Evaluation on Failure Analysis of an Automobile Differential Pinion Assembly

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Abstract

Bevel gears have become a subject to research interest because the dynamic load, attention of the noise level during operation and demand for lighter and smaller. In such type of gears there is a problems of failures contact at meshing the teeth. This can be avoided or minimized by proper method analysis and modification of the different gear parameters. This thesis presents characteristics of a bevel gear in dynamic condition involving meshing stiffness and other stresses produce. The purpose of this thesis is by using numerical approach to develop theoretical model of bevel gear and to determine the effect of meshing gear tooth stresses by taking material case hardened alloy steel (15Ni4Cr1). To estimate the meshing stiffness, three-dimensional solid models for different number of teeth are generated by Solid works and the numerical solution is done in Ansys which is a finite element analysis.

Keywords: Design of Spiral Bevel Gear, Analysis of Spiral Bevel Gear

I. INTRODUCTION

Gear is a mechanical device used in transmission systems that allows rotational force to be transferred to another gears. The gear teeth allow force to be fully transmitted without slip and depending on the configuration can transmit forces at different speeds, torques, and even in a different directions. Bevel gears are widely used because of the suitability towards transferring power between nonparallel shafts at almost any angle and speed. Spiral bevel gears have curved and sloped gear teeth in relation to the surface of the pitch cone. In bevel gear will have a tangential load, radial load and axial load due to the speed and torque. This will be a transient phenomenon and will need careful stress analysis for determining life of the gears.

II. LITERATURE REVIEW

Luciana Sgarbi Rossinoa An investigation was made to determine the causes of surface contact fatigue failure of a case hardened pinion. The examination of the component revealed the presence of a cemented layers substantially thicker than that generally specified for pinions devised for this application. This associate with the massive presence of brittle threadlike carbon-rich cementite phase (Fe3C) in prior austenite grain boundaries of the pinion teeth favored surface crack nucleation and propagation during cyclic loading leading to spallation of the contact surface with the counterpart gear which impaired the system's operation.

C.Veeranjaneyulu The main aim of this paper is to focus on the mechanical design and analysis on assembly of gears in gear box when they transmit power at different speeds i.e. 2500 rpm, 5000 rpm, 7500 rpm. Analysis is also conducted by varying the materials for gears Cast Iron, Cast Steels, Aluminum Alloy. The analysis is done in Cosmos software. It’s a product of Solid works. And also weight of the Aluminum alloy reduces almost 3 times when compared with Alloy Steel and Cast Iron since its density is very less. Analysis results, Aluminum Alloy is the best material.

Daniel Das.A In this paper is to focus on the mechanical design and analysis on assembly of gears in gear box when they transmit power at different speeds. Analysis also conducted by varying the materials for gears, Cast Iron, Cast Steels, Aluminum Alloy etc. The analysis done in ANSYS software. So using Aluminum Alloy is safe for differential gear. Comparing the stress values of the three materials for all speeds 2000rpm, 4500rpm, 6000 rpm, the values are less for Aluminum alloy than Alloy Steel and Cast Iron. And also weight of the Aluminum alloy reduces almost 3 times. By observing analysis results, Aluminum Alloy best material for Differential.
III. METHODOLOGY

Fig. 1: Flow Chart

Table 1: Theoretically Design of Bevel gear

<table>
<thead>
<tr>
<th>Input Parameters</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Input power (P)</td>
<td>46.3 kW</td>
</tr>
<tr>
<td>Input Pinion Speed (N_p)</td>
<td>3200 rpm</td>
</tr>
<tr>
<td>Input Gear Speed ((N_g)</td>
<td>1060 rpm</td>
</tr>
</tbody>
</table>

Constraints for Bevel Set

| Pinion teeth (Z1)                        | 15    |
| Gear teeth (Z2)                          | 45    |
| Angle between the two shaft (θ_s)        | 90°   |
| Pressure Angle (α)                       | 20°   |

1) Pitch angle for pinion (θ₁):

\[
\tan \theta_1 = \frac{d1}{d2} = \frac{z1}{z2} = \frac{1}{i} \\
\theta_1 = \tan^{-1}\left(\frac{15}{45}\right) \\
\theta_1 = 18.4^\circ
\]

2) Pitch angle for gear (θ₂):

\[
\theta_2 = 90^\circ - \delta_1 \\
= 90^\circ - 18.14^\circ \\
\theta_2 = 71.6^\circ
\]

3) Formative Number of theeth for Pinion (T_EP):

\[
T_{EP} = T_p \times sec \times \theta_k \\
= 15 \times sec \times 18.43
\]
4) Formative Number of teeth for Gear \((T_{EG})\):
\[
T_{EG} = T_G \times \sec \theta_2
\]
\[
= 45 \times \sec 71.57
\]
\[
T_{EG} = 142.83^\circ
\]

5) For 20° full depth involute teeth from factor.
Pinion \((Y'_p)\):
\[
Y'_p = 0.124 - \frac{0.684}{15.80^\circ}
\]
\[
= 0.124 - 0.044
\]
\[
= 0.080
\]

6) Gear \((Y'_g)\):
\[
Y'_g = 0.124 - \frac{0.684}{142.83^\circ}
\]
\[
= 0.124 - 0.0047
\]
\[
= 0.119
\]
\[
\theta_{op} \times Y'_p = 350 \times 0.080 = 28
\]
\[
\theta_{og} \times Y'_g = 350 \times 0.119 = 41
\]

7) Pinion is Weaker
Torque on the Pinion \((T)\):
\[
T = \frac{P \times 60}{2 \times \pi \times N_p}
\]
\[
= \frac{46.3 \times 10^3 \times 60}{2 \times \pi \times 3200}
\]
\[
= 138166 \text{ N/mm}
\]

8) Tangential Load on Pinion \((W_T)\):
\[
W_T = \frac{T}{D_p/2}
\]
\[
= \frac{138166}{75/2}
\]
\[
W_T = 3684 N
\]
\[
V = \frac{60}{\pi \times D_p \times N_p}
\]
\[
= \frac{60}{\pi \times 37.5 \times 3200}
\]
\[
= 6.28 \text{ m/sec}
\]

Modul is 5mm is \(e = 0.055\)
K = 0.107 for 14 \(\frac{1}{2}\) composite teeth \(E_p = 210 \times 10^3 \text{ N/mm} \), \(E_g = 84 \times 10^3 \text{ N/mm} \)
Dynamic factor \((C)\):
\[
C = \frac{K \times e}{\frac{1}{E_p} + \frac{1}{E_g}}
\]
\[
= \frac{0.107 \times 0.055}{210 \times 10^3 + 84 \times 10^3}
\]
\[
= 0.000353 \text{ N/mm}
\]

9) Dynamic Load on the Pinion \((W_D)\):
\[
W_D = W_T + \frac{21 \times V(b \times C + W_T)}{21 \times V + \sqrt{b \times C \times W_T}}
\]
\[
= 3684 + \frac{21 \times 6.2 \times (32 \times 353 + 3684)}{21 \times 6.2 + \sqrt{6.2 \times 353 + 3684}}
\]
\[
W_D = 11485 N
\]

10) Static tooth load or endurance strength of the teeth \((W_s)\):
\[
W_s = \sigma_e \times b \times \pi \times m \times Y'_p
\]
\[
= 350 \times 32 \times \pi \times 5 \times 0.080
\]
\[
W_s = 10423.8 N
\]

11) \(e = 0.015\)mm for module 5 mm
Dynamic factor \((c)\):
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\[ C = \frac{K \times e}{\frac{1}{E_P} + \frac{1}{E_G}} \]
\[ = \frac{0.107 \times 0.015}{1} \]
\[ = \frac{1}{210 \times 10^3} + \frac{1}{84 \times 10^3} \]
\[ = 96 \, N/mm \]

12) Dynamic Load on the Pinion (\(W_D\)) :

\[ D_W = W_T + \frac{21 \times V (b \times c + W_T)}{21 \times V + \sqrt{b \times c \times W_T}} \]
\[ = 3684 + \frac{21 \times 6.2(32 \times 96 + 3684)}{21 \times 6.2 + \sqrt{6.2 \times 96 + 3684}} \]
\[ W_D = 7845 \, N \]

13) Factor of safety :

\[ FOS = \frac{W_D}{W_G} = \frac{7845}{10423.8} = 0.75 \]

14) Check for wear load (\(\sigma_{es}\)) :

\[ \sigma_{es} = 661 \, Mpa = 661 \, N/mm \]

15) Load stress factor (k) :

\[ k = \frac{(\sigma_{es})^2 \times \sin \theta}{1.4} \times \left[ \frac{1}{210 \times 10^3} + \frac{1}{84 \times 10^3} \right] \]
\[ = 0.9 \, N/mm^2 \]

16) Ratio factor (Q) :

\[ Q = \frac{2 \times T_{EG}}{T_{EG} + T_{EP}} \]
\[ = \frac{142.83 + 15.82}{\sqrt{142.83}} \]
\[ Q = 1.8 \]

17) Maximum or limiting load for wear (\(W_w\)) :

\[ W_w = D_c \times b \times Q \times k \]
\[ = 180 \times 32 \times 1.8 \times 0.9 \]
\[ W_w = 11664 \, N \]

IV. RESULTS AND DISCUSSIONS

A. Ansys Analysis (Pinion Assembly):

1) 15Ni4Cr1 Material:

![Fig. 4.1: Equivalent Stress](image1)

![Fig. 4.2: Total Deformation](image2)
Evaluation on Failure Analysis of an Automobile Differential Pinion Assembly

Table 1: Ansys Analysis 15Ni4Cr1 (Pinion Assembly)

<table>
<thead>
<tr>
<th>Material</th>
<th>Equivalent Stress (Mpa)</th>
<th>Total Deformation (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Minimum</td>
</tr>
<tr>
<td>15Ni4Cr1</td>
<td>0.12045</td>
<td>0.00044056</td>
</tr>
<tr>
<td>Crown</td>
<td>0.28593</td>
<td>0.00031316</td>
</tr>
</tbody>
</table>

B. AISI 8620 Material:

Table 1: Ansys Analysis 15Ni4Cr1 (Pinion Assembly)

<table>
<thead>
<tr>
<th>Material</th>
<th>Equivalent Stress (Mpa)</th>
<th>Total Deformation (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Minimum</td>
</tr>
<tr>
<td>15Ni4Cr1</td>
<td>0.12045</td>
<td>0.00044056</td>
</tr>
<tr>
<td>Crown</td>
<td>0.28593</td>
<td>0.00031316</td>
</tr>
</tbody>
</table>
Table - 2
Anslysis Analysis AISI 8620 (Pinion Assembly)

<table>
<thead>
<tr>
<th>Material</th>
<th>Pinion</th>
<th>Equivalent Stress(Mpa)</th>
<th>Total Deformation(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AISI 8620</td>
<td></td>
<td>Maximum</td>
<td>Minimum</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.11222</td>
<td>0.00061371</td>
</tr>
<tr>
<td>Crown</td>
<td></td>
<td>0.28068</td>
<td>0.00030712</td>
</tr>
</tbody>
</table>

C. AISI 4130:

Fig. 4.7.9: Equivalent Stress  
Fig. 4.7.10: Total Deformation

Fig. 4.7.11: Equivalent Stress  
Fig. 4.7.12: Total Deformation

Table – 3
Anslysis Analysis AISI 4130 (Pinion Assembly)

<table>
<thead>
<tr>
<th>Material</th>
<th>Pinion</th>
<th>Equivalent Stress(Mpa)</th>
<th>Total Deformation(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AISI 4130</td>
<td></td>
<td>Maximum</td>
<td>Minimum</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.11731</td>
<td>0.00042726</td>
</tr>
<tr>
<td>Crown</td>
<td></td>
<td>0.27107</td>
<td>0.000350.55</td>
</tr>
</tbody>
</table>

D. Comparision of Different Material & Wearload:

Table –4
Comparision of Different Material & Wearload (Pinion Assembly)

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Material</th>
<th>15°/45</th>
<th>13°/45</th>
<th>15°/45</th>
<th>Ansys</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>15N4Cr1</td>
<td>11664 N</td>
<td>16416 N</td>
<td>12045 N</td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>4130</td>
<td>10368 N</td>
<td>10944 N</td>
<td>11731 N</td>
<td></td>
</tr>
<tr>
<td>3.</td>
<td>8620</td>
<td>9072 N</td>
<td>9576 N</td>
<td>11222 N</td>
<td></td>
</tr>
</tbody>
</table>
V. CONCLUSION

The existing pinion is manufactured by AISI 4130 steel material. The existing bevel pinion is redesigned by considering 15Ni4Cr1 steel as the pinion material. Decrease in the number of teeths on pinion leads to the increase in torque at the output. Calculating the design parameters for AISI 4130 steel material gives the margin of safety value of 0.66 for 15 teeth. Similar calculations are done for 15Ni4Cr1 steel material which gives margin of safety value of 0.75 for 15 teeth. Similar calculations are done for 15Ni4Cr1 steel material which gives margin of safety value of 0.92 for 13 teeth. From this we can conclude that margin of safety is high even though after reducing 2 tooth on pinion. Reduction of teeth also reduces the weight of the pinion.

REFERENCES