PERFORMED WEAR SIMULATION OF A BRAKE PAD BY USING A COMMERCIAL FEA SOFTWARE AND COMPARED THEIR RESULTS WITH
physical tests. They considered the real surface topography of the pad while building the finite element model by measuring height distributions with a gauge. Söderberg and Andersson [12] performed a simulation of wear and contact pressure distributions of the brake pad using a general purpose finite element analysis software. Vernersson and Lundén [13] studied the behavior of brakes numerically for repeated brake cycles. They used a 2D fully coupled FE model while considering the coefficient of friction as being constant and a temperature dependency of the wear rate. They found out that wear of the pad strongly depends on the stiffness of the friction material and its mounting.

Today, the prevalent way to simulate frictional heating of disc brakes in commercial softwares is to use the Lagrangian approach in which the finite element mesh of a disc rotates relative to a brake pad. Although this approach works well, it is not feasible due to extremely long computational times. Particularly, for simulating repeated braking this approach is of little importance for practical use. Sometimes two-dimensional FE models are used to reduce the computational time but this approach is not sufficient to model complex behavior. The rotational symmetry of the disc makes it possible to model it using an Eulerian approach, in which the finite element mesh of the disc does not rotate relative to the brake pad but the material flows through the mesh. This requires significantly lower computational time as compared to the Lagrangian approach. Nguyen et al. [14] developed an Eulerian algorithm for sequentially coupled thermal mechanical analysis of a solid disc brake. First they performed a 3D contact calculation to determine the distribution of the pressure. Then a sequentially coupled analysis is implemented by first performing a transient heat transfer Eulerian analysis followed by a steady-state mechanical analysis. Recently, Strömberg [15] developed a finite element approach using an Eulerian framework for simulation of frictional heating in sliding contacts. In his approach, the fully coupled problem is decoupled in one mechanical contact problem and a frictional heat problem. For each time step the thermoelastic contact problem is first solved for the temperature field from the previous time step. Then, the heat transfer problem is solved for the corresponding frictional power. In another paper [16] this approach was implemented for simulating frictional heating in disc-pad systems.

In this work, frictional heating of a disc brake, while taking the wear into account, is simulated by implementing an Eulerian approach. A toolbox developed by Strömberg, which is based and described in his earlier work [16] but now extended to include wear of the pad, is used to perform the frictional heat analysis. In this Eulerian approach the contact pressure is not constant, but varies at each time step taking into account the wear and thermomechanical deformations of the disc and the pad. This updated contact pressure information is used to compute wear, and heat generation and its flow to the contacting bodies at each time step. In such manner, the wear and nodal temperatures are updated accurately and their history is recorded at each time step. Then a Python script is used to write the wear and temperature history to an output file for subsequent use. The disc-pad system is simulated for several brake cycles. After each brake cycle pad geometry reflects the material removed by accumulated wear and this updated geometry of the pad is used in subsequent brake cycles. Because the finite element mesh of the disc does not rotate relative to the pad, the contact region is always well defined and a node-to-node based approach can be adopted. This allows the mesh to be refined only in the region where the brake pad is in contact with the disc, which results in lower computational time. The output file with temperature history can be used e.g. in a sequentially coupled stress analysis.

The results show the appearance of two hot bands on the disc surface after several brake cycles which cannot be predicted when wear is ignored. The Eulerian approach has proved tremendously cheap in terms of computational time when compared to a fully coupled Lagrangian approach. This is demonstrated by presenting numerical results.

2. FRICTIONAL HEAT ANALYSIS

The workflow of the approach used for frictional heat analysis is shown in Fig. 1. An input file, which contains the meshed geometry with appropriate boundary conditions and loads is required for the frictional heat analysis. During this analysis linear thermo-elasticity is adopted and the problem is decoupled in two parts. In the first part, for a

![Diagram](image-url)
given temperature distribution the contact problem is solved while taking the wear of the pad into account to obtain the nodal displacements and contact pressure distribution. The new contact pressure distribution is used to update the wear gaps. In the second part, for the obtained contact pressure distribution the energy balance is solved and new nodal temperatures are determined. These equation systems are then solved sequentially and, wear and temperature histories are developed. The nodal temperatures determined at a time step are taken into account in the next time step to update the deformed geometry of the disc and pad. This is shown schematically in Fig. 2. The wear and nodal temperature history is then written in an output file (called ODB file) by using a Python script. Details about the governing equations can be found in [16].

Three parts are considered for the frictional heat analysis. Materials assumed for the disc and the back plate are cast iron and steel, respectively. Friction material used as brake pad is a composite. Temperature independent material properties used for these parts are listed in Table 1.

### Table 1: Material properties for frictional heat analysis.

<table>
<thead>
<tr>
<th>Material Property</th>
<th>Disc</th>
<th>Pad</th>
<th>Plate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity [W/mK]</td>
<td>47</td>
<td>0.5</td>
<td>46</td>
</tr>
<tr>
<td>Young’s modulus [GPa]</td>
<td>92.9</td>
<td>2.2</td>
<td>210</td>
</tr>
<tr>
<td>Poisson’s ratio [-]</td>
<td>0.26</td>
<td>0.25</td>
<td>0.3</td>
</tr>
<tr>
<td>Thermal expansion coefficient [10⁻⁵/K]</td>
<td>1.55</td>
<td>1</td>
<td>1.15</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>7200</td>
<td>1550</td>
<td>7800</td>
</tr>
<tr>
<td>Heat capacity [J/kgK]</td>
<td>507</td>
<td>1200</td>
<td>460</td>
</tr>
</tbody>
</table>

#### 3. NUMERICAL RESULTS

The assembly of the disc-pad system considered in this paper is shown in Fig. 3. This is an assembly of a disc-pad system of a heavy Volvo truck. The outer diameter and thickness of the disc are 434 [mm] and 45 [mm], respectively. The ventilated disc is geometrically symmetric about a plane normal to the z-axis. It is assumed that thermomechanical loads applied to the system are symmetric so only half of this assembly is considered for the simulation and symmetry constraints are applied on the nodes lying on the symmetry plane. Some detailed geometry at the inner radius has been removed to simplify the model as that is not important for this analysis. The displacements along x and y directions of the nodes located at the inner radius of the disc are set to zero. All the surfaces of the disc, except the one lying on the the symmetry plane are considered to lose heat by convection.

The brake pad is supported by a steel plate at the back side as shown in Fig. 4. Some detailed geometry of the back plate which is not necessary for the simulation has been removed. Two cylindrical pins apply a normal force on the back surface of the back plate which transmits it to the pad. Displacements at the back surface of the back plate, other than along the force direction, are fixed. Furthermore temperature is set to zero on the back surface.
The disc is meshed such that it has smaller elements where it contacts the pad as shown in Fig. 5 (only a small portion of the disc is shown). This is an advantage of the Eulerian approach because the finite element mesh of the disc does not rotate relative to the brake pad but the material flows through the mesh. The heat flux generated at the interface of the stationary pad and the disc is considered with convective heat transfer in the disc. In a Lagrangian approach a fine mesh should be applied on the complete surface of the disc because the finite element mesh of the disc rotates relative to the brake pad or some adaptive strategy should have to be applied. All the parts considered for the simulation are meshed with 4-node linear tetrahedron elements in HyperMesh (HyperWorks 10.0). These meshed parts are then used to prepare input file with boundary conditions and loads in Abaqus/CAE. The disc assembly is meshed with 269438 elements that has 64957 nodes and 185697 degrees of freedom.

Now the results of frictional heat simulations will be described for two different cases. In the first case, a brake application is simulated for one cycle and wear is not considered. Figure 6 shows the surface temperature as a function of time and disc radius for this case. The nodes of the disc chosen for this plot are located at 180° away from the middle of the pad. A brake force of 24.5 [kN] is applied for 45 [s] on the back surface of back plate. The angular velocity of the disc is 45 [rad/s] and held constant throughout the simulation. This loadcase corresponds to a truck moving downhill with a constant speed. The force is ramped up by using a log-sigmoid function during 20 time increments and then held constant for next 70 increments with time step = 0.5 [s]. The friction coefficient is $\mu = 0.3$, contact conductance coefficient is $\phi = 0.1$ [W/NK] and convection coefficient is set to 50 [W/m²K]. The brake force generates an average brake moment of 1240 [Nm] after the ramping up. The total CPU time is 4272 [s] on a workstation with Intel Xeon X5672 3.20 GHz processor. In the graph it can be seen that temperature is not uniformly distributed over the disc instead a narrow band with relatively higher temperature appears in approximately middle of the disc surface.

In the second case, brake application is simulated for several cycles and material removed due to wear in each cycle is considered in subsequent braking operations. During each brake cycle, the wear coefficient is set to $10^{-10}$ [m²/N] and rest of the parameters are same as for the first case. The total CPU time for a single cycle is 4289 [s] on a workstation with Intel Xeon X5672 3.20 GHz processor. Each brake cycle requires almost the same CPU time for each simulation. In Fig. 7, temperature of the disc surface is shown at the end of brake operation for first cycle. A ring of high temperatures, called a hot band, is evident in the middle of the disc. Figure 8a shows the surface temperature as a function of time and disc radius for the first cycle of brake application. The nodes of the disc chosen for this plot are located at 180° away from the middle of the pad. In the graph it can be seen that during the cycle there is only one hot band on the disc surface.

By intuition it can be thought that the high temperature ring should form near the outer radius of the disc. But the ring appeared approximately in the middle of the disc surface. It might be understood by studying the contact pressure plots at different time steps as shown in Fig. 10. In Fig. 10a the contact pressure plot for the first time increment or at the moment when the pad comes into contact with the disc is shown. It can be seen that the contact pressure is not the highest at the outer radius of the pad. The region where contact pressure is higher generates more heat and causes further expansion of the disc and the pad material near this area which in turn causes higher contact pressure. In the meantime convex bending caused by thermal deformation of the pad and the back plate, as shown in Fig. 9, also plays a major role in concentration of contact pressure towards the middle of the pad surface. This

![Figure 3: An assembly of the disc-pad system, also showing the cylindrical pins used to push the back plate.](image)
convex bending can be explained by the expansion of the pad surface material due to the increase in temperatures. The frictional heat causes the pad surface temperature to rise in a short period of time as compared to the inner region of the pad and the back plate as shown in Fig. 9. Consequently, the surface expands more than the inner region of the pad and the back plate which results in the convex bending. These phenomena combined with the ramping up of brake force in later increments, causes the higher contact pressure in an area which is away from the outer radius of disc as shown in Fig. 10d.

In Fig. 11 contact pressure plots are shown for further time steps when the brake force is held constant for the first cycle and Fig. 12 shows the wear on the pad for corresponding time steps. It can be seen that the contact pressure keeps on concentrating towards the middle of the pad with increasing time increments. It can also be observed that wear is higher in the areas where contact pressure is higher.

Figure 8b shows the surface temperature as a function of time and disc radius for the 41st cycle of brake application. It can be seen that in the beginning there are two hot bands which converge to one as the temperature increases with time. In Figure 13 which shows temperature of the disc surface at 13th time increment for the 41st cycle of brake application, two hot bands can be seen. In Fig. 14, temperature of the disc surface is shown at the end of brake operation for the 41st cycle of brake application. By comparing with Fig. 7, it can be concluded that after 41 brake cycles the maximum temperature has decreased and the hot band becomes wider at the end of brake operation. The appearance of two bands can be explained by the shifting of high contact pressure areas. Due to the concentrated wear in the middle of the pad during repeated brakings, a depression appears when the pad cools down and returns to its undeformed state at the end of a brake operation. So during next brake cycle, the high contact pressure first builds on the outer regions of the pad surface. In Fig. 15 accumulated wear of the pad is shown at the end of the 40th brake cycle. Fig. 16 shows the distribution of contact pressure during the 41st cycle. It can be seen that contact pressure first builds on the outer regions which are less worn out and then due to thermomechanical deformations of the pad, as discussed before, moves to the middle of the pad surface with increasing time increments. By comparing the results of the first case with those obtained for the first brake cycle of the second case, it can be concluded that for a pad without wear history there is no noticeable influence during braking due to wear. But accumulated wear does have a significant influence on the distribution of temperature after some brake cycles.

4. DISCUSSION

The temperatures predicted by the in-house software have been compared with the temperatures recorded by a thermal imaging camera during a physical test and found to be relatively higher. Moreover, two hot bands predicted after repeated brake cycles are not as distinct as observed in the thermographs. These differences could be due to temperature independent material data, friction coefficient, and wear coefficient used during the frictional heat analysis. For more realistic results, temperature dependent material data should be used. Furthermore, the friction
coefficient of a brake pad is generally dependent on temperature, velocity and contact pressure [17] but in this work it is assumed to be constant at $\mu = 0.3$ to represent an average behavior. Similarly, the wear coefficient is generally dependent on temperature and velocity [6, 18] but in this work it is assumed to be constant at $10^{-10}$ [m$^2$/N]. In a very near future, we will extend this work such that a temperature dependent behavior of the friction and wear coefficients is included in the proposed method. At present the in-house software assumes constant angular velocity of the disc that corresponds to a vehicle moving downhill with a constant speed but in the future it could also be extended to non-constant angular velocities.

5. CONCLUDING REMARKS

In this work frictional heat analysis of a disc brake has been performed taking into account wear of a pad. This analysis is performed in an in-house software based on the Eulerian approach. It has been shown that braking history affects the evolution of temperature distribution during a brake cycle. The analysis predicts concentrated wear in the middle of the pad which results in the appearance of two hot bands after repeated brake cycles.

It has been shown that other than the local factors e.g. thermal expansion, convex bending of the pad and the back plate also plays a major role in the contact surface evolution. Phenomenon of convex bending has been described in other works [2, 3], to the best of our knowledge, but no experimental observation or numerical simulation results have been presented to support it. In this paper it has been shown with numerical simulations that convex bending plays a major role in the concentration of contact pressure to the middle of pad.

This method has proved tremendously cheap in terms of computational time when compared to the Lagrangian approach. In the future this approach can be used to study the influence of different geometries of the pad and the disc on the maximum temperature with a reasonable simulation time. It can be very useful when studying new designs for real disc brake systems.
Figure 8: Temperature as a function of time and disc radius with the consideration of wear.

Figure 9: Thermally induced deformations of the pad and back plate during brake operation shown in different projections. The deformation is exaggerated for visual clarity.

Figure 10: Nodal contact forces represented as pressure plots on the pad surface shown at different time steps for the first cycle during ramping up of the brake force. The legend is given in [N].
Figure 11: Nodal contact forces represented as pressure plots on the pad surface shown at different time steps for the first cycle while the force is held constant. The legend is given in [N].

Figure 12: Wear on the pad surface, shown in [m], at different time steps for the first cycle.

Figure 13: Two bands of high temperatures on the disc surface at $t = 6.5$ [s] during the 41st cycle of brake application.
Figure 14: After the brake application at the 41st cycle, a ring of high temperatures develops on the disc surface.

Figure 15: Accumulated wear on the pad surface, shown in [m], at the end of the 40th cycle.

Figure 16: Nodal contact forces represented as pressure plots on the pad surface shown at different time steps for the 41st cycle. The legend is given in [N].
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References


