The surface temperature prediction on steel-tool steel sliding pairs

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Abstract—This study examines the theoretical and numerical prediction of temperature at the sliding contact surfaces of steel-tool pairs using the pin-on-disc and Archard models. The steel-tool contact pair is important to the tribological and forming community. The results show a good correlation between numerical and theoretical calculation of contact temperature. There was abrupt increase in the contact temperature at the contacting surfaces as the sliding speed was increased beyond 35 mm/s. Localization of high peak temperature at the contact regions of the sliding surfaces observed in this study may be important in the wear mechanism of sliding bodies and wider manufacturing community.

Keywords—Pin-on-disc, forming process, steel, temperature, contact surfaces

I. INTRODUCTION

Flash temperature is the temperature that occurs at the nanometre scale asperity during contact between sliding bodies. Due to the magnitude of these flash temperatures, compared to bulk and surface temperatures of the sliding surfaces, the local mechanical properties of the interacting materials can be affected [1]. The surface temperature is the average of the flash temperature over the surface of the sliding bodies. While flash temperature occurs at the asperities of the contacting surfaces [2]. It is likely that the magnitude of the flash temperature will be greater than that of the surfaces temperature in any given tribological set up. The generated temperature can cause or increase the severity of wear mechanisms on the sliding surfaces and hence influence the tribological behavior of the sliding surface [3,4]. In sheet metal stamping, the sliding contact between the sheet and the tool can result in high frictional heat generation at the die radius-blank contact [5]. The generated temperature may be influential in the increasing die maintenance cost and scrap rate [4]. Understanding the evolution of temperature in relation to the sliding surface pairs may help in reducing tool wear in sheet metal stamping.

Previous attempts have been made to use Finite Element (FE) analyses to predict the moderate temperature increases and relate it to the wear of tool steels used in sheet metal forming [6-8]. The temperature rise on the die surface experienced during a continuous strip drawing process of aluminum sheet was examined by Groche et al. [6], using FE analysis. It was predicted that a peak contact temperature rise of 36°C occurred near the point of peak contact pressure, corresponding to the location at which the adhesive wear mechanism was found initiate. Recently, Pereira et al. [5] predicted a temperature rise of up to 130°C during ‘cold’ forming of advanced high strength steels, which was attributed to the increased contact stresses and increased plastic work associated with stamping higher strength sheet materials.

The purpose of the present study is to simulate and predict the surface temperature at the sliding contact of low carbon steel against hardened D2 tool steel with a maximum contact pressure of 1084 MPa. This contact pressure is typical of the peak pressures experienced during sheet metal forming operations using high strength sheet steel [9]. This analysis will aid in understanding the temperature distribution on the surfaces during the sliding contact condition.

A. Numerical simulation setup

In this study, a 3D FE model was developed to simulate the surface temperature generated as a result of frictional heating during the pin-on-disc tests wear tests. The FE model of the pin-on-disc test setup is shown in Figure 1.

Figure 1: a) Full model showing pin in sliding contact with the disc, b) the pin surface showing mesh refinement at the contact surface, c) surface of the disc showing mesh refinement at the region of contact between the disc and pin.
A general-purpose non-linear implicit finite element analysis code (ABAQUS/Standard Version 6.14-1 [10]) was used to conduct the simulation. All pre- and post-processing was carried out using ABAQUS/CAE.

1) Key parameters and properties

The details of the thermal and material properties of the pin and the disc used in the simulation are shown in Table 1.

<table>
<thead>
<tr>
<th>Thermal and material properties</th>
<th>Disc</th>
<th>Ball</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material definition</td>
<td>Deformable-solid</td>
<td>Deformable-solid</td>
</tr>
<tr>
<td>Elastic modulus (MPa)</td>
<td>210000</td>
<td>210000</td>
</tr>
<tr>
<td>Poisson’s ratio (-)</td>
<td>0.3</td>
<td>0.3</td>
</tr>
<tr>
<td>Friction energy heat dissipation factor, $\eta_F$ (-)</td>
<td>Default</td>
<td>Default</td>
</tr>
<tr>
<td>Density (kg.mm-3)</td>
<td>7.9x 10^-6</td>
<td>7.9x 10^-6</td>
</tr>
<tr>
<td>Thermal conductivity (mJ.s^-1.mm^-1°C^-1)</td>
<td>20</td>
<td>51.4</td>
</tr>
<tr>
<td>Expansion coefficient ( °C^-1)</td>
<td>12x10^-6</td>
<td>12x10^-6</td>
</tr>
<tr>
<td>Specific heat capacity (J.kg^-1.°C^-1)</td>
<td>4.5x10^2</td>
<td>4.5x10^2</td>
</tr>
</tbody>
</table>

The key parameters required to develop the coupled thermal-stress solution are listed in Table 1. These parameters were defined for both the pin and the disc based on characteristics appropriate for steel. The friction energy heat dissipation fraction, $\eta_F$, defines the amount of work from friction that is converted into heat which is distributed to the disc-pin surfaces. Due to the short time scale involved, the effects of convection and radiation were assumed to be negligible and were not accounted for in the model.

2) FE mesh and geometry

The FE mesh of the disc and pin were refined at the region of the contact to allow detailed analysis of the contact between the pin and disc surfaces (as shown in Figure 1). Four-node, thermally coupled tetrahedron elements with linear displacement and temperature (C3D4T) were selected and used to mesh all parts of both the pin and the disc. Tie constraints were used to merge together dissimilar regions of the model, allowing mesh transition from fine mesh at the contact surfaces to coarse mesh at other parts of the pin and the disc. The length of the element sides at the contact regions of the pin and the disc were approximately 0.002mm. The objective was to reduce the number of elements at the non-contact regions of the model, thereby reducing the computation time, whilst allowing sufficient mesh density at the contact zones to allow the contact pressure and temperature distributions at the pin-disc interface to be accurately simulated. In the experimental setup, the disc is harder than the pin material [11]; hence, the disc surface was set as the master surface in each of the contact interactions, while the pin was set as the slave.

3) Load and boundary conditions

Load of 5N was applied on the top of the pin typical of that used in the experimental study resulting in maximum contact pressure experienced during a typical stamping process [12]. Angular velocities of 1.52rad/s, 4.45rad/s, 21rad/s and 31.76 rad/s which is equivalent to 12mm/s, 35mm/s, 170mm/s and 250mm/s sliding speed respectively were applied at the last step of the model. Time steps of 41.08s, 14.14s, 2.99s and 2.02s for 12mm/s, 35mm/s, 170mm/s and 250mm/s were also applied at the last step to give four revolutions each of the simulation tests. A Coulomb friction model was used to represent the friction between the blank and the tool surface using an isotropic penalty friction formation in Abaqus [10]. The friction coefficient applied ($\mu=0.50$) was taken from the steady state value obtained from the experimental pin-on-disc tests conducted at ambient temperature [11].

The simulation process was divided into two steps: a contact step; and then a rotating step. At the contact step, a normal load was applied downwards through the ball to apply contact loading between the disc and the pin. Then, followed the rotation step where the disc was made to rotate in the z-axis. The top of the pin and back of the disc were set to zero to act as heat sink. The purpose was to allow any heat dissipated from the top of the pin and the bottom of the disc not to go back to the system. Temperature was queried at different nodal on the pin from the contact to some distance away from the contact with the disc as shown in Figure 2. The objective was to investigate how the temperature varies as it goes away from the contact. The model is first validated and then the simulation results are subsequently presented and discussed below.

a) Pin and disc reference nodal positions

The cross section of the pin and the disc show different nodal positions as shown in table 2.

<table>
<thead>
<tr>
<th>Pin Id Number</th>
<th>Distance away from contact [mm]</th>
<th>Disc Id Number</th>
<th>Distance away from contact [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>P1</td>
<td>0.0</td>
<td>D1</td>
<td>0.0</td>
</tr>
<tr>
<td>P2</td>
<td>0.01</td>
<td>D2</td>
<td>0.01</td>
</tr>
<tr>
<td>P3</td>
<td>0.03</td>
<td>D3</td>
<td>0.02</td>
</tr>
<tr>
<td>P4</td>
<td>0.23</td>
<td>D4</td>
<td>0.03</td>
</tr>
<tr>
<td>P5</td>
<td>0.57</td>
<td>D5</td>
<td>0.05</td>
</tr>
<tr>
<td>P6</td>
<td>1.96</td>
<td>D6</td>
<td>0.08</td>
</tr>
</tbody>
</table>
B. Theoretical calculations and FE model validation

The results from theoretical calculations of contact pressure and surface temperature, using well-known Hertz [13] developed models from the literature are presented. The results from these calculations were used to help to develop and validate the FE model developed.

1) Contact pressure validation.

The FE model was validated against a standard analytical Hertz. The numerical calculation result of the Hertz contact pressure using Hertz [13] developed models are shown in Figures 2 and 3. During the FE model development process, the static contact pressure distribution was compared to the Hertzian contact pressure distribution in order to check the accuracy of the contact model and choose an appropriate mesh density at the contact zone. The result of the predicted FE model at the end of the first step (where the pin is loaded against the disc prior to sliding) is shown in Figure 2. As shown, both models predict similar trends, but there are some discrepancies at both edges away from the center of contact. This is difference in sensitivity of the mesh relative to the radius geometry. However, minimum number of elements of about nine at the small contact zone of the surface compared to 12 elements distributed at the other areas of the sliding surface provides a good correlation of the contact pressure at the contact region. Hence it can be concluded that the FE model accurately captures the theoretical results and can be used to validate the FE model for further investigations.

2) Peak surface temperature

The results obtained from the simulation are presented and compared with Archard’s theory for temperature of rubbing surfaces to validate the sliding speeds selected for this study. The sliding speeds were selected based on Archard’s theory to give an equal surface temperature.

C. Results

Our validation shows that the FE model results correlate well with theoretical calculations of contact pressure and peak surface temperature. Hence, the validated FE model can be used to investigate the temperature distribution at the pin and disc surfaces.

1) Pin temperature distribution

Figure 4 shows the peak temperature of P1 at different sliding speeds. The P1 nodal temperature, which is at contact with the disc node D1 shows highest peak temperature for the different sliding speed tests. The result also shows that the peak temperature increases approximately linear with the sliding speeds.
**a) Temperature distribution along the pin surface**

The contour plots result of the temperature distribution along the surfaces of the pin for all the sliding speeds are shown in Figure 8. The results show that temperature is localised at the pin contact area with the disc and gradually decreases as it goes away from the contact region. The localisation of the peak temperature at the contact area on the pin surface was observed for all the sliding speed tests. Figure 5a shows the peak temperature rise of 2.2°C at the contact area of the pin for the 12mm/s tests, followed by temperature rise of 8.2°C for the 35mm/s tests. Temperature rise of approximately 30°C and 36°C each were recorded for the 170 and 250mm/s tests respectively (Figure 5c and 5d). These peak contact nodal temperatures agree with the contact nodal temperature observed down the pin surface (Figure 4).

**2) Disc temperature distribution**

The result of the nodal temperature measured at different contact points on the disc to investigate the temperature distribution on the disc surface are displayed in Figure 6. Similarly, reference nodes were selected some distance away from the pin-disc contact for different sliding speeds. The peak temperature increases, almost linearly, with the sliding speed. The result is in agreement with the similar increase in the peak temperature observed for the pin reference node P1 shown in Figure 4.

Figure 6: Peak temperature for Disc node D1 against sliding speed

**a) Temperature distribution along the disc surface**

The contour plots of the temperature distribution along the disc surface for all the four sliding speeds are shown in Figure 7. The general result shows that the magnitude of localised peak temperature at the contact area decreases as the distance away from the contact region increases. Peak temperatures of 4.2°C and 16.1°C were observed for 12mm/s and 35mm/s speed tests respectively. For the higher sliding speed tests of 170mm/s and 250mm/s, 51.7°C and 61.7°C peak temperature rise were observed at the contact region on the disc surface (Figure 7). Another observation was that the peak temperatures at different speeds on the disc surfaces (Figure 7) were generally higher than the peak temperature rise observed on the pin surface (Figure 5). This was attributed to the difference in the thermal conductivities of the two sliding pairs (Table 1).
D. Discussion

The results show that the prediction of accurate surface temperature could be made using either the Archard’s theory [3] or the simulation method. Increase in the surface temperature was observed as the sliding speed was increased. It has been shown previously in the literature that frictional heat is generated due to welding and plastic work of the asperities [14]. In a high sliding speed process, the generated heat does not have sufficient time to dissipate and can cause a significant rise in temperature as observed in this result. Figure 4 shows that the surface temperature at the contact region of the sliding surface of the pin is significantly higher than at other positions on the sliding surface. Similar increase in the contact surface temperature compared to the other region on the contacting bodies have been reported by Ashby et al. [15] to be due to flash temperature generated as a result of asperities deformation. The result of temperatures at different nodes on the disc also showed increasing surface temperature as the sliding speed was increased (Figure 6). However, sliding at lower sliding speed did not show significant difference between the contact nodal surface temperature and those from other points away from the disc contacting surface (Figure 4).

Notably, at 170 mm/s and 250 mm/s, the temperatures at the contact node of the disc are significantly higher than temperatures on the other nodes, similar to that of the pin (Figure 7c). It was found in the results (Figure 4 and 6), that the surface temperature rises for the disc showed higher temperature values for the contacting nodal temperature compared to that of the pin nodal temperature. An explanation to the above observation could be that in pin-on-disc test, the heat flows into the sliding bodies. Subsequently, the temperature distribution on the sliding bodies is influenced by the partitioning factor and the thermal conductivities of the material pair in sliding contact as reported by Bowden and Tabor [16]. The above observation is of practical value to wear mechanisms and the manufacturing industry.

E. Conclusion

The study investigates two techniques of estimating surface temperature. The results show that the well-known behaviour that surface temperature increases as the sliding speed is increased was observed for all tests, even at the slowest sliding speeds (<20mm/s). There is a good correlation between the theoretical and numerical calculation of surface temperature. The result provides more confidence in the predicted FE results. The magnitude of the surface temperature predicted at the surface of the disc and pin are significantly high at the contact region and increased abruptly as the sliding speed was increased (<35mm/s). Considering the distribution of the temperature along and below the sliding surfaces, it becomes clear that frictional heating does not result in any significant bulk temperature heating. The numerical models show that, at the higher sliding speeds, the surface of the disc experiences over 10 times larger temperature rise compared to the material that is just 0.01mm below the surface.

REFERENCES
13. H. Hertz, Über die berührung fester elastischer körper (on the contact of elastic solids) (English translation in Miscellaneous Papers by Hertz, H.), J. Reine Angewandte Math, 94, 156–171 (1822)