Design of Synchronomesh Mechanism and Stress Analysis of Gear for Hijet

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Abstract -- The gears are generally used to transmit power and torque. Gears are one of the most critical components in mechanical power transmission systems. The efficiency of power transmission is very high when compared to other kind of transmission. In the gear design the bending stress and surface strength of the gear tooth are considered to be one of the main contributors for failure of the gears in gear set. Thus, analysis of stresses has become popular as an area of research on gears to minimize or to reduce the failures and for optimal design of gears. In this paper bending stress at the root of the helical gear tooth can be calculated by using analytical method which is calculated by the Lewis stress formula. In this project, helical gear is designed by using solid works 2016 and the numerical solution is done by ANSYS, which is a finite element analysis package. The main objective of this research has to reduce the stresses induced in gear tooth profile by changing five different helix angles (13°, 18°, 23°, 28° and 33°). In this paper, 23° of helix angle is selected for helical gear used in synchronomesh gearbox according to the ranges of helix angles and AISI 5160 QQT 400 is chosen for helical gear the maximum von-Mises stress and strain values of AISI 5160 QQT 400 are less than other two materials. The results are then compared with both the Lewis equation and ANSYS procedure.

Indexed Terms: Bending stress, helical gear, Helix angle, Lewis equation, surface strength

I. INTRODUCTION

The power developed inside the engine cylinder is ultimately aimed to turn the wheels so that the motor vehicle can move on the road. The reciprocating motion of the piston turns a crankshaft which rotates the flywheel through the connecting rod. The circular motion of the crankshaft is now to be transmitted to the rear wheels. It is transmitted through the clutch, gearbox, universal joints, propeller shaft or drive shaft, differential and axles extending to the wheels. The application of engine power to the driving wheels through all these parts is called power transmission. The power transmission system is usually the same on all modern passenger cars and trucks, but its arrangement may vary according to the method of drive and type of the transmission units. Figure 1 shows the power transmission of an automobile. The motion of the crankshaft is transmitted through the clutch to the gear box or transmission, which consists of a set of gears to change the speed. From gear box, the motion is transmitted to the propeller shaft through the universal joint and then to the differential through another universal joint. The differential provides the relative motion to the two rear wheels while the vehicle is taking a turn.

Fig. 1: Automobile power transmission system [6]

Thus the power developed in the engine is transmitted to the rear wheels through a system of transmission. The gear box is necessary in the transmission system to maintain engine speed at the most economical value under all conditions of vehicle movement [3]. A gearbox as usually used in the transmission system is also called a speed reducer, gear head, gear reducer etc., which consists of a set of gears, shaft and bearings that are factory mounted in an enclosed lubricated housing. Speed reducers are available in a broad range of sizes, capacities and speed ratios. Their job is to convert the input power into the output power with lower speed and correspondingly higher torque [8]. There are two types of transmission, namely manual transmission and automatic transmission. Manual transmission type is used so many cars in various models and locations. If the car has a manual gear box, the driver moves the shift lever to shift gears. Automatic
gearbox does the job automatically without any effort
by the driver. Manual transmission consists of a cast
iron or aluminium housing, shafts, bearings, gears,
synchronizing devices and shifting mechanisms.
Automatic transmission includes a torque converter,
compound planetary gear set, two or more disc
clutches and one or more bands. In an automatic
transmission model, the clutch and gear shifting
operation is done automatically to the best suit with
the driving condition and torque is converted
continuously. These transmission systems are
efficient, convenient, easy to operate, durable and
reliable, but they are relatively expensive to
manufacture and service compared to standard
transmission system.

Synchromesh gear box is an extension of constant
mesh gear box where synchronizers are used instead
dog clutches to avoid noise and jerk. Figure 2
shows the Synchromesh gearbox [5]. In this types of
gearbox, all the gears of the main shaft are in
constant mesh with corresponding gears of the
countershaft. The gears on the main shaft which are
bushed are free to rotate. The synchronizers are
provided on main shaft. The gears on the lay shaft
are, however, fixed. When the left synchronizer is
slid to the left by means of the selector mechanism,
its teeth are engaged with those on the clutch gear
and direct gear can be obtained. The same
synchronizer, however, when slid to right makes
contact with the third gear and third gear is obtained.
Generally two types of gear are used in synchromesh
gear box include are spur gear and helical gear. In
spur gears, the teeth are parallel to the axis whereas
in helical gears the teeth are inclined to the axis. Both
the gears transmit power between two parallel shafts.

However in this paper, helical gear is used for
synchronmesh gear box because their contact ratio is
higher than spur gears and they operate smoother and
quieter than spur gears [7].
II. DESIGN CALCULATION OF HELICAL GEAR PAIR

The design calculation of the helical gear pair has the following steps. The material for the gear pairs is taken as AISI 5160 OQT400. TABLE I shows the parameters considered for design a helical gear.

<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power (P)</td>
<td>20 kW</td>
</tr>
<tr>
<td>R.P.M (Np)</td>
<td>6500</td>
</tr>
<tr>
<td>Helix angle (ψ)</td>
<td>23°</td>
</tr>
<tr>
<td>Pressure angle (ϕ)</td>
<td>20°</td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>207 GPa</td>
</tr>
<tr>
<td>Ultimate Strength</td>
<td>2220 MPa</td>
</tr>
<tr>
<td>Yield Strength</td>
<td>1790 MPa</td>
</tr>
<tr>
<td>Brinell Hardness (BHN)</td>
<td>627</td>
</tr>
<tr>
<td>Number of teeth of pinion</td>
<td>11 mm</td>
</tr>
<tr>
<td>Number of teeth of gear</td>
<td>34 mm</td>
</tr>
<tr>
<td>Poisson ratio (ν)</td>
<td>0.3</td>
</tr>
</tbody>
</table>

Table 1: Parameters considered for design a helical gear

Pinion and gear are same material and so pinion is weaker. So based design on pinion.

1. Unknown diameter case
The actual induced stress can be calculated by using Lewis equation.

\[ S_{\text{ind}} = \frac{2M_t}{m \times \kappa \times \gamma_p \times n_p \times \cos \psi} \]  

2. Calculation of Torque (M_t)

\[ M_t = \frac{9550 \times P}{N_p} \]  

3. Calculation of pitch line velocity (V)

\[ V = \frac{\pi \times D_p \times N_p}{60} \]  

4. Calculation of allowable stress (S_{all})

\[ S_{\text{all}} = S_0 \times \frac{5.6}{5.6 + \sqrt{V}} \]  

5. Calculation of endurance stress (S_o)

\[ S_o = \frac{S_u}{3} \]  

6. Calculation of number of teeth

\[ n = \frac{D}{m} \]  

7. Strength Check

Compare \( S_{\text{all}} \) and \( S_{\text{ind}} \)

If \( S_{\text{all}} > S_{\text{ind}} \), Design is satisfied. If not so, keeping on calculating by increasing the module until it is satisfied need to be done.

8. Calculation of the face width of helical gear

\[ b_{\text{min}} = k_{\text{red}} \times \pi \times m \]  

\[ b_{\text{max}} = k \times \pi \times m \]  

\[ k_{\text{red}} = k_{\text{max}} \times \frac{S_{\text{ind}}}{S_{\text{all}}} \]  

After determining the design from strength point of view, it is necessary to check the dynamic effect.

9. Dynamic Check

\[ F_l = \frac{2M_t}{D_p} \]
Calculation of dynamic load \( (F_d) \)

\[
F_d = F_t + \frac{21V(b\cos^2\psi + F_r)\cos\psi}{21V + \sqrt{(b\cos^2\psi + F_r)}}
\]

Calculation of limiting endurance load \( (F_0) \)

\[
F_0 = S_0b\pi m \cos(\psi)
\]

Calculation of limiting wear load \( (F_w) \)

\[
F_w = \frac{D_p \times b \times K \times Q}{\cos^3\psi}
\]

\[
K = \frac{S_e^2 \times \sin\phi_{h.d}}{1.4} \times \left[ \frac{2}{E} \right], \quad S_e = (2.75BHN - 70) \text{MN/m}^2
\]

\[
Q = \frac{2 \times D_e}{D_g + D_p}
\]

\[
\tan\phi_{h.d} = \tan\psi \cos\psi
\]

Table 2: Design result data for helical gear pair

<table>
<thead>
<tr>
<th>No. of teeth</th>
<th>Symbol</th>
<th>Pinion</th>
<th>Gear</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( n )</td>
<td></td>
<td>11</td>
<td>34</td>
<td>-</td>
</tr>
<tr>
<td>Pitch circle diameter</td>
<td>( D )</td>
<td>27.5</td>
<td>85</td>
<td>mm</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>( D_0 )</td>
<td>32</td>
<td>89</td>
<td>mm</td>
</tr>
<tr>
<td>Root diameter</td>
<td>( D_R )</td>
<td>21</td>
<td>78</td>
<td>mm</td>
</tr>
<tr>
<td>Face width</td>
<td>( b )</td>
<td>21</td>
<td>21</td>
<td>mm</td>
</tr>
<tr>
<td>Module</td>
<td>( m )</td>
<td>2.5</td>
<td>2.5</td>
<td>mm</td>
</tr>
<tr>
<td>Speed</td>
<td>( N )</td>
<td>6500</td>
<td>5000</td>
<td>rpm</td>
</tr>
</tbody>
</table>

The required condition to satisfy the dynamic check is \( F_0, F_w > F_d \). If not, keeping on calculating by increasing the module until it is satisfied need to be done [6].

### III. MODELLING OF HELICAL GEAR

In this paper, AISI 5160 OQT 400 is used as the helical gear materials. The material properties of AISI 5160 OQT 400 is given in the TABLE III.

Table 3: Properties of AISI 5160 OQT 400 helical gear

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young modulus</td>
<td>207 GPa</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>0.3</td>
</tr>
<tr>
<td>Density</td>
<td>7850 kg/m³</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion</td>
<td>1.15e-05</td>
</tr>
<tr>
<td>Tensile Yield Strength</td>
<td>2220 MPa</td>
</tr>
<tr>
<td>Tensile Ultimate Strength</td>
<td>1790 MPa</td>
</tr>
</tbody>
</table>

Finite Element Method is a numerical technique for finding approximate solutions to boundary value problems. A boundary value problem is a differential equation together with a set of additional restraints, called boundary conditions. FEM uses various methods to minimize an error function and produce a stable solution [2].

Fig. 5: Solid model of helical gear

Fig. 6: Meshed 3-D model of helical gear
IV. LEWIS EQUATION USED TO CALCULATE BENDING STRESS

The bending stress is one of the crucial parameters during the analysis of helical gears. When the total repetitive load acting on the gear tooth is greater than its strength then the gear tooth will fail in bending. According to Lewis equation, the Beam Strength of helical gear tooth is given by

\[ \sigma_b = \frac{F_l}{b y_p P_c \cos(\psi)} \]  \hspace{1cm} (15)

<table>
<thead>
<tr>
<th>Helix angle (Degree)</th>
<th>Bending Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>13</td>
<td>214.98</td>
</tr>
<tr>
<td>18</td>
<td>206.98</td>
</tr>
<tr>
<td>23</td>
<td>201.70</td>
</tr>
<tr>
<td>28</td>
<td>196.86</td>
</tr>
<tr>
<td>33</td>
<td>194.82</td>
</tr>
</tbody>
</table>

Table 4: Theory Result for Bending Stress by Using Lewis Equation

<table>
<thead>
<tr>
<th>Helix angle (degree)</th>
<th>von-Mises stress for Lewis (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>13</td>
<td>215.56</td>
</tr>
<tr>
<td>18</td>
<td>207.58</td>
</tr>
<tr>
<td>23</td>
<td>202.32</td>
</tr>
<tr>
<td>28</td>
<td>197.49</td>
</tr>
<tr>
<td>33</td>
<td>195.45</td>
</tr>
</tbody>
</table>

Table 5: Theory Result for Bending Stress by Using Lewis Equation

V. FEM ANALYSIS USED TO CALCULATE THE BENDING STRESS

Helical gear assembly was imported in Ansys 14.5 and the boundary conditions were applied to the gear model. The fixed support was applied to the inner rim of the pinion and torque which the value is 37.054Nm is applied to the helical pinion in anticlockwise direction. The model was analysed for the root bending stress for the applied tangential force (2694.836N). Only 3-D analysis was performed because of the helical profile of its teeth.

Figure 9 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 215.32 MPa.

Figure 10 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 208.17 MPa.
Figure 11 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 201.63 MPa. Figure 12 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 200.11 MPa. The maximum von-Mises stress developed by FEA is almost nearer to the calculated von-Mises stress.

Figure 13 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 193.28 MPa.

In this section, the modelled helical gear is analyzed to study the effect of helix angle on bending stress under static load with different parameters. Throughout the analysis, each gear is studied for five different helix angle ($\psi=13^\circ, 18^\circ, 23^\circ, 28^\circ, 33^\circ$). Face width, gear ratio, speed, module and the applied load are kept constant.[2] From the Figure 14, there is a variation in the maximum bending stresses with the change in helix angle. The maximum bending stress value decreases with the increase of helix angle. When comparing, Lewis values are little lower than the ANSYS values. Error percentages are around 1%.

Helix angle is important geometrical parameter in determining the state of stresses during the design of gears. The helix angle $\psi$, is always measured on the cylindrical pitch surface. It ranges between 10° and 45°. Commonly used values are 15, 23, 30 or 45°. Lower values give less end thrust. Higher values result in smoother operation and more end thrust. In single helical gears, the helix angle ranges from 20° to 35°, while for double helical gears(herringbone gears), it may be made up to 45°[5]. Keeping the face
width, gear ratio, speed, module constant and for variation of helix angle, the von-Mises stress decreases linearly. The helix angle 23˚, corresponding to the range of helix angle is taken for further optimization.

REFERENCES


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