Coupled Fluid-Thermal-Structural Modeling of Motorized Spindle to Reduce Thermal Distortion

Mallinath N. Kaulagi$^1$ Srinivas N. Grama$^2$
Ashok N. Badhe$^3$ J. Sharana Basavaraja$^4$

$^1,^4$B. M. S. College of Engineering, Bengaluru, KA, India
$^2,^3$Dr. Kalam center for innovation, Bharat Fritz Werner Ltd., Bengaluru, KA, India

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Outline

Introduction
Need for built-in motor spindle
Problem definition and objectives

Heat characterization by inverse techniques
Boundary conditions
Methodology
Heat source estimation

Structural analysis
Coupled fluid-thermal-structural analysis

Summary

Optimization of milling spindle
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Motorized spindle reduces transmission loss but compounds thermal effects

- Manufacturing industries - aiming at reducing production time and increasing productivity
- Conventional spindles (power transmission loss) are replaced by motorized spindle
- Thermal issues in motorized spindle because of built-in motor
- Needs external cooling to reduce thermal issues
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The problem statement is two fold: heat characterization and spindle coolant channel optimization

The aim of present work is to

- Estimate heat generation rate of motorized spindle in an experimental-numerical framework.
- Analyze the motorized spindle in coupled fluid-thermal-structural simulation framework and optimize the coolant flow channel for minimizing the thermal distortion
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External forced Coolant flow channel is used to reduce the temperature of spindle

Navier-stokes equation is used for fluid model
Forced air convection heat transfer coefficient \( (h_1) \) based on empirical relation [4]

\[
\text{Rotation (rpm)} \rightarrow \text{Re (laminar or turbulent)} \rightarrow \text{Nu} \rightarrow Nu = \frac{h_1 \times L}{k}
\]

Free air convection \( h_2 = 10 \text{ W/m}^2\text{K} \)

Boundary conditions

Thermal contact conductance (TCC) coefficient at contact interface based on empirical relation

The TCC at contact interface is given by [4]

\[ R = \frac{\delta_{race}}{k_{race}} + \frac{\delta_{gap} - (T_{race} - T_{h}) \times \gamma \times r_h}{k_{air}} \]

\[ C = \frac{1}{R} \]

TCC between outer race of bearing and housing

TCC between inner race of bearing and shaft

Iterative technique involved in Inverse Methodology [3]

Inverse Algorithm Validation for Spindle problem

Knowing sensor point temperature ($T_i$) and initial guess value ($Q_i$), Inverse Methodology is used to estimate Heat sources.

$T_1$, $T_2$ are outer race temperature of front & rear bearing, $T_3$ is motor surface temperature.
Solution converges to true value

Levenberg Marquadt Method: Irrespective of initial guess value, solution is converging towards true value with maximum error of 7.64%

True heat sources: $Q_1 = 250\ W$, $Q_2 = 100\ W$, $Q_3 = 250\ W$

![Graphs showing convergence of heat sources with lower and higher initial guess values](image)
Heat source estimation

Actual experimental setup [2]

Heat source estimation

Heat generation rate from spindle unit are estimated under steady state condition

Based on experimental temperature heat sources are estimated

<table>
<thead>
<tr>
<th>4,500 rpm</th>
<th>15,000 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_1 = 111.28 \text{ W}$</td>
<td>$Q_1 = 238.44 \text{ W}$</td>
</tr>
<tr>
<td>$Q_2 = 37.1 \text{ W}$</td>
<td>$Q_2 = 125.96 \text{ W}$</td>
</tr>
<tr>
<td>$Q_3 = 66.76 \text{ W}$</td>
<td>$Q_3 = 132.69 \text{ W}$</td>
</tr>
</tbody>
</table>

Higher heat generation is observed for higher spindle speed.
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Higher temperature is observed near rear bearing as it is not cooled

Temperature distribution of motorized spindle at 15,000 rpm
Coupled fluid-thermal analysis of spindle is validated from experimental temperatures measured near front bearing.
Non uniform temperature distribution observed in current fluid channel at front bearing region

Spindle and coolant channel analysed using COMSOL multi-physics

Maximum temperature variation across sections E-E and F-F for current spindle design is $2.98^\circ C$ & $4.26^\circ C$
Optimization of coolant channel leads to almost uniform temperature distribution near front bearings.

The coolant entry and exit angles near the front bearings are made diametrically opposite.

Maximum temperature variation across sections E-E and F-F for optimized spindle is 0.87°C & 1.4°C.
Boundary condition for structural analysis are provided through bearing stiffness and fixing collar of housing.

For current preload condition, radial stiffness is twice that of axial stiffness \[1\]

For front & rear bearings, \( K_a = 146.7 \times 10^6 \) & \( 121.6 \times 10^6 \) N/m, respectively, for \( 25^\circ \) contact angle, \( K_r = 2 \times K_a \)

\[1\] FAG manual, Schaeffler technologies (2010)
Optimized spindle shows reduced angular distortion through analysing coupled fluid-thermal-structural framework using COMSOL multi-physics.

<table>
<thead>
<tr>
<th>Distortion</th>
<th>Current spindle</th>
<th>Optimized spindle</th>
</tr>
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<tbody>
<tr>
<td>Axial ((\mu)m)</td>
<td>26.49</td>
<td>25.44</td>
</tr>
<tr>
<td>Angular ((\mu)rad)</td>
<td>12.8</td>
<td>3.23</td>
</tr>
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► Heat generation characterization using Levenberg-Marquardt method.
► Non-uniform temperature distribution near front bearings has been reduced from 2.5°C to around 1°C by coolant channel optimization.
► Spindle angular deformation is reduced from 12.8 to 3.23 μrad in the optimized design.
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Acknowledgements:

The authors like to thank Bharat Fritz Werner Limited, Bengaluru, for valuable support to conduct the above work and providing all the required lab facilities.
References:

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Thank you for your attention
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Additional Material
Iterative Steps involved in LMM

- Direct heat transfer: -analyse the model with initial guess value \( Q^T = [Q_1, Q_2, Q_3] \) are vector of unknown parameter

- Objective function, \( S(Q) = \sum_{i=1}^{M} [T_{\text{estimated}} - T_{\text{experimental}}]^2 \)

- Sensitivity matrix, \( J(Q) = \left[ \frac{\partial T^T(Q)}{\partial Q} \right] \)

- The \( \Delta Q \) for Levenberg-Marquadt method is calculated by

\[
((J^k)^T \times J^k + \mu^k \times \Omega^k) \times \Delta Q^k = (J^k)^T \times [T_{\text{experimental}} - T_{\text{estimated}}(Q^k)]
\]

where \( k \) represent iteration, \( \mu^k = \) damping parameter, \( \rho^k = \) diagonal matrix

- New value of parameter, \( Q^{k+1} = Q^k + \Delta Q^k \)
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New value of parameter, \( Q^{k+1} = Q^k + \Delta Q^k \)
Boundary conditions

Fourier law of heat conduction is used for solid heat transfer problem

- The three dimensional governing equation for steady state heat transfer

\[
\frac{\partial}{\partial x} \left[ \frac{\partial T}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \frac{\partial T}{\partial y} \right] + \frac{\partial}{\partial z} \left[ \frac{\partial T}{\partial z} \right] + \frac{Q_1}{k} + \frac{Q_2}{k} + \frac{Q_3}{k} = 0
\]

\( Q_1, Q_2, Q_3 \) are heat generation at front bearing, rear bearing and motor respectively.

- Heat transfer at contact interference \((Q_c)\)

\[
Q_c = C \times (A \times \Delta T)
\]

\( C = \) Thermal contact conductance, \( A = \) Contact area, \( \Delta T = \) Temperature drop at the contact interface
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Q₁, Q₂, Q₃ are heat generation at front bearing, rear bearing and motor respectively.

- heat transfer at contact interference (Qₖ)

\[
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- Convection heat transfer \( Q_{out} \) equation is

\[
Q_{out} = h \times (T(x, y, z) - T_\infty)
\]

\( Q_{out} \) = Heat dissipation due to convection,
\( h \) = Convective heat transfer coefficient,
\( T(x, y, z) \) = Temperature of spindle,
\( T_\infty \) = Surrounding temperature

- Radiation heat dissipation is neglected in the analysis
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