CHAPTER 6

EFFECT OF GEAR MATERIAL IN GEAR DESIGN

6.1 INTRODUCTION

When designing a gear reducer, there are many important factors to be considered such as weight, size, strength, durability material and geometry. Material of the gear reducer has a key impact on its weight. Buiga et al (2012) has proposed that gear materials have a key impact on the gear reducer’s weight. Hence, different types of gear materials are considered for spur and helical gear design optimization problems.

6.2 SPUR GEAR PROBLEM

The spur gear design problem is solved by using C15 and Alloy Steel (15Ni 2Cr 1Mo15) as gear materials. The test spur gear drive problem is considered as, ‘Design a spur gear drive to transmit 22.5 kW at 900 rpm. Gear ratio is 2.5. The gears are made of C15 steel’.

The complete optimized problem of spur gear drive in terms of design variables Power (P), Module (m), Gear thickness (b) and Number of teeth on Pinion (Z_1) for the above problem with C15 material, after simplification is,

Maximize \( f_1 = P \) where, \( P^{(L)} \leq P \leq P^{(U)} \) \hspace{1cm} (6.1)

Minimize \( f_2 = 4.47 \times 10^{-5} \times b \times (mZ_1)^2 \) \hspace{1cm} (6.2)

Subject to,
\[ mZ_1 \ b^{0.5} \ P^{-0.5} \geq 104.59 \]  \hspace{1cm} (6.3)

\[ m^2 \ (Z_1 + 8) \ b \ P^{-1} \geq 416.35 \]  \hspace{1cm} (6.4)

\[ m \ Z_1 \ P^{-0.333} \geq 27.52 \]  \hspace{1cm} (6.5)

\[ m^3 \ (Z_1+8)^{0.333} \ P^{-0.333} \geq 3.47 \]  \hspace{1cm} (6.6)

The material properties of Gear drive is tabulated in Table 6.1

### Table 6.1 Material Properties of Spur Gear drive

<table>
<thead>
<tr>
<th>Material</th>
<th>Density ((\rho)) (\text{kg/mm}^3)</th>
<th>Bending Stress ((\sigma_b)) (\text{N/mm}^2)</th>
<th>Compressive Stress ((\sigma_c)) (\text{N/mm}^2)</th>
<th>Young’s Modulus ((E)) (\text{N/mm}^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C-15</td>
<td>7.85 x 10^{-6}</td>
<td>110</td>
<td>610</td>
<td>1.7 x 10^5</td>
</tr>
<tr>
<td>Alloy Steel</td>
<td>8.80 x 10^{-6}</td>
<td>250</td>
<td>950</td>
<td>2.10 x 10^5</td>
</tr>
</tbody>
</table>

After required number of iteration performed by SFHM for the two different gear materials for the specified spur gear design problem, the optimized results were tabulated in Table 6.2 in compared with TM.

### Table 6.2 Comparison of Spur gear optimized results by SFHM

<table>
<thead>
<tr>
<th>Parameters / Material</th>
<th>TM</th>
<th>SFHM</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>C-15</td>
<td>Alloy Steel</td>
<td>C-15</td>
<td>Alloy Steel</td>
</tr>
<tr>
<td>Power (P) kW</td>
<td>22.50</td>
<td>22.50</td>
<td>22.622</td>
<td>22.651</td>
</tr>
<tr>
<td>Module (m) mm</td>
<td>5.00</td>
<td>5.00</td>
<td>4.82</td>
<td>4.72</td>
</tr>
<tr>
<td>Gear Thickness (b) mm</td>
<td>47.25</td>
<td>47.25</td>
<td>42.862</td>
<td>38.720</td>
</tr>
<tr>
<td>No. of teeth on pinion ((Z_1))</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
</tr>
<tr>
<td>Gear Weight (kg)</td>
<td>17.11</td>
<td>17.73</td>
<td>14.323</td>
<td>14.02</td>
</tr>
</tbody>
</table>
By varying all design parameters such as power, thickness, number of teeth and module, SFHM performs well and shows a huge reduction of 21% in gear weight compared with trail method and nominal power increase. Alloy steel
shows better results between the two materials with increase in power and reduction of weight.

### 6.3 A HELICAL GEAR PROBLEM

The helical gear design problem states that, Design a pair of helical gears to transmit 12.5 kW at 1200 rpm. The transmission ratio is 3.5 and material adapted as Alloy steel material 40Ni 2Cr1Mo28. The helix angle is 15º and pressure angle is 20º. The above helical gear design is to be solved with two different gear materials, such as Carbon Steel C40 and Alloy Steel (40Ni 2Cr1Mo28). The material properties of gear drive are tabulated in Table 6.3.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density ((\rho)) kg/mm(^3)</th>
<th>Bending Stress ((\sigma_b)) N/mm(^2)</th>
<th>Compressive Stress ((\sigma_c)) N/mm(^2)</th>
<th>Young’s Modulus(E) N/mm(^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C-40</td>
<td>7.8 x 10(^{-6})</td>
<td>120</td>
<td>750</td>
<td>2 x 10(^{5})</td>
</tr>
<tr>
<td>Alloy Steel</td>
<td>8.89 x 10(^{-6})</td>
<td>400</td>
<td>1100</td>
<td>2.15 x 10(^{3})</td>
</tr>
</tbody>
</table>

The Simplified complete helical design problem is,

\[
\text{Maximize } f_1 = P \text{ where, } P^{(L)} \leq P \leq P^{(U)} \tag{6.7}
\]

\[
\text{Minimize } f_2 = 98.58 \times 10^{-6} \times b \times (m_nZ_1)^2 \tag{6.8}
\]

Subject to,

\[
m_nZ_1 \ b^{0.5}P^{-0.5} \geq 65.741 \tag{6.9}
\]

\[
m_n^2 (3.5Z_1 + 20) b P^{-1} \geq 250.51 \tag{6.10}
\]
\[
m_n Z_1 P^{-0.333} \geq 18.35 \quad (6.11)
\]
\[
m_n^3 (3.5Z_1+20)^{0.333} P^{-0.333} \geq 2.94 \quad (6.12)
\]

After required number of iteration performed by SFHM for the two different gear materials for the specified helical gear design problem, the optimized results were tabulated in Table 6.4 in compared with existing design.

6.3.1 Illustration of Traditional Design Method for Helical Drive

Traditional design method adopted from Design Data book (2012)

a. Calculation of initial design torque

\[
[M_t] = M_t \times K \times K_d
\]

Where \( M_t = \frac{60 \times P}{2 \pi N} = \frac{60 \times 12.5 \times 10^3}{2 \pi N} = 99.47 \text{ N-m} \)

Assume \( K.K_d = 1.3 \)

\( [M_t] = 99.46 \times 1.3 = 129.31 \text{ N-m} \)

b. Calculation of \([\sigma_b]_{al}\) and \([\sigma_c]_{al}\)

\[
[\sigma_b]_{al} = \frac{1.4 K_{bl}}{n K_{\sigma}} x \sigma_{-1}
\]

\( K_{bl}= 0.7 \) for HB > 350

\( K_{\sigma}= 1.5 \) for Steel hardened
\[ n = 2.5 \text{ for Steel hardened} \]

\[ \sigma_1 = 0.35 \sigma_u + 120 \text{ for alloy steel} \]

Where \( \sigma_u = 1550 \text{ N/mm}^2 \)

\[ \sigma_1 = 0.35 \times 1550 + 120 = 662.5 \text{ N/mm}^2 \]

\[ [\sigma_b]_{al} = \frac{1.4 \times 0.7}{2.5 \times 1.5} \times 662.5 = 173.133 \text{ N/mm}^2 \]

\[ [\sigma_c]_{al} = C_R \times \text{HRC} \times K_{cl} \]

\( C_R = 26.5 \text{ for alloy steel} \)

\( \text{HRC} = 40 \text{ to } 55 \)

\( K_{cl} = 0.585 \), for steel \( \text{HB} > 350 \)

\[ [\sigma_c]_{al} = 26.5 \times 55 \times 0.585 = 852.6 \text{ N/mm}^2 \]

c. Calculation of Centre distance (a)

\[ a \geq (i + 1) \sqrt[3]{\left[ \frac{0.7}{\sigma_c} \right]^2 \times \frac{E[M_i]}{i\Psi}} \]

\[ \Psi = b/a = 0.3 \]

\[ a \geq (3.5 + 1) \sqrt[3]{\left[ \frac{0.7}{852.6} \right]^2 \times \left( \frac{2.15 \times 10^5 \times 129.31 \times 10^3}{3.5 \times 0.3} \right)} \]
a ≥ 117.6 mm = 120mm
Assume z1 = 20, then z2 = i x z1 = 3.5 x 20 = 70

d. Calculation of normal module($m_n$)

\[
m_n = \frac{2a}{(Z_1 + Z_2)} \times \cos \beta = \frac{2 \times 120}{(20 + 70)} \times \cos 15^\circ = 2.576 \text{mm}
\]

the nearest higher standard normal module is 3mm.

e. Revision of centre distance

\[
a = \frac{m_n}{2 \cos \beta} (Z_1 + Z_2) = \frac{3}{2 \cos 15^\circ} (20 + 70) = 139.76 \text{mm}
\]

f. Calculation of b and $\Psi_p$

\[
b = \Psi \times a = 0.3 \times 139.76 = 41.93 = 42 \text{mm}
\]

\[
\Psi_p = \frac{b}{d_1} = \frac{42}{62.12} = 0.676; \quad v = \frac{\pi dN}{60} = 3.9 \text{ m/s}
\]

g. Revision of design torque

\[
[M_t] = M_t \times K \times K_d
\]
K = 1.045 for \( \Psi_p = 0.676 \)

\[ K_d = 1.2 \]

\[ [M_t] = 99.47 \times 1.045 \times 1.2 = 124.74 \text{ N-m} \]

h. Check for bending stress

\[
\sigma_b = 0.7 \left( \frac{i+1}{(\text{am}_n \text{ by})} \right) \times [M_t] \quad (4.38)
\]

\[
z_{v1} = \frac{Z_t}{\cos^3 \beta} = \frac{20}{\cos^3 15^\circ} = 22
\]

\( y = 0.402 \) for \( z_{v1} = 22 \)

\[
\sigma_b = 0.7 \left( \frac{3.5+1}{139.76 \times 42 \times 3 \times 0.402} \right) \times 124.47 \times 10^3 = 55.5 \text{ N/mm}^2
\]

\( \sigma_b \leq [\sigma_b]_{al} \). Thus design is satisfactory

i. Check for Compressive stress:

\[
\sigma_c = 0.7 \times \left( \frac{i+1}{a} \right) \times \sqrt{\left( \frac{i+1}{ib} \right) \times E \times [M_t]} 
\]

\[
\sigma_c = 0.7 \times \left( \frac{3.5+1}{139.76} \right) \times \sqrt{\left( \frac{3.5+1}{3.5 \times 42} \right) \times 2.15 \times 10^5 \times 124.74 \times 10^3} 
\]

\( = 645.8 \text{ N/mm}^2 \)
\[ \sigma_c \leq [\sigma_c]_{al} \]. Thus design is satisfactory.

Table 6.4 Comparison of Helical gear drive optimized results by SFHM

<table>
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<tr>
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</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Alloy Steel</td>
<td>C-40 Alloy Steel</td>
</tr>
<tr>
<td>Power (P) kW</td>
<td>12.512</td>
<td>12.574</td>
</tr>
<tr>
<td>Normal Module (m_n) mm</td>
<td>3.00</td>
<td>2.86</td>
</tr>
<tr>
<td>Gear Thickness (b) mm</td>
<td>42.00</td>
<td>37.132</td>
</tr>
<tr>
<td>No. of teeth on pinion (Z_1)</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Gear Weight (kg)</td>
<td>14.74</td>
<td>10.569</td>
</tr>
</tbody>
</table>

From the Table 6.4, SFHM performs well and shows a huge reduction in gear weight. Alloy steel shows better results among the two materials in power, better reduction of weight when compared with the existing design.

6.4 SUMMARY

In this work, two test problems have been considered for the design optimization of gear pairs. One problem is on spur gear pair and other is on helical gear pair with different gear materials. The SFHM have been implemented to solve these problems and the results are compared. Numerical illustration for the helical gear is discussed in this chapter. The two stage gear box design optimization is discussed in next chapter.