A VIRTUAL TEST RIG FOR BRAKE DISCS
BY ADOPTING A THERMO-FLEXIBLE
MULTI-BODY APPROACH

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Outline

- Background
- New computational approach
- Governing equations
- Algorithm
- Simulation of heat bands in brake discs
- Temperature history, contact pressure, wear
- Simulation of thermal stresses
- Performance
- Test rig - dynamometer
- Virtual test rig - dynamometer
- Temperature dependent friction
- Design optimization
- Concluding remarks
Background
Background
Traditional approach - Lagrangian

Lagrangian formulation

+ Established approach that can be found in many commercial softwares

- Very long CPU-times
New suggested approach - Eulerian

Eulerian formulation

- Not available on commercial softwares

+ Node-to-node contact
+ Fine mesh locally at the contact between disc and pad
+ Much shorter CPU times
Governing equations, disc and pad – linear thermoelasticity

\[ Q = \sigma_0 = d + \text{B.C.} \]

\[ F = 1D \]

\[ T = 0 \]

\[ d = 0 \]

\[ F = 3D \]

\[ K_{ik}^{BA} = \int_{\Omega_m} E_{ijkl} \frac{\partial N^A}{\partial x_l} \frac{\partial N^B}{\partial x_j} \, dV, \]

\[ \hat{K}_i^{BA} = \int_{\Omega_m} \alpha (3\lambda + 2G) N^A \frac{\partial N^B}{\partial x_i} \, dV, \]

\[ \epsilon = \frac{\sigma}{E} + \alpha \Delta T \]

\[ \frac{\partial \sigma}{\partial x} = 0 \quad +\text{B.C.} \]

\[ \sigma_{ij} = \lambda \epsilon_{kk} \delta_{ij} + 2G \epsilon_{ij} - (3\lambda + 2G') \alpha T \delta_{ij} \]

\[ \frac{\partial \sigma_{ij}}{\partial x_j} = 0 \quad +\text{B.C.} \]

\[ Kd - \hat{K}T = F \]

\[ \text{FEM} \]
Governing equations, pad – heat balance, Lagrangian formulation

\[ d = 0 \]

\[ T = 0 \]

3D

\[ \rho c \dot{T} = k \sum_{i=1}^{3} \frac{\partial^2 T}{\partial x_i^2} + \text{B.C.} + \text{I.C.} \]

FEM

\[ M \dot{T} + OT = Q \]

\[ M^{BA} = \int_{\Omega^m} \rho c N^A N^B \, dV, \]

\[ O^{BA} = \int_{\Omega^m} k \frac{\partial N^A}{\partial x_i} \frac{\partial N^B}{\partial x_i} \, dV, \]
Background, disc – heat balance, Eulerian formulation

\[ x_i = x_i(X, t) \]

or

\[ X_i = X_i(x, t) \]

\[ T = T(x, t) \]

\[ \dot{T} = \frac{\partial T}{\partial x_i} \frac{\partial x_i}{\partial t} + \frac{\partial T}{\partial t} = v_i \frac{\partial T}{\partial x_i} + \frac{\partial T}{\partial t} \]

\[ \rho c \frac{\partial T}{\partial t} + \rho cv \frac{\partial T}{\partial x_1} = T \sum_{k=1}^{3} \frac{\partial^2 T}{\partial x_i^2} + \text{B.C. + I.C.} \]

\[ M \dot{T} + OT + NT = Q \]

\[ M \dot{T} + OT + NT + RT = Q \]

\[ N^{BA} = \int_{\Omega^1} \rho c v_i \frac{\partial N^A}{\partial x_i} N^B \, dV \]

\[ R^{BA} = \bar{k} \int_{\Omega^1} v_i \frac{\partial N^A}{\partial x_i} v_j \frac{\partial N^B}{\partial x_j} \, dV \]
Governing equations, contact interface – equilibrium

\[ p_t = \mu p_n \]

\[ p_n \geq 0, \quad \delta_n \leq 0, \quad p_n \delta_n = 0 \]

\[ p_n = \frac{p_n + r \delta_n + |p_n + r \delta_n|}{2} \]

\[ \delta_n = d_{n1} + d_{n2} - g \]

\[ K_1 d_1 - \hat{K}_1 T_1 = F + F_{N1} + F_{T1} \]

\[ K_2 d_2 - \hat{K}_2 T_2 = F_{N2} + F_{T2} \]
Governing equations, contact interface – heat balance

\[ Q_{C1} = \varphi^0 p_n (T^2 - T^1) + \xi \mu p_n v \]
\[ Q_{C2} = \varphi^0 p_n (T^1 - T^2) + (1 - \xi) \mu p_n v \]

\[ M_1 \dot{T}_1 + O_1 T_1 = Q + Q_{C1}(T, P_n) \]
\[ M_2 \dot{T}_2 + (N + R + O_2) T_2 = Q_{C2}(T, P_n) \]

\[ Q_{c1}(T, P_n) = S_{P2}(P_n) T_2 - S_{P1}(P_n) T_1 + Q_{\mu}^1(P_n) \]
\[ Q_{c2}(T, P_n) = S_{P1}(P_n) T_1 - S_{P2}(P_n) T_2 + Q_{\mu}^2(P_n) \]

\[ S_{Pm}^A = \begin{bmatrix} 0 & [\varphi^0 P_n]^A & 0 \end{bmatrix} \]
Specify initial temperatures of disc and pad.

Thermoelastic contact problem is solved, contact pressure distribution is determined and frictional power is established.

Heat transfer problem is solved and new nodal temperatures are determined.

$t=0$

$t=t+\Delta t$
Benchmark 2D
Simulation of heat bands in disc brakes
Temperature history – no wear

Test 12-Thermography camera  In-house software
Temperature history – including wear

Test 12-Thermography camera

In-house software
Contact pressure

First cycle

After a period of running in.
Wear on pad surface

First cycle

(a) $t = 17.5\ s$

(b) $t = 25\ s$

(c) $t = 35\ s$

(d) $t = 45\ s$

After a period of running in.
Convex bending

Support Plate

Brake Pad
Stress analysis – decoupled approach

Frictional heat analysis

Stress analysis

- Input File
- In-house Software
- ODB File
- Abaqus
- ODB file
Temperature dependent material data

- 20°C
- 200°C
- 400°C
- 600°C
- 800°C
- 1120°C
Residual stresses – simulation & experiments
Circumferential stresses
## Performance

<table>
<thead>
<tr>
<th></th>
<th>Run Time (hours)</th>
<th>Brake Time (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sequential Approach</td>
<td>3.8</td>
<td>45</td>
</tr>
<tr>
<td>Fully Coupled Approach</td>
<td>480</td>
<td>5</td>
</tr>
</tbody>
</table>

**Input File**

**In-house Software**

**ODB File**

**Abaqus**

**ODB file**
Virtual test rig - dynamometer

\[ \begin{align*}
&d_n, P_n, T_n, \dot{\theta}_n \\
&\text{Contact analysis, (12)}_1 \\
&d_{n+1}, P_{n+1} \\
&\text{Multibody dynamics, (12)}_2 \\
&\ddot{\theta}_{n+1} \\
&\text{Heat transfer analysis, (12)}_3 \\
&T_{n+1}
\end{align*} \]
Friction model – temperature dependent

\[ \mu(T) = \mu_0 + \mu_1 T + \mu_2 T^2 \]
Virtual test data

(a) Clamping force

(b) Angular velocity

(c) Brake moment

(d) Maximum temperature
Design optimization

Minimize $T_{\text{max}}(\mathbf{x})$
Maximize $E_{\text{brake}}(\mathbf{x})$
Minimize $m(t)$

$17.15E + 3 \leq F \leq 24.5E + 3$
$2.2E + 8 \leq E \leq 2.2E + 9$
$5 \leq t \leq 14$
Concluding remarks

- A new idea for thermomechanical contact problems is suggested, derived and implemented.
- A toolbox for simulating brake disc is developed based on the new approach.
- A thermo-flexible multi-body model is suggested for simulating dynamometers for brake discs.
- A virtual test rig is developed by implementing the thermo-flexible multi-body model in the toolbox.
- A decoupled approach for simulating corresponding residual stresses is developed.
- Design optimization of the disc pad system is initiated.

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