periphery of the abrasive disc, which matches with the experimental trend. The contact length and width of the pressure distributions are listed in Table 1. The experimental contact length of the hard and soft pad with the disc is 30.38mm and 30.88 mm respectively. Although, the diameter of pad (50.8mm) is 5.1 mm larger than the rubber pad diameter (45.7mm), because of the additional abrasive disc stiffness there is a reduction in contact length. And the experimental contact width of the hard and soft pad with disc decreased by 10.75% and 2.08% which can be related to the reduction in the displacement (as observed from load-displacement responses) compared to the case of rubber pads alone. The experimental contact area of the soft pad with disc is 51.68% higher compared to that of hard pad with disc.

Figure 8 (a) Experimental, (b) numerical pressure distribution for hard pad at 20N (c) experimental (d) numerical pressure distribution for soft pad at 20N

Table 1 Experimental and simulated pressure distribution values for the backing pads

Contact results	Hard		Soft	
	Experimental	Simulation	Experimental	Simulation
<u>Rubber pad</u>				
Overall contact length (mm)	31.5	32.0	32.0	38.0
Overall contact width (mm)	5.75	7.0	12.0	14.0
Actual contact area (mm ²)	185.32	186	223.53	247
<u>Rubber pad with disc</u>				
Overall contact length (mm)	30.38	28.0	30.88	32.0
Overall contact width (mm)	5.13	6.0	11.75	9.5
Actual contact area (mm ²)	120.54	113	204.5	163

5.3 Wear simulation results

Initial pre-processing modelling steps like meshing, boundary conditions and assigning the material properties definition are carried out as discussed in section 3. Additionally, the rotational speed of 261.9 rad/s (2500 RPM) is given, and the **Adaptive-Lagrange Euler (ALE)** surface is selected. The average contact pressure computed using python code in the first and second iterations are shown in Figure 9 (a) and (b) respectively. Similarly, the sliding distance range is calculated as shown in Figure 9 (c) and (d). From these results, it is inferred that the assumption of having acceleration factor is justifiable as there is no considerable change (<5%) in the averaged values of the contact pressure and sliding distance. Moreover, because of the contact pressure singularities near the mesh periphery of the composite abrasive disc, high-pressure regions are observed as shown in Figure 9 (a) and (b). The material removal depth obtained in these nodes are nullified using appropriate threshold limit at the end of each iteration 'j' such that abnormal material removal depth values are avoided.

In between the two iterations, the mesh updation is done using the subroutine in 'General-static' and later the second iteration is restarted in Dynamic implicit, and the computational process is repeated. From the Figure 9 (e) and (f) show the material removal distribution obtained from the FE simulation. It can be observed that the maximum material removal depth obtained after the first iteration 'j' is **-0.21 mm** and the final maximum material removal depth obtained after the last iteration is -0.366mm (negative sign indicates the material removal).

Figure 9 (a), (b) Average dynamic contact pressure (MPa), (c),(d) sliding distance range (mm) and (e),(f) material removal depth distribution (mm) for iteration 1 and iteration 2 respectively.

Figure 10 Comparison between 3D (a) experimental (b) numerical material removal distribution and 2D (c) experimental (d) numerical material removal profiles for hard backing pad

In Figure 10 (a) and (b), the experimentally measured and numerical simulated 3D material removal distribution are compared. The contact length of the considered experimental distribution is 25 mm, and that of numerical distribution is 30.0 mm and whereas the contact width dimensions are 5.76 mm and 6.0 mm for experimental and simulation respectively. But the aforementioned comparison is done for a single experimental material removal distribution. Since four experimental trials are conducted, the average value of contact length and width are 25.52 ± 0.46 mm and 5.65 ± 0.21 respectively. Similarly, from the Figure 10 (c) and (d), for the considered experimental profile, the maximum material removal depth value (-0.388 mm) agrees with the numerical value of -0.366 mm (5.6% error), and the overall maximum material removal depth range calculated from the four experimental trials is -0.389 ± 0.007 mm ($-389 \pm 7 \mu\text{m}$) which agrees with the numerical and simulated values (5.9% error). Also, there is a shift in the location of the maximum material removal depth, this can be attributed to the randomness in the abrasive disc geometry (warp) and hence a shift in the maximum pressure position. But in machining trials with feed motion, the material removal depth parameter plays a key role than the maximum material removal depth location.

Figure 11 (a), (b) Average dynamic contact pressure (MPa), (c), (d) sliding distance range (mm) and (e),(f) material removal depth distribution (mm) for iteration 1 and iteration 2 respectively.

Figure 11 (a) and (b) show the average dynamic contact pressure distribution obtained from the two iterations and Figure 11 (c) and (d) depicts the sliding distance parameter. Like the hard backing pad simulation results, there are contact singularity points noticed in the periphery of the disc. It is observed from Figure 11 (b) that the maximum pressure value is 0.295 MPa and the maximum pressure value that of the hard backing pad (Figure 9 (b)) is 0.457 MPa. Thus, it can be inferred that this contact pressure difference induced by the change in backing pad compliance plays a role in controlling the material removal depth. Additionally, from CSLIP contours of hard pad (Figure 9 (d)) and soft pad (Figure 11 (d)) it can be observed that CSLIP values at these maximum pressure points are 1057.0 mm and 562.4 mm for hard and soft backing pads respectively. Because of this, for the hard backing pad the maximum material removal depth is observed at the same point where the pressure is

maximum, but in the case of the soft backing pad because of this reduction in CSLIP, another point where CSLIP is higher and pressure is lesser has the maximum material removal depth. Thus, confirming that the sliding distance also plays a major role in determining the material removal depth according to the Archard's wear equation.

Figure 12 Comparison between 3D (a) experimental (b) Numerical material removal distribution and 2D (c) experimental (d) numerical material removal profiles for soft backing pad

Figure 12 (a) and (b) shows the material removal distribution of the experimental trial and numerical simulation respectively. The mean contact length obtained from the experimental trial is 25.5 ± 0.867 mm and from the numerical simulation is 33.0 mm, whereas the experimental mean contact width is 9.43 ± 0.33 mm and the numerical result is 8.0 mm. The mean of the experimental maximum material removal depth values is 0.133 ± 0.009 mm ($133.25 \pm 9.81 \mu\text{m}$), and the maximum material removal depth obtained through numerical simulation is 0.144 mm which agrees with the experimental measurement (8.27%). Figure 12 (c) and (d), shows the 2D profile of the material removal depth and it can be observed that the maximum material depth matches as mentioned but due to randomness in the abrasive disc geometry such as inherent warps, the maximum material removal depth position is different from the experimental position.

6 Conclusion

In order to investigate the effect of backing pad compliance and load on contact pressure distribution, three-dimensional models are created along with the relevant material properties and boundary conditions. To study the behaviour of backing pad and abrasive disc, both separately and combined, two scenarios are considered in this study. For the 20N applied load, the numerical contact pressure distribution and load-displacement responses for the both the scenarios are compared with the experimental measurements. The overall contact

length and width agrees well with the experimental results, and except for the stress singularity regions, the computed pressure matches the experimental data. A numerical methodology to compute the overall material removal depth distribution using Archard abrasive wear and adaptive meshing technique is proposed. Accordingly, based on the contact pressure and sliding distance values obtained from each node on the workpiece surface, the wear depth is computed using Archard wear formula, and the workpiece is re-meshed using UMESHMOTION subroutine in ABAQUS®. The maximum material removal depth computed using the FE model for both hard and soft pad agrees well with the experimental material removal (<10% error). From the experimental and numerical material removal distribution, it can be observed that the material removal nature in hard pad for the particular load is more concentrated in a particular region which is more useful in finishing parts with weld seams. In the case of the soft backing pad, the material removal is distributed, and the overall material removal depth is almost half compared to the hard pad which indicates that backing pad with better compliance is suitable for blending and fine polishing operations. Further, this model can be extended to polishing scenarios with feed rate and investigate the applicability of the model in workpieces with different geometries, where additionally, the geometry influences the contact pressure and material removal. Presently, the polishing of aerospace or marine components is based on operator's skills and any improper selection of parameter would lead to rework of these components which is expensive and time consuming. Thus, the proposed numerical methodology will be helpful in providing more insight to the material removal predicting the material removal depth, once the wear coefficient and material properties of both the backing pad and abrasive discs are given as input to the model.

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Appendix

Figure A.1 (a) Woven fabric as backing material of the abrasive disc (b) Three-point bending test for the specimens cut from abrasive disc

Table A.1 Material properties of abrasive disc

E_1 (MPa)	E_2 (MPa)	ν	G_{12} (MPa)	G_{13} (MPa)	G_{23} (MPa)
1013.8	682.5	0.3	550	550	500