Topology Optimization of Gear Web using OptiStruct

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Overview

• Background
• Introduction
• Problem Setup
• Results
• Conclusion
A framework has been developed to study and optimize rotorcraft drive systems.
One of the biggest problems a designer is faced with is understanding what **fidelity** of analysis is required to obtain the **detail** of information needed.

The balance between fidelity and detail at different stages of the design life-cycle, among different disciplines, is one of the keenly researched topics in **MDAO**.

Weight saving is critical in aerospace applications

- How should **topology optimization** be treated in drive system design?
Introduction

• Gear web and non-standard gear designs can produce up to 30% weight saving.
• Gear webs have been designed in detailed stages through bench tests so far.
• Topology Optimization is a method that can be used to study this using FEA.
Since the design space is discrete, if after topology optimization, design B weighs less than design A, topology optimization cannot be considered a design refinement, instead must be used to select the optimum design.
## Problem Setup

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Unit</th>
<th>Baseline</th>
<th>Design 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>P</td>
<td>HP</td>
<td>700</td>
<td></td>
</tr>
<tr>
<td>RPM</td>
<td>-</td>
<td>RPM</td>
<td>5000</td>
<td></td>
</tr>
<tr>
<td>Gear ratio</td>
<td>(m_g)</td>
<td>-</td>
<td>3.3</td>
<td></td>
</tr>
<tr>
<td>Diametral pitch</td>
<td>(P_d)</td>
<td>(\text{in}^{-1})</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>Face width</td>
<td>F</td>
<td>in</td>
<td>2.4</td>
<td></td>
</tr>
<tr>
<td>Pressure angle</td>
<td>(\phi)</td>
<td>deg</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Material</td>
<td>-</td>
<td>-</td>
<td>AISI 9310</td>
<td></td>
</tr>
<tr>
<td>Number of teeth</td>
<td>(N_p)</td>
<td>-</td>
<td>23</td>
<td>24</td>
</tr>
<tr>
<td>Tangential load</td>
<td>(P_t)</td>
<td>lbf</td>
<td>4616.7</td>
<td>4424.4</td>
</tr>
<tr>
<td>Radial load</td>
<td>(P_n)</td>
<td>lbf</td>
<td>1680.4</td>
<td>1610.3</td>
</tr>
<tr>
<td>Bending Stress (AGMA)</td>
<td>(\sigma_b)</td>
<td>psi</td>
<td>35000</td>
<td>32951</td>
</tr>
<tr>
<td>Weight</td>
<td>W</td>
<td>lb</td>
<td>7.233</td>
<td>7.885</td>
</tr>
</tbody>
</table>
Problem Setup

- **Radioss run for calibrating load**
  - Gear model brought in from CATIA
  - Load for FEA application was calibrated to obtain analytical bending stress value
Problem Setup – Topology Optimization

• **Components**
  • Gear Teeth (blue)
  • Design Area (red)
  • Shaft hole (blue)

• **Mesh Setup**
  • Shaft hole and Design Area: Edge Deviation, 75 elements per edge
  • Gear Teeth: Q1 Optimize, element size 0.06
  • 3D Solid Map, size 0.06

• **Load Setup**
  • Load was distributed across a single line of nodes at pitch circle
  • Inside of Shaft hole constrained for 6 DoF

• **Calibrated Load**

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<th>Baseline</th>
<th>Design 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tangential load</td>
<td>$P_t$</td>
<td>lbf</td>
<td>6463.4</td>
<td>6194.2</td>
</tr>
<tr>
<td>Radial load</td>
<td>$P_n$</td>
<td>lbf</td>
<td>2352.6</td>
<td>2254.4</td>
</tr>
</tbody>
</table>
Objective: \( \min : F(x) = Volume_{design} \)

Subject to: \( x \in X = \{ x \in \mathbb{R}^n \mid g_i(x) \leq 0, \quad i = 1, \ldots, m \} \)

Where,

\[ g_1(x) = (f \times \sigma_{design}) - \sigma^* \]

\( g_2(x) \to \) manufacturing constraint (draw)

\( g_3(x) \to \) manufacturing constraint (cyclic and symmetry)
Problem Setup – Topology Optimization

- $\sigma_{\text{design}} < 20\text{ksi}$
- **Cyclic Pattern**, plane normal to axis of the gear, node (0,0,1.2) and node (0,0,0). Instances = multiple of teeth
  - Ensure cyclic symmetry since single tooth loading is simulated
- **Split Draw Constraint**, draw direction given by line through center of gear (0,0,1.2) and center of front face (0,0,0)
  - Ensure symmetry through the thickness for simpler manufacturing and ease of assembly
Problem Setup – Topology Optimization

- Optimized result only maintains structural integrity within design area
- Optimized result would lead to increased bending stress in gear teeth
Problem Setup – Topology Optimization

Objective: \( \min : F(x) = \text{Volume}_{\text{design}} \)

Subject to: \( x \in X = \{ x \in \mathbb{R}^n \mid g_i(x) \leq 0, \quad i = 1, \ldots, m \} \)

Where,
\[
\begin{align*}
g_1(x) &= (f \times \sigma_{\text{design}}) - \sigma^* \\
g_2(x) &= \sigma_b - \sigma_b^* \\
g_3(x) &\rightarrow \text{manufacturing constraint (draw)} \\
g_4(x) &\rightarrow \text{manufacturing constraint (cyclic and symmetry)}
\end{align*}
\]
Problem Setup – Topology Optimization

- **Optimization Responses**
  - Volume – Design region
  - $\sigma_b$ - Tooth bending stress

- **Optimization Constraint**
  - $\sigma_b < 35\text{ksi}$

- **Optimization Objective**
  - Minimize volume (design region)

- **Design Variable Constraints**
  - $\sigma_{\text{design}} < 20\text{ksi}$
  - Cyclic constraint (number of instances = number of teeth)
  - Split draw constraint
Problem Setup – Initial Results

N = 23
Cyclic instances = 23
Problem Setup – Initial Results

N = 24
Cyclic instances = 24
Problem Setup – Further Refinement

- Uniform gear web is desired for manufacturing and operation purposes
- Increase number of instance = \( i \times \text{number of teeth} \), \( i = 1, 2, 3 \ldots n \)
  - Number of instances for Baseline: \( 8 \times \text{number of teeth} = 184 \)
  - Number of instances for Design 2: \( 4 \times \text{number of teeth} = 96 \)
  - Eliminates holes through thickness
Results - Baseline

N = 23
Cyclic instances = 184
Results – Design 2

N = 24
Cyclic instances = 96
## Results

### Baseline vs. Design 2

<table>
<thead>
<tr>
<th>Weight</th>
<th>Baseline</th>
<th></th>
<th></th>
<th>Design 2</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Before</td>
<td>After</td>
<td>% Difference</td>
<td>Before</td>
<td>After</td>
<td>% Difference</td>
</tr>
<tr>
<td>Gear Teeth</td>
<td>3.777</td>
<td>3.777</td>
<td>-</td>
<td>3.972</td>
<td>3.972</td>
<td>-</td>
</tr>
<tr>
<td>Web</td>
<td>3.243</td>
<td>2.674</td>
<td>18%</td>
<td>3.691</td>
<td>2.731</td>
<td>26%</td>
</tr>
<tr>
<td>Shaft</td>
<td>0.213</td>
<td>0.213</td>
<td>-</td>
<td>0.222</td>
<td>0.222</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td>7.233</td>
<td>6.664</td>
<td>8%</td>
<td>7.885</td>
<td>6.925</td>
<td>12%</td>
</tr>
</tbody>
</table>
• Topology Optimization was used to estimate material removal in the gear web.
• For a gear designed to carry a specific torque, 12% weight saving was obtained.
• An overdesigned gear does not translate to a lighter gear, post-topology optimization, based on preliminary studies.
• Further work is required to understand feature manipulation, application for helical gears, and implementation of stiffness criteria.
THANK YOU

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